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Exergy-based Comparative Assessment of "Zero CO_2 Emission" Coal Gasification Processes Feeding H_2 –Fueled Power Plants

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Abstract

A modular Process Simulator, CAMEL®, developed by the University of Roma 1, has been applied to analyse a series of "zero emissions" high efficiency cycles. This paper compares three different cycles integrated with CO_2 separation technologies based on chemical or physical absorption *upstream* of the combustion process: pure hydrogen is burnt in presence of pure oxygen to produce superheated steam. All solutions are based on non-conventional plant configurations: two of them are H_2/O_2 cycles and the third one is the so called ZECOTECH® cycle. The main features of all three configurations are presented and their thermodynamic cycles are simulated in order to perform an exergy analysis.

Keywords: *Process simulation, Zero emission cycles, exergy analysis*

1. Introduction

The need to cover an ever-increasing final energy demand and the consciousness of the necessity of setting a limit to the related emissions are prompting design engineers to concentrate their attention not only on the repowering of technically obsolete power plants but also on the development of "innovative" solutions.

A long time perspective directs these studies towards the possibility of using renewable and "clean" energy sources. If one considers the present world situation (where energy resource availability, state-of-the-art of the energy conversion technologies and foreseen developments in the short and medium term scenarios are concerned), it is immediately clear that the energy resources that can be realistically exploited for large scale production are still those of fossil origin, in particular coal. One of the problems related to the use of fossil fuels is of course the technology employed for the fuel treatment and combustion: any process involving a carbon-rich fuel implies CO_2 emissions in the atmosphere. Several removal and sequestration techniques have been proposed. Some are based on chemical or physical fuel treatment upstream of the power cycle, which is fed by a carbon-free syngas (for example coal gasification with production of H_2 and CO_2 removal through MEA or $CaCO_3$-methods; natural gas decarbonization. See Calabrò et al., 2004; Ertesvåg et al, 2005; Lozza et al., 2002). Other techniques are based on CO_2-capture in an intermediate phase of the power cycle (Anderson et al., 2004).

In the present paper, the energetic and exergetic performance of some of the so-called "zero emission" coal-based cycles are analyzed. All process calculations are performed by means of a modular process simulator, CAMEL®, originally developed by the authors' group at the Mechanical and Aeronautical Engineering Department of the University of Roma 1 "La

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Sapienza" (Falcetta et al., 1995, 1998) and further refined and extended in the last years (Fiorini et al., 2005 a and c). The work is the result of three consecutive stages:

1) Updating of the code so that it can perform a comparative analysis of innovative cycles that had been studied by means of different codes. In this step the selected cycles have been used as a "test bench" to verify the performance of CAMEL®. Since these "test cases" had been simulated before, and their respective plant configurations are the results of accurate analyses published in archival references (Calabrò et al., 2004; Gambini et al., 2003), only a brief description thereof will be given here: we refer the reader to the above references for all the details concerning their process design and optimization.

2) Simulation of the cycles and testing of the code: the results of our calculations have been compared with the available results of the simulation performed with other codes by the designers of the plants: these original calculations have been taken as a reference data set. This stage has been successfully completed, and CAMEL® now allows for the simulation of a sufficient number of different types of power plants. The integration of additional modular elements is presently in the making, with the goal of extending similar comparisons to the entire process, including all the connected sub-processes (coal gasification, CO_2 sequestration, O_2 production) that are not explicitly simulated at present.

3) After the code was satisfactorily implemented and tested, a specific utility that performs an exergetic analysis has been added, which will be discussed here in some detail in the following.

2. The Processes Selected for the Analysis

The three plant configurations examined here enact an internal combustion steam cycle fed by hydrogen: the oxidizer being pure oxygen, superheated steam the product of the combustion, while cogenerated steam is injected in the combustion chamber as "inert", to control the gas temperature and to limit oxygen consumption. The selected cycles are:

1-2) Two H_2/O_2 cycles, case "1" (T_{max} = 1350°C, *Figure 1*) and "2" (T_{max} = 1700°C, *Figure 2*). Both were proposed and discussed in (Gambini et al., 2003).

3) The ZECOTECH® cycle, jointly developed by the Energy Department of ENEA (Italian National Agency for New Technologies, Energy and Environment) and the Ansaldo Group, in cooperation with a consortium of Italian Universities (Calabrò et al., 2004) (*Figure 3*).

3. Description of the Process Layouts

The proposed plant layouts are described in this section: TABLES I-IV contain the values of the most relevant thermodynamic parameters evaluated during the simulations.

3.1 Cycles n° 1 and 2 (H_2/O_2 cycles)

The two examined cycles result in similar plant layouts. While cycle n° 1 is designed in accordance with the present or near-future state of the art (T_{max}=1350°C and β=30), cycle n° 2 is designed by considering medium and long term possible developments in gas turbine technology (T_{max}=1700°C, β=70) (Okamura et al., 2000).

Cycle n° 2 differs from cycle n° 1 with the addition of a low pressure section (steam turbine and condenser) downstream of the HRSG.

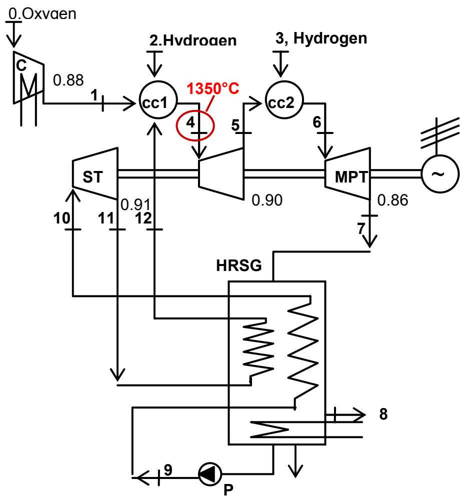

Figure 1. Plant layout of Cycle n°1.

3.2 Cycle n° 3 (ZECOTECH®)

As a result of a previous optimization (Calabrò et al., 2004), this process was somewhat modified from its original configuration: we consider here only the latest "optimal" plant configuration (*Figure 3*).

The sub-processes upstream of the power unit (gasifier and carbonator) limit the pressure in the combustion chamber to 30 bar. The MPT outlet pressure ($p_{out,MPT}$) was found to be one of the most influential parameters on cycle efficiency; for this reason, as a first step in this work, a sensitivity analysis was carried out to quantify the dependence of the cycle efficiency on the variation of $p_{out,MPT}$ for different T_{max}.

TABLE I. RELEVANT THERMODYNAMIC PARAMETERS FOR CYCLE n° 1.

	m [kg/s]	p [bar]	T [K]	h [kJ/kg]	ex [kJ/kg]
0	1.000	1.01	298.15	0	122
1	1.000	33.60	633.75	332	501
2	0.0601	31.30	553.15	4079	125272[1]
3	0.0411	5.64	513.15	3494	122898
4	3.515	30.40	1623.15	4919	2605
5	3.515	2.00	1050.15	3615	1228
6	3.556	2.02	1623.15	5293	2439
7	3.556	1.07	1473.15	4912	2037
8	1.100	1.01	370.75	2182	388
9	2.455	346.0	315.95	209	36
10	2.455	294.0	838.15	3346	1517
11	2.455	36.70	541.15	2884	1020
12	2.455	33.80	773.15	3451	1317

TABLE II. RELEVANT THERMODYNAMIC PARAMETERS FOR CYCLE n° 2.

	m [kg/s]	p [bar]	T [K]	h [kJ/kg]	ex [kJ/kg]
0	1.000	1.01	298.15	0	122
1	1.000	77.50	634.65	332	567
2	0.0765	71.80	501.15	3321	125975
3	0.048	17.40	501.15	3321	124218
4	3.170	69.60	1973.2	5918	3546
5	3.170	7.40	1402.2	4511	2043
6	3.218	7.20	1973.2	6418	3449
7	3.218	1.07	1493.2	5127	2108
8	3.218	1.01	479.15	2837	543
9	3.218	0.051	305.7	2423	60
10	2.094	346.0	309.75	184	35
11	2.094	294.0	838.15	3346	1517
12	2.094	84.10	643.15	3043	1198

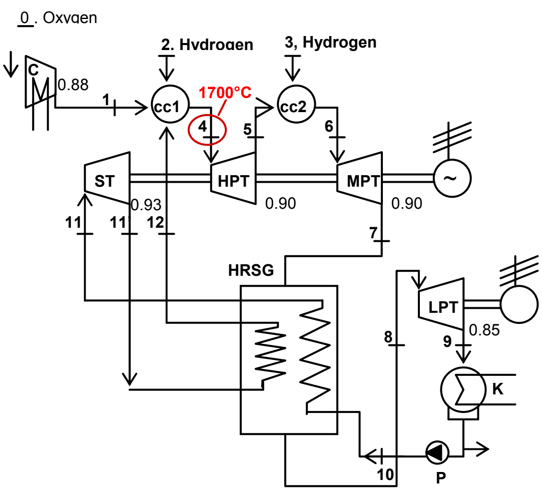

Figure 2. Plant layout of Cycle n°2.

[1] The fuel "total exergy" is evaluated considering both the chemical exergy (assumed, in first approximation, equal to its LHV; see Baehr, 1979; Baehr & Schmidt, 1963) and the physical exergy (determined by pressure and temperature).

Throughout the tests, the following assumptions were made:

a) Turbine and compressor adiabatic efficiencies are a function of the total pressure ratio and of the consequent number of stages as reported in equations (1) and (2), (Sciubba, 2002). For the expansion and compression ratio of individual stages, the value of 2 for the turbine and of 1.25 for the compressor were assumed.

For a turbine we have:

$$\eta_{ad,TOT} = \frac{1-\left[1-\eta_s\left(1-\beta^{(1-k)/Nk}\right)\right]^N}{1-\beta^{(1-k)/k}} \qquad (1)$$

whereas for a compressor:

$$\eta_{ad,TOT} = \frac{\beta^{(k-1)/k}-1}{\left[1+\left(\beta^{(k-1)/Nk}-1\right)/\eta_s\right]^N-1} \qquad (2)$$

The stage adiabatic efficiency was assumed equal to 0.90 for the turbine and to 0.89 for the compressor.

b) Pressure losses in the piping have been neglected; in the exchangers and in the combustion chamber they are equal to 2%.

c) Combustion energy efficiency is equal to 0.98.

d) The hydraulic efficiency of the pump is equal to 0.85.

e) The energy efficiency of the generator is equal to 0.98.

f) The coolant used in the intercooler and in the condenser is water at ambient conditions (298.15 K and 1.01 bar).

Figure 4 shows the results of the analysis: the optimal MPT discharge pressure decreases with increasing T_{max}: for T_{max}=1350°C the optimal pressure is equal to 1.5 bar, while for T_{max}=1700°C it is only slightly less than 1 bar.

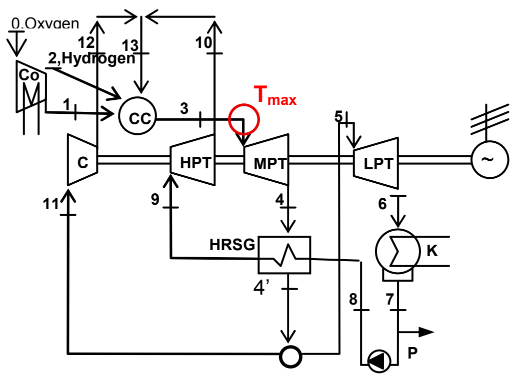

Figure 3. Plant layout of the ZECOTECH® cycle.

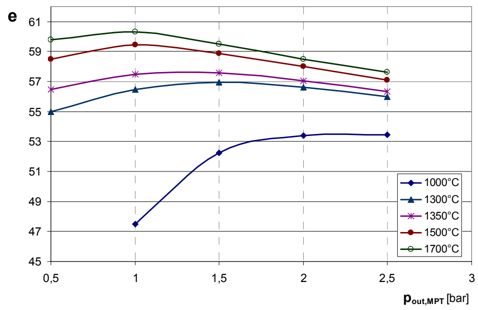

Figure 4. Exergetic efficiency as function of $p_{out,MPT}$ (Cycle n°3).

The following simulations of cycle n° 3, whose results are reported in TABLES III and IV, have been performed with T_{max} respectively equal to those reached in cycles n° 1 and 2 (and with the optimal values of $p_{out,MPT}$), to compare the overall performance.

4. Mass and Energy Balances

On the basis of the results of the simulations, the so-called First Law performances have been calculated: they are reported in TABLE V. For better comparison, the results obtained for the three plants are all referred *to the unit of oxygen consumed.* The LHV of the hydrogen is assumed equal to 120 MJ/kg.

In addition to the analysis performed with the simulations, the influence of the ASU and of the CO_2-capturing on the performances of the cycles has been estimated. In particular, the synergies between the coal-gasification unit and the power plant have been analyzed in a parallel study conducted for the ZECOTECH® cycle (Fiorini et al., 2005 c): considering a cryogenic air separation unit; the power consumption is equal to 800 kJ/kg_{O2} (Aceves et al., 2003). For the CO_2 sequestration, the $CaCO_3$-technology is adopted: with an operating pressure respectively of 1 bar in the calciner, 30 bar in the carbonator and a CO_2 disposal pressure of 80 bar, the energy required for the compression is equal to 300 kJ/kg_{CO2} (Fiorini et al., 2005). A coal-hydrogasification process has been also analyzed in order to estimate the total O_2 consumption and CO_2 production resulting from the hydrogen synthesis. Accounting for the exergy losses in the gasifier, in the reformer and in the calcinator (equal to 18% of the total fuel exergy input, considering the coal LHV equal to 29 MJ/kg), the resulting efficiency is 49.6% (cycle n° 3, 1350°C, β=30). A thermo-economic analysis of cycle n° 3 is presently in the making and shall be the object of a future publication.

5. Exergy Analysis

TABLE III. RELEVANT THERMODYNAMIC PARAMETERS FOR CYCLE n° 3, T_{max}=1350°C.

	m [kg/s]	p [bar]	T [K]	h [kJ/kg]	ex [kJ/kg]
0	4.498	1.01	298.15	0.0	122
1	4.498	30.00	633.00	332	492
2	0.5622	30.00	553.15	4079	125272
3	32.164	30.00	1623.15	6509	2853
4	32.164	1.50	985.62	4369	1224
5	14.744	1.50	518.35	2964	634
6	14.744	0.05	306.03	2488	60
7	14.744	0.05	305.38	135	0.3
8	9.684	170.00	305.81	152	17
9	9.684	170.00	833.00	3455	1528
10	9.684	30.00	578.84	3007	1055
11	17.419	1.50	518.35	2964	634
12	17.419	30.00	1062.15	4347	1751
13	27.103	30.00	871.72	2841	1295
Adiab. efficiency: C:0.85; MPT:0.90; HPT:0.91; LPT:=0.87					

TABLE IV. RELEVANT THERMODYNAMIC PARAMETERS FOR CYCLE n° 3, T_{max}=1700°C.

	m [kg/s]	p [bar]	T [K]	h [kJ/kg]	ex [kJ/kg]
0	4.498	1.00	298.15	0.0	122
1	4.498	30.00	633.00	332	492
2	0.5622	30.00	553.15	4079	125272
3	20.259	30.00	1973.15	6509	3660
4	20.259	1.00	1161.31	4369	1465
5	13.677	1.00	518.35	2964	579
6	13.677	0.05	306.03	2488	61
7	13.677	0.05	305.38	135	0.3
8	8.617	170.00	305.81	152	17
9	8.617	170.00	833.00	3455	1528
10	8.617	30.00	578.84	3007	1054
11	6.582	1.00	518.35	2964	579
12	6.582	30.00	1156.93	4347	1917
13	15.199	30.00	805.89	2841	1129
Adiab. efficiency: C:0.84; MPT:0.89; HPT:0.91; LPT:=0.87					

TABLE V. FIRST LAW EFFICIENCY COMPARISON.

	Cycle1	Cycle2	ZECOTECH®	
T_{max} [°C]	1350	1700	1350	1700
b	30	70	30	30
P [kW]	6299	9582	8864	9300
$\dot{m}_{fuel}\left[kg/s\right]$	0.1012	0.1245	0.1250	0.1250
η=P/ Fuel Energy	**50.26**	**61.30**	**57.20**	**60.10**

One of the purposes of this work was to perform an explicit exergy analysis of the plant, to identify - for each process configuration - the components that are affected by the highest irreversible losses, and for which a more accurate analysis ought to be made in the design phase. In fact, the results of an exergy analysis provide the designer with better insight as to where a design modification is necessary, both at the single component and at a process level, to better exploit the available resources. For each cycle the following quantities were computed (Bejan et al., 1996; Szargut et al., 1988):

a. The exergetic efficiency of each component, ε:
$\varepsilon = E_U / E_F$
with E_U and E_F respectively the useful product and the resource inflow of the component, evaluated in exergetic terms.
b. The exergy destroyed in each component, E_d (kW), and in the whole plant, $E_{d,tot}$ (kW).
c. Two dimensionless loss parameters:
 - $E_d\% = E_{d(component)} / E_{d,tot} \times 100$
 i.e., the ratio, expressed in %, of the exergy destroyed in a component to the total exergy destroyed in the plant.
 - $E'_d\% = E_{d(component)} / E_F \times 100$
 i.e., the percentage of the total exergy resource influx into the process that is destroyed in the component.
d. (Only for cycles n° 1 and n° 3, both with T_{max}=1350°C and β=30) The so-called coefficients of structural bond ("CSB"), defined by equation (3) (Beyer et al., 1974; Kotas, 1995).
π represents the ratio between the variation of the exergy destroyed in the process ($E_{d,tot}$) to that destroyed in the k^{th} single component (E_k) if *the process parameter x_i* (and only it) is varied. The CSBs measure the influence of each single component on the overall plant performance when some of the working parameters are changing. It is useful to recall the values π can assume and their meaning:

$$\pi_{k,i} = \frac{\left(\dfrac{\partial E_{d,tot}}{\partial x_i}\right)}{\left(\dfrac{\partial E_k}{\partial x_i}\right)} = \left(\frac{\partial E_{d,tot}}{\partial E_k}\right)_{x_i = var} \approx \left(\frac{\Delta E_{d,tot}}{\Delta E_k}\right)_{x_i = var} \quad (3)$$

– $\pi_{k,i} > 0$: when the parameter x_i is changing, to a decrease in the exergy destruction in the k^{th}-component corresponds to a decrease in the total exergy destruction of the plant. A value much higher than one can be explained as follows:
 1. Considering the third term in Equation (3), a small variation of E_k corresponds to a large variation of $E_{d,tot}$;
 2. Considering the second term in Equation (3), a high value of $\pi_{k,i}$ means that the derivative $\partial E_k/\partial x_i$ is much lower than $\partial E_{d,tot}/\partial x_i$: the parameter x_i has a much lesser influence on the exergy losses of the k^{th}-component than it has on those of the whole plant.

 When $0<\pi_{k,i}<1$, to a decrease of E_k corresponds to a proportionally lower decrease of $E_{d,tot}$: this is due to the fact that other components at the same time are causing a $\Delta E_{d,tot}$ of opposite sign than the one caused by the k^{th}-component.
– $\pi_{k,i} = 0$: the k^{th}-component does not influence the variation of $E_{d,tot}$.
– $\pi_{k,i} < 0$: when the parameter x_i is varied, to a decrease in the exergy destruction in the component *k* corresponds to an increase in the total exergy destruction of the plant. The more negative $\pi_{k,i}$, the higher the total increase of the irreversibility.

5.1 Cycles n° 1 and 2 (H_2/O_2)

For the intercooled compressor the analysis has been performed first on each single stage and the connected intercooler, and then for the multistage machine as a whole, in order to evaluate the effect of intercooling on the exergetic efficiency. It is well known that intercooling reduces the overall power absorption, but the exergetic cost of the use of an external coolant is such that the total efficiency decreases: individual stages have an ε higher than 90% and intercooling decreases this value to 85.75%. The thermodynamic reason is clear: heat is transferred to a cooler medium (the cooling water) and dispersed in the environment with no further recovery. However, it must be considered that the ratio between the power gain obtained with the intercooling (ΔP) and the exergetic cost associated to it ($\Delta ex_{coolant}$) is much higher than one: ($\Delta P = P_{non\ interc} - P_{interc}$= 496- 442 = 54 kW; $\Delta ex_{coolant}$ = 3.5 kW), and therefore there is a global advantage, at process level, in adopting an intercooled configuration.

TABLE VI. RESULTS OF THE EXERGY ANALYSIS OF CYCLE n° 1.

	ε	**E_d[kW]**	**E_d%**	**E'_d%**
C (I)	91.24	17.8	0.36	0.14
C (II)	92.73	18.5	0.38	0.15
Intercooler	5.02	38.8	0.79	0.31
Interc. C	85.75	75.1	1.53	0.60
HPT	91.86	394.1	8.05	3.13
MPT	91.93	115.3	2.35	0.92
ST	90.17	120.0	2.45	0.95
PUMP	70.74	35.0	0.71	0.28
HRSG	73.95	1358.3	27.73	10.80
cc1	72.02	2106.5	43.00	16.74
cc2	86.25	694.4	14.18	5.52
$E_{d,tot}$	**-**	**4898.6**	100.00	38.94

The highest loss, in percentage, is found, as expected, in the two combustion chambers and in the heat-recovery steam generator (HRSG). The former enacts an irreversible process (the transformation of chemical into thermal exergy): the lower the temperature of the reactants, the higher the exergy losses. The temperature of the mixture steam - O_2 at the inlet of cc 2 is 1050°C versus 500°C at the inlet of cc1, and the value of E_d for the latter component is in fact much higher than that for cc 2.

In the HRSG the losses are, percentage-wise, even higher than in cc 2: such a high value is caused by the high temperature difference between the two streams inside the heat exchanger: both the TTDs and the LMTDs for the two media (gasification with superheating and re-superheating) are very high:

$LMTD_1$ (gas, sh)=236 °C; TTD_1 (gas,sh)= 615°C;

$LMTD_2$ (rh) = 200 °C; TTD_2 (rh) = 700°C.

Such an analysis *directly suggests a process modification:* that of substituting the hypercritical boiler with a subcritical one with more than one pressure level. With this modification, the thermal profiles of the two streams are shifted towards lower LMTDs.

For cycle n° 2 the typology and the distribution of the losses are similar to those of cycle n° 1, but globally it reaches a higher efficiency (ε = 61.3%).

5.2 Cycle n° 1: evaluation of the CSB

In this section we shall examine the modification of the plant operating conditions with a variation of the top cycle temperature.We shall do this by evaluating the values of the CSBs for each component. All remaining working parameters imposed as input in the simulation have been maintained constant, while T_{max} is changed about its nominal value of 1350°C (±10%). The analysis has been performed considering equal exergetic input for each run (equal fuel consumption); all the variations induced in the values of the working parameters not assigned as an input (like for instance the steam mass flow rate) have been recorded in order to evaluate π.

It is well known that increasing the top cycle temperature leads to an increase of the overall plant efficiency; this result is also recovered in our exergy analysis, because the total losses decrease. The analysis of the signs and of the magnitude of CSB are useful to examine which components provide a higher contribution to this improvement and which ones, on the contrary, present a trend of their losses E_k discordant with the $E_{d,tot}$: as recalled above; a negative value of CSB means that the optimization of the given component cannot be conducted by varying the parameter x_i (in this case T_{max}), because this causes negative effects on the overall plant performance. *Figure 5* presents the trends of the CSB for the components more affected by the variation of T_{max}: the intercooled compressor is insensitive to this parameter while the HPT, the MPT and the LPT have a CSB >>1, which reflect their secondary influence on the value of $E_{d,tot}$.. It is interesting to investigate the CSB of the other components; we can observe the following:

a) Combustion chambers: they present opposite signs of CSB: to an increase of T_{max} corresponds to a decrease of the exergy destruction ($CSB_{cc1,Tmax}>0$) for the first cc and an increase for the second one ($CSB_{cc2,Tmax}<0$). This can be justified considering that for the same total fuel consumption, the necessity of reaching higher temperature for the combustion products has opposite consequences on the streams that flow through the two chambers. In cc1 there is a *decrease* of the fuel consumption and a higher T_{max} is obtained by injecting a lower flow rate of steam as inert: limiting the fuel mass flow rate leads here to a decrease of the combustion irreversibility rate. In cc 2, even if the reactants enter with a higher value of enthalpy, an increase of the temperature of the products can be obtained only by *increasing* the specific fuel consumption: consequently, the irreversibility of the chemical-to-thermal conversion increases as well.

b) HRSG: CSB<0, and its losses are always increasing with $T_{max.}$ This trend is due to the departure of the thermal profiles of the two streams, because of the fact that the hot gas outlet temperature (section n° 8 in *Figure 1*) is maintained equal for each test (to ensure equal inlet conditions into the pump) while the inlet temperature is increased. Consequently, the total heat transfer per unit of mass, q, is increasing as well. The outlet temperatures of the SH and RH steam flows (points 10 and 12) are maintained fixed because they represent respectively the value of the TIT for the ST and the temperature of the steam injected in the cc. This means that the excess q is exchanged with the external coolant in the phase of condensation and undercooling of the hot steam, introducing higher irreversibilities.

It is clear that, in a global perspective, a higher value of the top cycle temperature has positive effects on the plant performance; the negative value of the CSB characteristic of the HRSG is in fact more than compensated by the value of the coefficients for the other components.

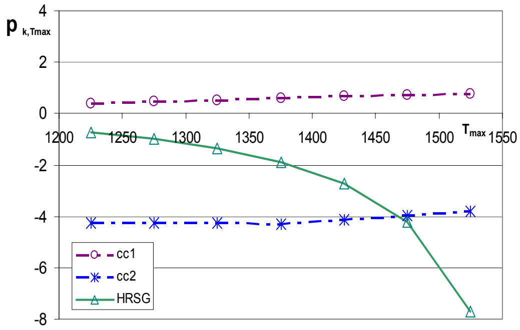

Figure 5. CSB for the components of cycle 1.

Such an *analysis can guide the designers' attention of the HRSG design*: it is the only element that suffers from higher values of T_{max}, and finding new solutions for its design could allow for a better exploitation of the technological efforts connected with the attainment of higher TIT.

5.3 Cycle n° 3 (ZECOTECH®)

The highest contribution to the total exergy destruction is again provided by the cc, the only one, in this case, of the plant. The losses in the HRSG on the contrary are much lower with respect to Cycles n° 1 and 2, even if a direct comparison makes little sense due to the different connectivity of the processes.

The thermal profiles of the two streams are closer for the case 1350°C (LMTD=180°C) than for the case 1700°C (LMTD=260°C), and in this latter case the losses of the component are consequently higher (E_d=20.5%).

Percentage wise, the losses for the MPT are quite high; this is the turbine with the highest outlet temperature and also the highest efficiency; such high losses can be explained considering that while the MPT processes the entire mass flow rate of steam, only a fraction of this mass flow rate flows in the other two turbines and in the compressor: 46% of the total flow rate is directed to the LPT and evolves in the HPT as well, while the remaining 54% constitutes the feed of the compressor.

5.4 Cycle n° 3: evaluation of CSB

We evaluated the CSB for a variation of T_{max} for this plant as well. *Figure 6* reports the results obtained by changing T_{max} around its nominal value and maintaining $p_{out,MPT}$ at the optimal value (1.5 bar).

As expected, increasing the top cycle temperature leads to an improvement of the overall efficiency, and it is accompanied by a decrease in the total exergy losses.

The components not displayed are those that are less influenced by a variation of T_{max} (LPT, HPT, K): the conditions of the streams at their inlet and outlet in fact remain constant during the tests (the TIT and expansion ratio of these two turbines are fixed as well as the outlet temperature of the condensed water), so that their decreasing losses are due only to the decrease of the steam mass flow rate in the plant.

TABLE VII. SYNTHESIS OF THE EXERGETIC ANALYSIS OF CYCLE n° 3-T_{max}=1350°C.

	ε	**E_d[kW]**	**E_d%**	**E'_d%**
C	95.58	900.3	3.61	1.33
HPT	93.77	285.5	1.15	0.42
MPT	96.27	1956.9	7.85	2.90
LPT	90.72	785.3	3.15	1.16
PUMP	83.50	32.0	0.13	0.05
K	-	644.5	2.59	0.96
HRSG	77.11	4344.2	17.43	6.44
cc	77.31	15977.9	64.10	23.68
$E_{d.tot}$	**-**	**24926.6**	100.00	36.95

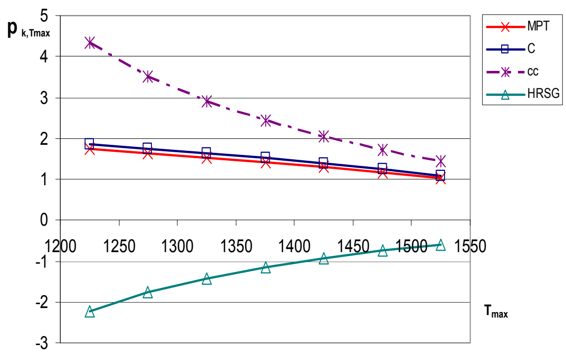

Figure 6. CSB for the components of ZECOTECH® cycle, with a variation of T_{max}.

The components which present CSBs closer to one are more sensitive to a variation of T_{max}, and this is the case for cc, MPT, HRSG and C (*Figure 6*). The cc and the MPT are directly influenced by T_{max}, which represents the gas outlet temperature of the burner and the inlet temperature of the downstream turbine; the compressor is influenced only by the decrease of

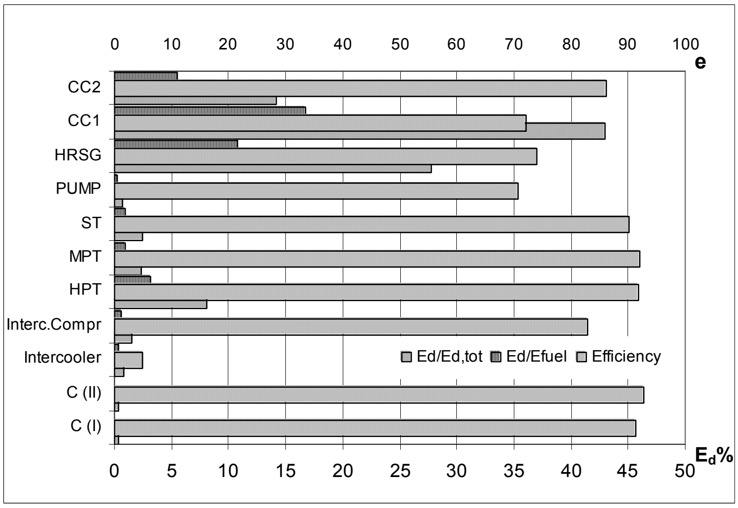

Figure 7. Cycle n° 1, T_{max}=1350°C.

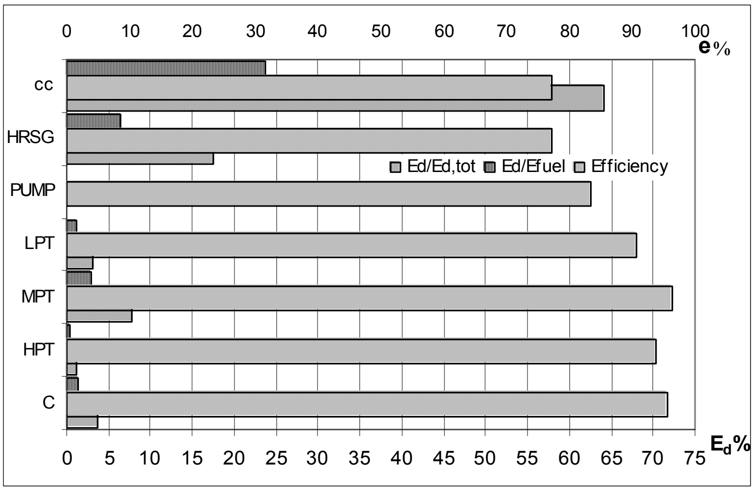

Figure 8. Cycle n° 3, ZECOTECH® (T_{max}=1350°C).

the steam mass flow rate (that is necessary to guarantee, with equal fuel consumption, a higher top cycle temperature), but more than the other components already analyzed: the decreasing steam rate is higher for the stream directed to the compressor than for the one fed into the LPT. In this plant, too, the HRSG has a negative CSB. It is possible to observe that the total heat transfer decreases by increasing T_{max}: the inlet condition of the pressurized water (section 8, *Figure 3*) and the outlet temperature of the SH-steam (section 9) both remain constant, while the water mass flow rate decreases. The hot gas outlet temperature (section 4') is fixed too (because it coincides with the fixed LPT inlet temperature), while the inlet gas temperature (section 4) is increasing with T_{max}: the heat transfer is lower with respect to the base case because of the contemporary decreasing of the hot gas mass flow rate. Consequently, the growing irreversibility rate is due to an excessive distance between the thermal profiles of the two streams (excessive LMTD).

Figures 7 and *8* show the results of the exergy analysis for cycles n° 1 and 3: the percentage of the fuel exergy destroyed within each component ($E_d' = E_d/E_{fuel}$), the percentage of $E_{d,tot}$ due to each component and their efficiency are reported.

6. Conclusions

The importance of the analysis of "innovative power cycles" lies in the fact that there is general agreement on the fact that energy (exergy) resources ought to be exploited with the maximum possible efficiency compatible with the concepts of sustainability and eco-compatibility. The H_2-fed cycles examined here are certainly more eco-compatible and sustainable than current fossil-fueled plants, but to define them as "high efficiency cycles" does not appear proper. The isolated power plant sections of these processes attain very high efficiencies, but the analysis ought to be carried out *considering all of the sub-processes* *connected with the power generation unit:* fuel treatment, gasification of coal, oxygen separation, CO_2 sequestration and removal. The ZECOTECH® plant attains the highest performance in the case of T_{max}=1350°C and β=30 (compared to cycle n° 1), and an efficiency only slightly lower in the case of T_{max}=1700°C and β=30 (compared to cycle n° 2, that however has an higher maximum pressure, β=70), managing to reach an effectiveness of 49.6 (case 1350°C) even considering the exergy losses connected to the gasification and removal processes.

The exergy analysis can highlight possible ways to improve the plant design: the results indicate the combustion chambers and, particularly, the HRSG as the components which need more attention in the choice of their working conditions.

From a technical point of view, environmental and economic issues must be brought into the picture. This can be done only by either a modified thermo-economic analysis or an extended exergy accounting. Both methods are outside of the scope of the present paper.

Nomenclature

C	Compressor
cc	Combustion chamber
CSB	Coefficient of structural bond
E_d	Exergy destroyed in a component [kW]
E_F	Exergy resource [kW]
E_U	Useful exergy production [kW]
ex	Specific exergy [kJ/kg]
h	Specific enthalpy [kJ/kg]
HPT	High pressure turbine
HRSG	Heat recovery steam generator
K	Condenser
k	Specific heat ratio
LHV	Lower heating value [kJ/kg]
LMTD	Logarithmic mean temperature difference [°C]
LPT	Low pressure turbine
m	Mass flow rate [kg/s]
MPT	Medium pressure turbine
N	Number of stages
P	Net power output [kW]
$p_{out,MPT}$	Medium pressure turbine outlet pressure [bar]
q	Heat transfer for unit of mass [kJ/kg]
ST	Steam turbine
T	Temperature [K]
Tmax	Top cycle temperature [°C]
TTD	Terminal temperature difference [°C]
β	Total pressure ratio
ε	Exergetic efficiency
η	Energetic efficiency
η_s	Stage adiabatic efficiency
$\pi_{k,i}$	Coefficient of structural bond for component k, with the variation of the parameter x_i

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NOTE

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