THERMOECONOMIC ANALYSIS AND OPTIMIZATION OF A GAS TURBINE PLANT COMBINED WITH AN ABSORPTION UNIT

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ABSTRACT

The aim of the paper is to study the performance of a power plant for the combined production of electrical, thermal and cooling thermal energy.

The exergy analysis was developed from the system's operating conditions measured in a previous experimental phase, and allowed description and quantification of causes of efficiency loss in the plant.

The following thermoeconomic analysis, based on the exergy balance, allowed appraisal of the actual costs of each component and possible optimization of the plant for higher efficiency and cost saving.

The thermoeconomic results lead to a better understanding of the influence of off-design operating conditions on the performance of the whole plant and on this basis further improvements and modifications are envisaged.

Three modifications of the plant layout are described and discussed, in greater detail for the most promising of them, i.e., compressor inlet air cooling with absorber excess cooling power production.

Results show that this solution is particularly effective in the present case, not only from the energetic point of view, but, as is not always the case, also form the economic one.

The application of thermoeconomic analysis to the pharmaceutical factory under study has the aim of identifying those components which have the highest cost quantifying losses in cost terms.

NOMENCLATURE

c' unitary cost connected with exergetic loss
cc unitary capital cost
Cc capital cost
ce unitary exergetic cost as heat
cf unitary exergetic cost as fuel
Cf exergetic cost as fuel
Cf_n fuel cost per volume unit: $/Nm^3
cs unitary service cost (maintenance and running)
Cs service cost: (maintenance and running)
ct total unitary cost per exergetic unit produced
Ct total cost $(Cf + Cc + Cs)$
Ea useful steam exergy produced by heat recovery boiler
Ec inlet exergy as fuel
Ef useful cold water exergy produced by absorber
Las exergetic loss in absorber
Lcaux exergetic loss in auxiliary boiler
Lcc exergetic loss in gas turbine combustion chamber
Lc1 exergetic loss at first compressor stage
Lc2 exergetic loss at second compressor stage
Le expansion exergetic loss
Leco exergetic loss in economizer
Leva exergetic loss in evaporator
Lgen exergetic loss in generator
Lr1 exergetic loss in first interrefrigerator exchanger
Lr2 exergetic loss in second interrefrigerator exchanger
Lsc exergetic loss in last hot gas side exchanger
Lst exergetic loss in stack
Wel electric power
Wg electric power produced by gas turbine
η efficiency

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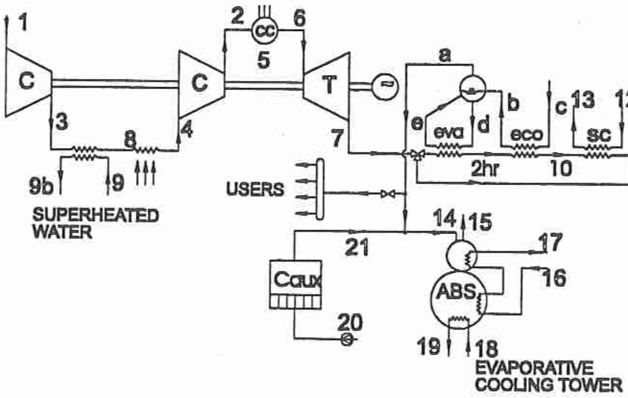

FIGURE 1 : Technical Schema for Plant Simulation

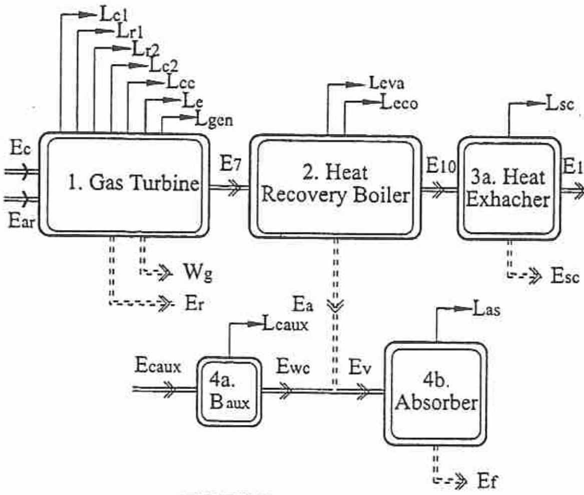

FIGURE 2 :Exergetic Scheme

PLANT DESCRIPTION

The plant, described in detail by De Lucia et al. (1997), is located in a pharmaceutical factory near Frosinone in Central Italy. It was initially composed of five compression cooling units and two boilers. With the aim of reducing energy consumption, in 1994 a cogeneration system was built. A gas turbine cogeneration plant for production of power and thermal energy was coupled with a two-stage absorption unit, fueled by the steam produced in HRSG, for production of cooling thermal energy. The plant layout is shown in Figure 1.

The turbine is a heavy-frame gas turbine with a net electric power of 1000 kW in ISO conditions, and it has a two-stage centrifugal compressor with interrefrigerator.

Interrefrigeration is carried out by two heat exchangers. The first one supplies superheated water at 120-130°C. The remaining thermal energy, being at a temperature too low for use in the industrial processes, is dissipated in the second heat exchanger.

The HRSG on the exhaust gases has a single pressure level and is a vertical type one with water tubes. At the boiler exit, a further low-temperature recuperator on the gas circuit is for hot water (85°C) production for sanitary and ambient heating purposes.

Steam produced in the recovery boiler is used both for industrial processes and as a hot source for the absorber unit. When recovery boiler production cannot totally satisfy high steam demand two auxiliary gas-fueled boilers are used, which could supply all steam requirements, in case of cogeneration plant faults or during maintenance periods.

The absorption cooling unit designed for production of cooled water down to 7°C starting form 12°C is a two-stage machine, using steam at 8 bar and 180°C as high temperature heat source.

The absorber is connected to two basins with a capacity of 75 m^3 each, one containing "hot" water returning from the process, the other containing "cold" water from where it is then drawn for production needs. Five compression units and an ice cooling energy storage unit are also connected to the same basins in parallel with the absorber.

The plant was started up in June 1994, and data collected constantly by the data acquisition and monitoring systems. Ever since the plant went on line, there have been no particular problems, and it has always operated according to the pre-established programs. By February 1995, it had accumulated 5000 hours of operating time (at present the operating time is now 15000 hours).

THERMOECONOMY AND EXERGY

The main aim of thermoeconomy accounting is the determination of unitary costs of plant products. This is very difficult when many different products are involved, particularly, in the case of an energetic power plant producing both electric power and heat at different temperatures. To avoid this problem a tool is needed which is able to differentiate the various forms of energy, even in connection with the thermal energy temperature. This analysis must reveal the different qualitative values of the various forms of energy. This can be obtained by mean of the plant energetic balance on the second principle of thermodynamics, through the concept of exergy (De Lucia and Manfrida, 1990).

In most energy systems, the exergy losses fall into five main categories (Manfrida and Stecco, 1992) : chemical reaction (combustion), expulsion of heat into the environment, heat transfer, mixing and friction.

Exergetic Scheme

In exergy/thermoeconomic analysis the parts need to be identified for which the annual exergetic costs are required. Thus the economic behavior of these parts can be examined, even in off-design performance, so as to identify which of them most influence the performance of the whole plant and how they do that. For this purpose actual measured results were compared with those obtained by supposing that the plant was operating in ISO conditions.

The plant scheme can been seen in figure 2 where the plant was split into four main components:

1. GAS TURBINE ;
2. HEAT RECOVERY BOILER ;
3. HEAT EXCHANGER + STACK ;

4. ABSORBER + AUXILIARY BOILER.

In table 1, exergy efficiencies and fuel cost formulation are shown for each plant section.

An important choice which will influence the presentation of results of this analysis is that of considering the absorber, and auxiliary boiler to be closely linked, like an independent plant. It is an easy way to schematize the plant useful for the periods where the total steam required is higher then that produced by the HRSG.

Thermoeconomic Analysis

Table 2 provides a summary of the component capital costs, as an economic characterization of the plant.

The annual investment quota is calculated on the base of a ten year life span at 15% interest. The service and component inlet energy costs (on the basis of 1 $=1500 Italian lira) should be added to this. Service costs include all annual maintenance, repair and running expenses. Maintenance costs are estimated for each component as a percentage of investment. Running costs, (estimated at $33300 per year) only concern the gas turbine which is the main plant component and requires the attention of the majority of staff. Finally the cost of inlet exergy in each component can be calculated once the fuel cost (Cf_n) is known, in this case assumed to be equal to 0.2 $/Nm^3. Thus the exergetic unitary cost is :

$$cf = \frac{Cf_n}{ec_n} \quad [\$/kJ]$$

The total costs are composed by three terms: fuel cost, capital cost and service cost ($C_t = C_f + C_c + C_s$).

The assumed working hours are 1649 for spring, fall, winter and 1099 for summer.

THERMOECONOMIC RESULTS

Thermoeconomic analysis led to calculation of the various component cost items divided into seasons.

To limit elaboration time and allow cost and exergy balance over a whole year, only four weeks were selected (one per season). In this paper, for reasons of space, the results of only the winter and summer weeks are given, since they represent two extremes. This approximation, in the authors' opinion, is consistent with the aim of ascertaining global plant energy and cost performance.
In order to ensure complete plant characterization, considering that the turbine load varies from 90% in day time to 60% at night with reference to the numbers on the plan in figure 1, some calculations were curried out to evaluate the data not recorded by Data Acquisition System (DAS) but necessary in energy evaluation. For more detail see part 1 of this work (De Lucia et al., 1997).

ISO Condition Results

Table 3 shows the thermoeconomic ISO condition results, while losses compared with the other periods are shown in table 6. These values are assumed as reference values for the real operation conditions described below.

COMPONENT	Exergy Efficiency	Fuel Cost
1. Gas Turbine	$\eta_{II1} = \frac{W_g + E_r}{E_c - E_7}$	$Cf_1=((E_c-E_7)-(W_g+E_r))\cdot c_f$
2. Heat Recovery Boiler	$\eta_{II2} = \frac{E_a}{E_7 - E_{10}}$	$Cf_2=(E_7-(E_{10}-E_a))\cdot c_f$
3. Heat Exchanger + Stack	$\eta_{II3} = \frac{E_{sc}}{E_{10}}$	$Cf_3=(E_{10}-E_{sc})\cdot c_f$
4. Absorber + Auxiliary Boiler	$\eta_{II4} = \frac{E_f}{E_a + E_{caux}}$	$Cf_4=((E_a+E_{caux})-E_f)\cdot c_f$

TABLE 1 : Exergy Efficiency and Fuel Cost of Each Plant Component

COMPONENT	Cc (*10^3 $)
1. Gas Turbine	958
2. Heat Recovery Boiler	100
3. Heat Exchanger + Stack	20
4. Absorber + Auxiliary Boiler	184

TABLE 2 : Capital Cost of Plant Components

Summer Period

Table 4 shows fuel, service and capital costs in summer for each component. Viewing the table horizontally the total cost composition is shown in term of fuel, capital and service/ maintenance. Viewing the table vertically cost divisions for each component can be noted. This gives us an initial idea of improvement opportunities. In the same way information can be obtained on the behavior of single components as compared with the whole plant. Finally, data shown concerning hourly, useful exergetic and component exergetic costs supply all information required to quickly identify the behavior of the most defective component, thus showing where intervention optimization is needed in connection with higher costs: services, rather than energetic and/or capital improvements. Comparison with ISO conditions (table 3) supplies further information on plant design and performance.

With reference to the left hand side of table 4, which concerns absolute summer hour costs, it can be seen that inlet exergy costs predominate for the first three components (exergy related to fuel), while the highest figure for the absorber coincides with the invested capital cost. This suggests that the cogenerative plant has excessive losses when, actually, the predominating inlet exergetic cost is simply due to the higher production of effective utility.

Gas turbine service costs are the most noticeable. This is due to its portion of running costs over the whole plant. The absolute total turbine cost in conformity with ISO condition calculations dominates all the others (by 77%). However, its value should be reconsidered on the basis of production. Considering the plant exergetic costs per kWh, a decrease in gas turbine expenses (23.7%) can be noted, while those concerning the absorber are still very high (75%). Besides, the total exergetic cost is about 50% higher than expected (ISO conditions), mostly due to increase in the total cost of absorber useful

Element	Absolute Hour Cost							Hour Cost relative to Useful Global Exergy							Hour Cost relative to Useful Exergy of each Component				Exergetic Efficiency
	Cf		Cc		Cs		Ct	cf		cc		cs		ct	cf	cc	cs	ct	
	($/h)	(%)	($/h)	(%)	($/h)	(%)	($/h)	($/kWh)	(%)	($/kWh)	(%)	($/kWh)	(%)	($/h)	($/kWh)	($/kWh)	($/kWh)	($/h)	η(II)
1. Gas Turbine	48.34	44.2	31.56	28.8	5.51	5.0	85.41	0.030	14.6	0.020	9.5	0.003	1.7	0.053	0.023	0.015	0.003	0.040	0.2988
2. Heat Recovery Boiler	5.25	4.8	3.29	3.0	0.00	0.0	8.54	0.003	1.6	0.002	1.0	0.000	0.0	0.005	0.007	0.004	0.000	0.011	0.6334
3. Heat exchanger	4.45	4.1	0.33	0.3	0.00	0.0	4.78	0.003	1.3	0.000	0.1	0.000	0.0	0.003	0.062	0.005	0.000	0.067	0.2366
4. Absorber	4.64	4.2	6.06	5.5	0.00	0.0	10.70	0.062	30.4	0.082	39.8	0.000	0.0	0.144	0.062	0.082	0.000	0.144	0.2360
Total Cost							109.43							0.205					

TABLE 3 : Costs and Exergy Efficiency in *ISO* Conditions

Element	Absolute Hour Cost							Hour Cost relative to Useful Global Exergy							Hour Cost relative to Useful Exergy of each Component				Exergetic Efficiency
	Cf		Cc		Cs		Ct	cf		cc		cs		ct	cf	cc	cs	ct	
	($/h)	(%)	($/h)	(%)	($/h)	(%)	($/h)	($/kWh)	(%)	($/kWh)	(%)	($/kWh)	(%)	($/h)	($/kWh)	($/kWh)	($/kWh)	($/h)	η(II)
1. Gas Turbine	39.54	39.5	31.56	31.5	5.51	5.5	76.62	0.037	11.9	0.029	9.5	0.005	1.7	0.071	0.025	0.020	0.004	0.049	0.2674
2. Heat Recovery Boiler	5.25	5.2	3.29	3.3	0.00	0.0	8.54	0.005	1.6	0.003	1.0	0.000	0.0	0.008	0.010	0.006	0.000	0.016	0.5032
3. Heat exchanger	4.11	4.1	0.33	0.3	0.00	0.0	4.44	0.004	1.2	0.000	0.1	0.000	0.0	0.004	0.077	0.006	0.000	0.083	0.2004
4. Absorber	4.41	4.4	6.06	6.1	0.00	0.0	10.47	0.095	30.8	0.131	42.4	0.000	0.0	0.225	0.095	0.131	0.000	0.225	0.2030
Total Cost							100.06							0.308					

TABLE 4 : Costs and Exergy Efficiency in Summer Period

Element	Absolute Hour Cost							Hour Cost relative to Useful Global Exergy							Hour Cost relative to Useful Exergy of each Component				Exergetic Efficiency
	Cf		Cc		Cs		Ct	cf		cc		cs		ct	cf	cc	cs	ct	
	($/h)	(%)	($/h)	(%)	($/h)	(%)	($/h)	($/kWh)	(%)	($/kWh)	(%)	($/kWh)	(%)	($/h)	($/kWh)	($/kWh)	($/kWh)	($/h)	η(II)
1. Gas Turbine	41.50	39.9	31.56	30.4	5.51	5.3	78.58	0.038	9.9	0.029	7.5	0.005	1.3	0.072	0.026	0.020	0.004	0.050	0.2642
2. Heat Recovery Boiler	5.13	4.9	3.29	3.2	0.00	0.0	8.42	0.005	1.2	0.003	0.8	0.000	0.0	0.008	0.010	0.006	0.000	0.016	0.4704
3. Heat exchanger	4.29	4.1	0.33	0.3	0.00	0.0	4.62	0.004	1.0	0.000	0.1	0.000	0.0	0.004	0.076	0.006	0.000	0.082	0.2033
4. Absorber	6.26	6.0	6.06	5.8	0.00	0.0	12.32	0.153	39.7	0.148	38.4	0.000	0.0	0.301	0.153	0.148	0.000	0.301	0.164
Total Cost							103.94							0.385					

TABLE 5 : Costs and Exergy Efficiency in Winter Period

global exergy (+56%). So as to highlight the true component expenses, costs compared with exergetic production of each of them must be examined (last section on right hand side of table 4). As can be noted, the level of expenses per absorber is identical to the former one, while gas turbine expenses and those of the remaining components are reduced according to their use in the period under study.

Apart from the absorber, the inlet exergetic cost of last hot gas side exchanger is particularly high if compared with the useful exergy produced by the same component. It is also necessary to consider that most exergy degradation is due to hot gas delivered into the environment (L_{st} =65%). Their temperature cannot be reduced below a certain level for technical reasons. The L_{st} stack loss is greater than that following heat exchange. However, seeing that the former cannot be altered, action on the latter could be considered. Decline in energy under consideration, as in the previous case is mainly due to heat exchanges with a finite temperature difference between the two flows, whose value can only be limited by modifying the component substantially. Nevertheless, the value of exergetic efficiency of the exchanger unit + stack (L_{st}=19%) does not justify any action of this type, since this is not too far from design condition expectations (23.7%).

Analyzing the percentages of exergetic losses (in table 6), one can notice that the most serious one in the gas turbine is a combustion chamber loss. This is the cause of 41.6% of exergetic component reduction. This cannot be corrected. The other percentages are rather low. On the whole, on site gas turbine behavior (η_{GT}=26.7%) should

Losses	Summer		Winter		ISO	
	kW	comp.%	kW	comp.%	kW	comp.%
Lc1	127.6	3.52	142.8	3.94	127.3	2.75
Lc2	141.0	3.89	142.5	3.93	137.1	2.96
Lr1	23.2	0.64	10.2	0.28	20.4	0.44
Lr2	31.9	0.88	34.4	0.95	38.4	0.83
Lcc	1507.9	41.60	1555.3	42.90	1972.5	42.60
Le	164.9	4.55	170.0	4.69	140.8	3.04
Lgen	29.4	0.81	29.7	0.82	41.7	0.90
Leco	28.8	3.56	31.0	3.88	48.5	4.63
Leva	243.7	30.10	239.5	30.00	224.1	21.40
Lsc	74.4	27.90	75.6	28.00	72.7	24.00
Lst	138.9	52.10	139.6	51.70	158.8	52.40
Las	206.6	68.90	214.4	67.20	360.1	76.40
Lcaux	32.1	10.70	52.0	16.30	0.0	0.00

TABLE 6 : Exergetic Losses in Summer and Winter Period And in *ISO* Conditions

be considered satisfactory for this period.

In the case of the heat recovery boiler, the 30.1% of the evaporator section loss prevails over the 3.56% of the economizer (table 6). Exergetic loss in the first component is mainly due to a temperature jump between the boiler inlet hot gas and the steam it produces. This decline in energy could be reduced by using a two level pressure boiler. This choice cannot be economically justified, however, considering that the boiler exergetic performance (50.3%) is much higher than that of the remaining components: its cost share is

Element	Absolute Hour Cost							Hour Cost relative to Useful Global Exergy							Hour Cost relative to Useful Exergy of each Component				Exergetic Efficiency
	Cf		Cc		Cs		Ct	cf		cc		cs		ct	cf	cc	cs	ct	
	($/h)	(%)	($/h)	(%)	($/h)	(%)	($/h)	($/kWh)	(%)	($/kWh)	(%)	($/kWh)	(%)	($/h)	($/kWh)	($/kWh)	($/kWh)	($/h)	η(II)
1. Gas Turbine	40.19	39.5	31.56	31.1	5.51	5.4	77.26	0.037	9.8	0.029	7.7	0.005	1.3	0.071	0.026	0.020	0.004	0.049	0.2660
2. Heat Recovery Boiler	5.14	5.1	3.29	3.2	0.00	0.0	8.43	0.005	1.3	0.003	0.8	0.000	0.0	0.008	0.010	0.006	0.000	0.016	0.4935
3. Heat exchanger	4.18	4.1	0.33	0.3	0.00	0.0	4.51	0.004	1.0	0.000	0.1	0.000	0.0	0.004	0.076	0.006	0.000	0.082	0.2022
4. Absorber	5.36	5.3	6.06	6.0	0.00	0.0	11.42	0.139	36.7	0.157	41.4	0.000	0.0	0.296	0.139	0.157	0.000	0.296	0.1770
Total Cost							101.62							0.379					

TABLE 7 : Costs and Exergy Efficiency in Final Balance

only 8% of the global cost, of which the fuel quota is only 4.8%. These values are not really in agreement with ISO conditions, both in performance (-13%) and useful exergy cost (+60%). The last one was even highlighted, from energetic analysis where this component did not seem to have a very high heat recovery performance (De Lucia et al., 1997).

Component 4 (absorber + auxiliary boiler) in this period has an extremely low efficiency value: 20.3%. Its decrease is due to the exergetic loss in the absorber machine that represents the principal element, whose contribution is equal to 68.9%. That means that the absorber loss does not come counterbalanced by an adequate production of useful exergy E_f.

Even if its cost is relatively low, if compared with the winter value, it is due to the increase in summer demand for cold water in the factory (and thus an increase in absorber load), and to the lower use of the auxiliary boiler. This is confirmed by the low percentage of auxiliary boiler loss (10.7%). The cost of these two elements is high (around three times more than the gas turbine in terms of useful global exergy). This low key absorber behavior is confirmed by the slight reduction of capital cost of exergetic useful flow generated by the component.

To better understand the reason of this behavior we can see that both the absorber and heat exchanger work with low temperatures, so they have low useful exergy effects, because the Carnot coefficient $(1-T_0/T)$ penalizes them.

Thus, in the summer period, heat exchanger and stack (component 3) fuel cost is very high if compared with the useful exergy produced in the form of hot water.

Winter Period

The absolute cost of exergy introduced as fuel into the absorber in the winter period is very high. Its increase about 42% compared with the summer period (from 4.407 to 6.258 $/h - Table 5). It is due to the fact that in this period the HRSG does not produce enough steam for users and factory heating. Therefore with the previous assumption the additional fuel are charged to the absorber but in real case it is necessary for factory heating.

Low absorber use and high auxiliary boiler use are highlighted by the respective exergetic losses in the two periods considered and are shown in table 6. Main component loss percentage decreases from 68.9% (summer) to 67.2% (winter) ; in contrast with auxiliary boiler loss, which increases from 10.7% (summer) to 16.3% (winter).

Inlet gas turbine exergy hour cost decreases a little, mainly due to a greater production because the ambient temperature decreases. The exergetic cost of the last heat exchanger on the hot gas side is higher than the useful exergy produced by the component (table 5), but very close to the value seen in the summer period : about 0.076 $/kWh. All this means that the losses are almost unaffected by the seasons. On the other hand, the useful exergy unitary cost introduced into the absorber varies from 0.0949 $/kWh in the summer to 0.1528 $/kWh in the winter. In conclusion, the increase of about 61% in this cost highlights the seasonal use of this component, which, however, has very low exergy efficiency : in this period, it is 16.4% against 23.6% in nominal condition (ISO).

Losses	FINAL BALANCE	
	kW	(%)
Lc1	136.3	3.73
Lc2	144.0	3.94
Lr1	17.5	0.48
Lr2	32.9	0.90
Lcc	1538.6	42.10
Le	169.6	4.64
Lgen	29.6	0.81
Leco	29.6	3.71
Leva	237.7	29.80
Lsc	76.2	28.00
Lst	140.9	51.80
Las	194.3	66.00
Lcaux	48.0	16.30

TABLE 8 : Exergetic Losses in Final Balance

Balance Cost Results

The comparison between ISO conditions (table 3) and final balance (Table 7) thermoeconomic results is very important.

The absolute fuel costs for each component of the cogeneration plant are lower than those in nominal working conditions, while the absorbers behave in the opposite way. This is due, in the first case to the lower productive level which remains in real conditions with subsequent limitation of exergetic losses. Under nominal conditions the gas turbine load was taken to be 100%, while in real working conditions it varies from 90% during day time to 60% at night. The lower use of the absorber is in contrast with the increase in auxiliary boiler use and thus with the increased fuel demand, which is non existent under nominal conditions. As was to be expected, absolute capital and service costs turn out to be identical, since they are independent of plant behavior.

Comparison of cost per unit of useful exergy by the component highlights what has already been observed: all inlet exergy costs of components, in real working conditions are greater than those in ISO conditions. In particular, there is a considerable increase both in the cost of the last hot gas side exchanger and in that of the absorber. The first, under nominal conditions, has a cost of c_f= 0.0621 $/kWh against c_f= 0.0763 $/kWh under final balance (tables 3 and 7).

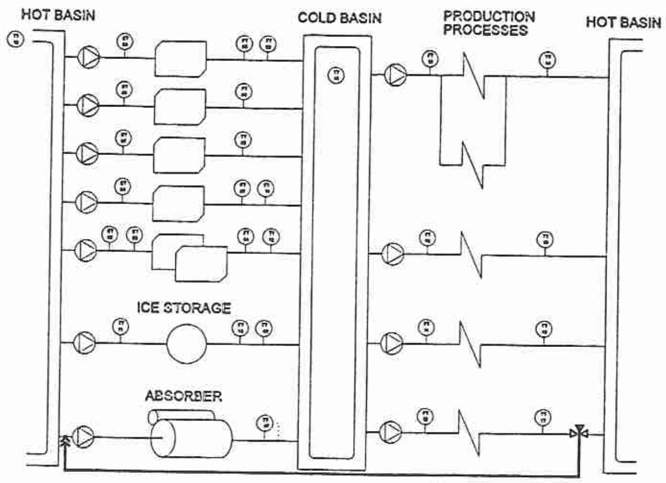

FIGURE 3 :Pseudo-Series Solution

The exergetic cost of the absorber increases of 123% (from 0.0624 \$/kWh under ISO conditions to 0.1391 \$/kWh under working conditions). Respect to ISO conditions, the reason for this increase can only be partly attributed to auxiliary boiler use, and thus to component fuel consumption. The main reason for this is the low production of useful exergy by the absorber in nominal working conditions which is unable to reduce the absolute cost of exergy introduced adequately. Further confirmation of this can be seen in the increase of 92.2% of unit capital cost, which passes from cc=0.0816 \$/kWh to cc=0.1569 \$/kWh, whose behavior does not depend on exergy losses but only on its operating condition.

POSSIBLE PLANT IMPROVEMENT

The result analysis shows that the components with a high unitary cost are the heat exchanger (component 3) and the absorber with the auxiliary boiler (component 4).

In all seasons the high cost of exergy introduced as compared with useful exergy produced for heat exchanger was noted. This component is greatly influenced by the exergetic loss due to hot gas discharge into the environment (L_{st}=65%). This cannot be reduced, unfortunately, for technical reasons. Exchanger loss cannot be reduced, except by modification of the same component; heat exchanger surfaces could be increased, for example, so as to decrease the temperature difference between the two flows; this is the main reason for the above mentioned exergetic loss. Doing this, however, would only lead to a very small increase in component exergetic performance, which is, in any case, perfectly acceptable considering the unavoidable loss at stack (36.2% in summer and winter, c. 38.9% in ISO conditions). In conclusion, seeing that the company requires production of a certain amount of hot water, the component can be considered to behave satisfactorily.

The same can be said for the absorber group, which has a very high cost of inlet fuel exergy as compared with useful exergy supplied by the component and also as low exergetic performance. Its value varies from 20.3% in the summer to 16.4% in the winter as compared with 23.7% under nominal conditions. By analyzing component exergy capital cost per unit it can be noted that it is unable to produce enough cold to reduce satisfactorily its extremely high capital cost (83%) as compared with the other components.

The reason for this behavior is to be sought in comparison between real absorber working conditions and design ideal ones. The main difference between these two conditions is linked to the temperature of the water arriving from the absorber inlet hot basin. This component was designed to cool inlet water from 12°C to 7°C. Actually the water on arrival already has a temperature of 8°C, and is thus cooled to 6°C. The operating area of the absorber thus falls below nominal temperature and its energetic efficiency is penalized.

In any case, on the whole, the behavior of the absorption cooling system can be considered reasonably satisfactory respect to the previous solution in view of the fact that the real COP for the traditional compressor units cooling station is c. 50% less than that expected (the mean COP of 2.5-3, during the fall and winter when the compressor cooling units operate at their best climatic conditions falling up to values of 1.9 in the worst condition in the summer).

Absorber Optimization

a) Series Absorber with Cooling Power Station

The first modification solution was based in increasing the absorber inlet water temperature by placing it in a series with the cooling power station. In this way, user return temperature varies between 8 and 10°C. It is lowered on entry into the hot basin because of direct contact with the cold basin water. Series modification of two groups implies the construction of a further basin with a temperature between the two previous ones. This would also have the task of limiting heat exchanges. Thus, the absorber would function at higher temperatures, increasing its efficiency. The water would then vary between 10°C and 6-7°C, then being lowered to 4°C by the cooling power station at the second cooling stage.

The proposed solution with present potentialities is not technically acceptable considering the incompatibility of the mass flow rates of the two groups. Actually, while the mass flow rate processed by the absorber remains at 79.1 kg/s, that supplied by the cooling power station is 105 kg/s. Therefore, only the mass flow rate required at the first stage could be allowed through, completing it after by drawing the missing portion from the hot basin. Even in this second case, absorber efficiency gains do not justify the high initial cost of plant modifications.

b) Pseudo-Series Absorber with Cooling Power Station

A more refined way of arriving at the previously noted incompatibility and limiting plant modifications is based on the selection of users with a higher feed back temperature. In this way the temperature in the component would increase and a middle basin would not be needed (figure 3).

Analyzing the mass flow rate of the select user feed water back, one can see that the mean temperature is about 9°C and the relative

Element	Absolute Hour Cost							Hour Cost relative to Useful Global Exergy							Hour Cost relative to Useful Exergy of each Component				Exergetic Efficiency
	Cf		Cc		Cs		Ct	cf		cc		cs		ct	cf	cc	cs	ct	
	($/h)	(%)	($/h)	(%)	($/h)	(%)	($/h)	($/kWh)	(%)	($/kWh)	(%)	($/kWh)	(%)	($/h)	($/kWh)	($/kWh)	($/kWh)	($/h)	η(II)
1. Gas Turbine	39.86	39.9	31.56	31.6	5.51	5.5	76.93	0.033	11.7	0.026	9.3	0.005	1.6	0.065	0.024	0.019	0.003	0.046	0.2674
2. Heat Recovery Boiler	5.30	5.3	3.29	3.3	0.00	0.0	8.60	0.004	1.6	0.003	1.0	0.000	0.0	0.007	0.010	0.006	0.000	0.016	0.5032
3. Heat exchanger	4.01	4.0	0.33	0.3	0.00	0.0	4.34	0.003	1.2	0.000	0.1	0.000	0.0	0.004	0.075	0.006	0.000	0.082	0.2004
4. Absorber	4.02	4.0	6.06	6.1	0.00	0.0	10.08	0.084	29.3	0.126	44.2	0.000	0.0	0.209	0.084	0.126	0.000	0.209	0.2190
Total Cost							*99.95*							*0.285*					

TABLE 9 : Costs and Exergy Efficiency in Inlet Air Cooling Solution

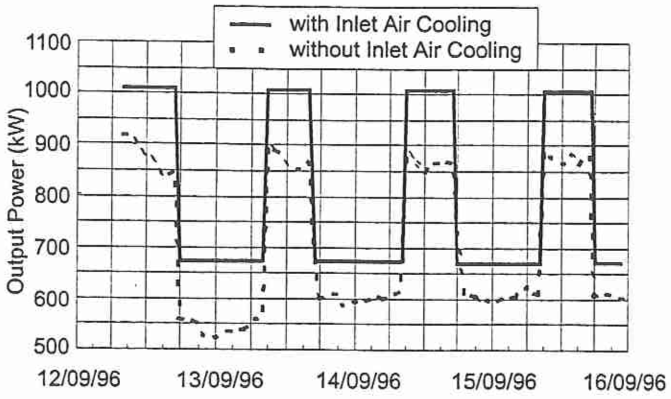

FIGURE 4 : Gas Turbine Electric Power Output

mass flow rate is almost constant (38.6 kg/s). This means that the absorber inlet becomes even colder, given that the remaining mass flow rate is taken from hot water at lower temperature. Due to the easy plant modification this solution has been realized but the results are not too satisfactory.

Inlet Air Cooling

To increase the absorber load, cold water can be used to cool the inlet air at the gas turbine compressor; up to 10°C. In this case, a greater absorber load, with a more appropriate temperature, and an increase in electric power are obtained (De Lucia et al., 1994). The power plant would undergo few modifications: only a heat exchanger at the gas turbine inlet being necessary.

Inlet air cooling at the compressor can be carried out mainly in the spring and summer, when the on site ambient temperature exceeds 15°C. In the other seasons, no improvement is expected.

Simulation Results

COMPONENT	c_t present ($/kWh)	c_t modification ($/kWh)	Reduction (%)
1. Gas Turbine	0.0488	0.0459	6.0
2. Heat Recovery Boiler	0.0159	0.0158	0.5
3. Heat Exchanger	0.0833	0.0815	2.2
4. Absorber	0.2254	0.2095	7.1

TABLE 11 : Total Unitary Cost and Relative Reduction Percentage

Figure 4 shows the output power trend in a second week of September with and without Inlet Air Cooling (IAC).

In table 10, one can observe that, in interrefrigeration, there is a clear loss reduction at the first exchanger used for the production of superheated water. This is justified by lower inlet air temperatures. There are no drastic variations in exergy degradation in the combustion chamber, economizer losses and heat recovery boiler evaporator. Furthermore, a clear improvement can be seen in expansion, owing to the gas turbine power increase. The exergy loss in the last exchanger on the hot gas side decreases thanks to a considerable gas temperature reduction in this part of the plant. The exergy loss at the stack is generally greater owing to the greater hot gas mass flow rate. Energy degradation of the absorber increases slightly. Finally there is reduction in the auxiliary heat loss, caused by lower component use. The boiler only operates during the evening when the gas turbine is operating at 60%

Losses	Inlet Air Cooling	
	kW	(%)
Lc1	133.4	3.68
Lc2	138.8	3.83
Lr1	10.5	0.29
Lr2	35.2	0.97
Lcc	1486.1	41.00
Le	144.6	3.99
Lgen	32.3	0.89
Leco	30.2	3.73
Leva	242.1	29.90
Lsc	66.9	25.10
Lst	145.3	54.50
Las	219.8	73.30
Lcaux	14.6	4.86

TABLE 10 : Exergetic Losses in Inlet Air Cooling Solution

Thermoeconomic Results

Power plant simulation with this new solution only takes place in the summer period, because this is the only season in which inlet air cooling at the compressor can be conveniently used.

In the first column in table 9, an inlet exergy cost increase compared with the already existing one con be noted both for gas turbine and recovery boiler. This is due to high production of two components, with internal loss increases. The last hot gas side exchanger together with the stack costs are less, owing to the substantial reduction in mean exergetic loss at the exchanger, which compensates for the increase in that at the stack. There is an even greater reduction in the exergetic cost entering the absorber which, contrary to initial expectations, increases exergetic production. Its loss percentage increased from 69.9% to 73.3% (table 10). At the same time auxiliary boiler use decreased with subsequent cost decrease. The boiler percentage loss fell from the 10.7% of the present situation to 4.86% of that with cooling. The capital costs, together with the absolute servicing costs of the various components did not change, seeing that they are independent of factory requirements.

So as to be able to be sure that production has increased in the whole plant, useful exergy unit costs produced by the component (central zone of table 9) must be compared. This comparison enables one to see how all the above mentioned cost items have been reduced. In particular, the absorber cost of inlet exergy as fuel passes from 0.0949 $/kWh to 0.083 $/kWh (a c. 12% decrease). The capital cost per unit of useful exergy produced by the component falls by 3.5%, while that of the gas turbine falls 6.4%.

Comparison of exergetic performances shows an improvement over the whole plant. This is especially the case with absorber efficiency, which passes from 20.3% to 21.9%, with an increase of 7.5%.

Table 11 shows percentages in total cost reduction for useful exergy produced by each component.

CONCLUSIONS

This method has been shown to be very useful in the analysis of an power plant producing both electric power and heat at different temperatures.

Using the exergy concept, power plant irreversibilities and their related costs can be easily determined in real operating conditions comparing them with nominal conditions (ISO conditions). Thermoeconomic analysis permits us to determine the components which have the highest costs (low efficiency) and which are farther from ISO conditions. This method is also useful in the design of optimization phases and in maintenance program decisions. By checking losses (tables 6, 8, 10) and simply looking at the related costs it is very easy to determine when and where to intervene.

The application of thermoeconomic analysis allows comparison from some alternative power plant solutions directly from the economic point of view.

A thermoeconomic analysis of a pharmaceutical factory has been carried out. This analysis determined good power plant performance, only the absorber having low efficiency; this is mainly due to a too low temperature of inlet cold water into absorber. To increase absorber efficiency, three solutions are analyzed: absorber in series with a central cooling group, absorber in pseudo-series with a central cooling group and inlet air cooling of compressor.

On the whole, the behavior of the gas turbine can be considered satisfactory in view of the fact that the small size machine and the working conditions behaved excellently even with partial loads.

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REFERENCES

De Lucia M., Bronconi R., and Carnevale E., 1994, "Performance and Economic Enhancement of Cogeneration Gas Turbines through Compressor Inlet Air Cooling," *ASME J. of Engineering for Gas Turbine and Power*, Vol. 116, pp.360-365.

De Lucia M., Lanfranchi C., 1994 ; "Experimental Testing and Thermoeconomic Analysis of an Icineration Plant Postcombustor With Oxycombution", *ASME J. of Energy Resources Technology*, Vol. 116, March 1994, pp.72-78.

De Lucia M., Lanfranchi C., and Boggio V., 1996, "Benefits of Compressor Inlet Air Cooling for Gas Turbine Cogeneration Plants," *ASME J. of Engineering for Gas Turbine and Power*, Vol. 118, pp.598-603.

De Lucia, M., Lanfranchi C., and Matucci, A., 1997, "A Small Gas Turbine Plant for Cogeneration of Electricity, Thermal and Cooling Thermal Energy with an Absorption Unit", *IGTI Turbo Expo '97*, Orlando, June 1997, 97-GT-.

De Lucia M., Manfrida G., 1990 ; "Breakdown of the Exergy Balance in a Glass Furnace", *ASME J. of Energy Resources Technology*, Vol. 112, June 1990, pp.124-129.

Hill J.P., 1992, "Assessing the Economics of Industrial Gas Turbines Cogeneration Applications," ASME Paper 92-GT-333.

Manfrida, G., Stecco, S.S., 1992 ; "Exergy-Loss Modeling of Power Plants and Industrial Processes," *17th IECEC*, Los Angeles, Calif., IEEE Paper 829004.