A SMALL GAS TURBINE PLANT FOR COGENERATION OF ELECTRICITY, THERMAL AND COOLING THERMAL ENERGY WITH AN ABSORPTION UNIT

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ABSTRACT

A cogeneration plant with a small gas turbine was installed in a pharmaceutical factory and instrumented for acquiring all the values necessary to appraise both its energetic and cost advantages. The plant was designed and built as a demonstrative project under a program for energy use improvement in industry, partially financed by the European Union.

The system comprises as its main components: 1) a gas turbine cogeneration plant for production of power and thermal energy under the form of hot water, superheated water, and steam; 2) a two-stage absorption unit, fueled by the steam produced in the cogeneration plant, for production of cooling thermal energy.

The plant was provided with an automatized control system for the acquisition of plant operating parameters. The large amount of data thus provided made it possible to compare the new plant, under actual operating conditions, with the previously existing cooling power station with compression units, and with a traditional power plant.

This comparative analysis was based on measurements of the plant operating parameters over nine months, and made it possible to compare actual plant performance with that expected and ISO values. The analysis results reveal that gas turbine performance is greatly affected by part-load as well as ambient temperature conditions. Two-stage absorber performance, moreover, turned out to decrease sharply and more than expected in off-design operating conditions.

NOMENCLATURE

W_e	= Electric output [kW]
m_c	= Fuel mass flow rate [kg/h]
M_{caux}	= Fuel consumption (auxiliary boiler) [kg/h]
HR	= Gas turbine Hear Rate [$kJ_{themal}/kJ_{electric}$]
Q_F	= Cooling output form absorption [kW]
COP	= Coefficient of Performance
Q_{SC}	= Output from heat exchanger SC [kW]
T	= Temperature [°C] (ref. points in figure 5)
η_{GT}	= Gas turbine electric efficiency
η_{CHP}	= Cogeneration efficiency

PLANT DESCRIPTION

The plant is located in a pharmaceutical factory near Frosinone in Central Italy. It was initially composed of five air compression cooling units (for a total installed power of 2200 kW) and two boilers. With the aim of reducing energy consumption, in 1994 a new cogeneration system was designed and built.

A gas turbine cogeneration plant for production of power and thermal energy (superheated and hot water, and steam) 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.
The plan and the measuring point of the gas turbine plant is shown in Figure 2.

Interrefrigeration is carried out by two heat exchangers. The first one supplies superheated water at 120-130 °C, which is especially important since, in this particular case, it substituted the heat supplied by the electrical resistors installed on the mounters. The remaining thermal energy, being at a temperature too low for use in the production processes, is dissipated in the second heat exchanger.

The HRSG on the exhaust gases has a single pressure level and is of vertical type one with water tubes. The boiler has a by-pass duct on the gas side which allows both exclusion of the boiler in case no steam is needed for the process and in case of HRSG failures. 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.

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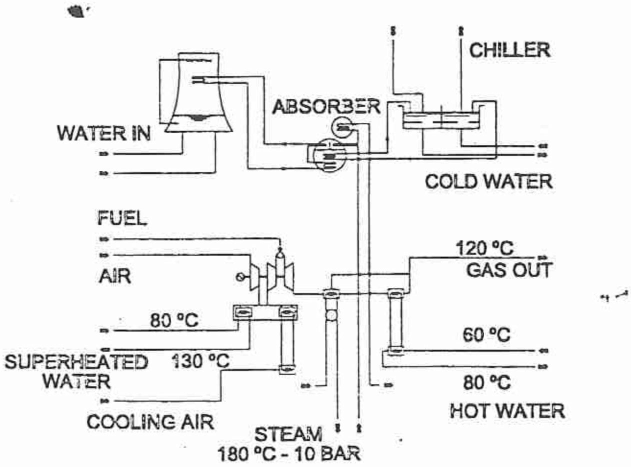

FIGURE 1 : Plant Layout

Steam produced in the recovery boiler is used both for production 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 for production of cooled water down to 5-7°C is a two-stage machine, using steam at 8 bar and 180°C as high temperature heat source. The absorber is equipped with a cooling evaporative tower for dissipation of condenser heat (Figure 3) (Ikari, Masaroni, 1985a, 1985b).

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 (Figure 4).

The traditional cooling power station was composed by five

The plant was started up in June 1994, and data collected constantly by the data acquisition and monitoring systems described below is available from that date on. 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. The plant was shut down for three weeks during the month of August 1994, because the entire factory was closed for holidays. The gas turbine was briefly shut down at the end of March 1995, because some cracks in the welding were found inside.

DATA ACQUISITION SYSTEM

The purpose of this experimental phase (presented both here and in a subsequent thermoeconomic evaluation (De Lucia et al., 1997) is to analyze the performance of the integrated system proposed for this project in comparison with a traditional system for producing thermal

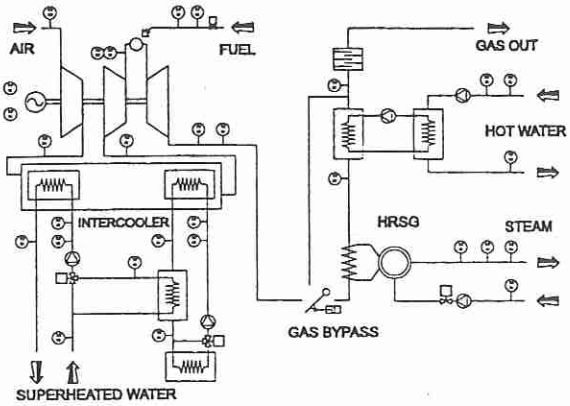

FIGURE 2 : Gas Turbine And HRSG

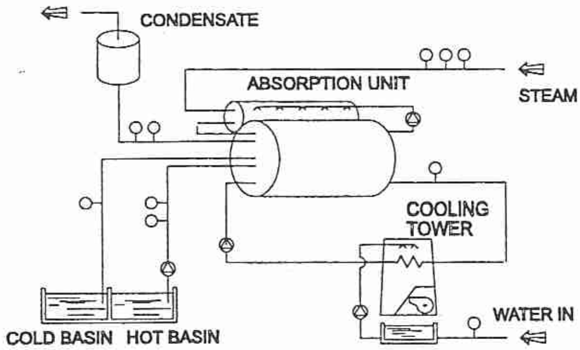

FIGURE 3 : Absorption Unit

and cooling energy. Particular attention was devoted to the cooling system. The monitoring phase was therefore carried out so as to highlight the different performance of the cooling power station (absorption and compression) and to compare the performance of the two plants and of the cooling-cogeneration system from both an energy and an economic point of view.

The system is made up of a Data Acquisition System (DAS) which has 256 kb RAM for storing the relative data, and a management program on EPROM. It is connected to a computer by an RS485 line. Another RS232 port is available for diagnostic functions.

More than 80 analog type and counting type signals were each recorded to characterize the system from an energy point of view. The whole plant was divided into three main sections:

- Gas turbine and heat recovery system (Figure 2);
- Absorption unit (Figure 3);
- Cooling power station with compression units (Figure 4).

During the first eight months of operation, various tests were carried out in order to obtain data on the plant's behavior and its capacity to adapt to different operating conditions. In particular, data

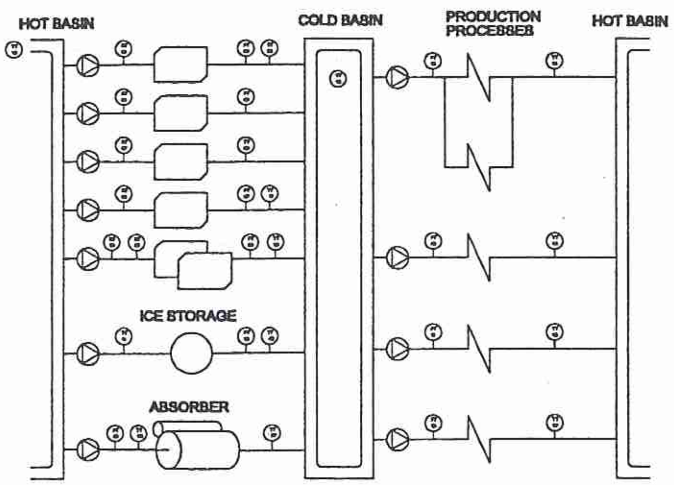

FIGURE 4 : Cooling Energy System

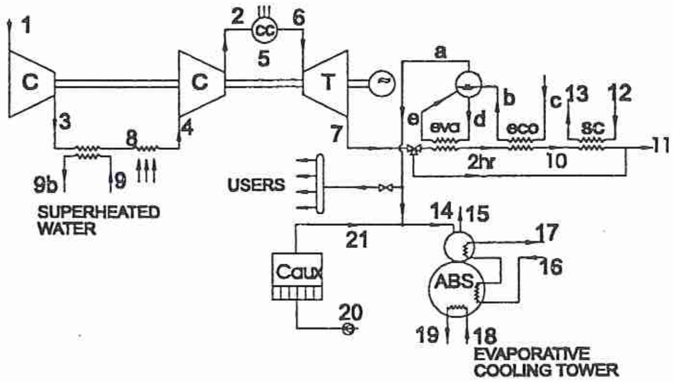

FIGURE 5 : Technical Scheme For Plant Simulation

were sought on plant performance when the turbine is constantly running at full load and when the load is reduced to 60% at night and over the weekend, in order to adapt to the productive needs of the company.

DATA ELABORATION

To limit elaboration time and allow costs and energy 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. For each week, hourly average values were calculated, from the 5-minute spaced values recorded by the DAS.

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 5, some calculations were curried out to evaluate the data not recorded by DAS but necessary in energy evaluation. For this purpose the

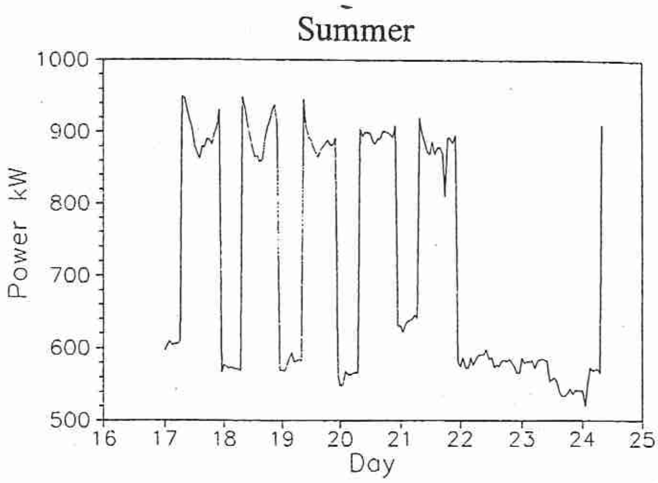

FIGURE 6 : Turbine Power Output

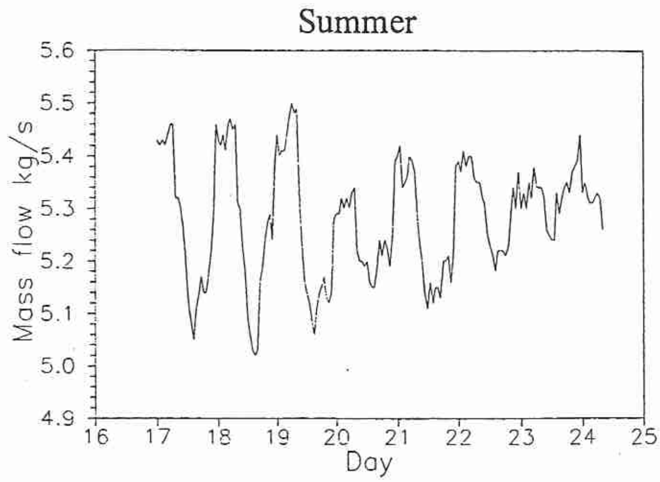

FIGURE 7 : Turbine Exhaust Gas Flow Rate

equations and curves supplied by the manufacturer of the gas turbine were used.

Apart form the gas turbine compressor efficiency (assumed to be the same for the two stages) and the minimum ΔT at the interefrigerator, the following parameters were established:

- pressure drops at the gas turbine inlet and outlet (80 e 150 mmH_2O respectively)
- relative air humidity (60%)
- combustion chamber pressure drop (3%)
- efficiency of heat exchanger SC (91%)

EXPERIMENTAL RESULTS

The graphs for the two weeks described are presented below in detail:

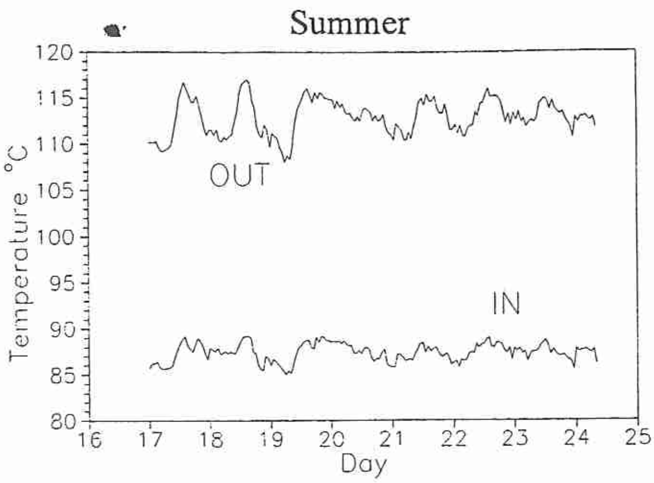

FIGURE 8 : Superheated Water Temperatures

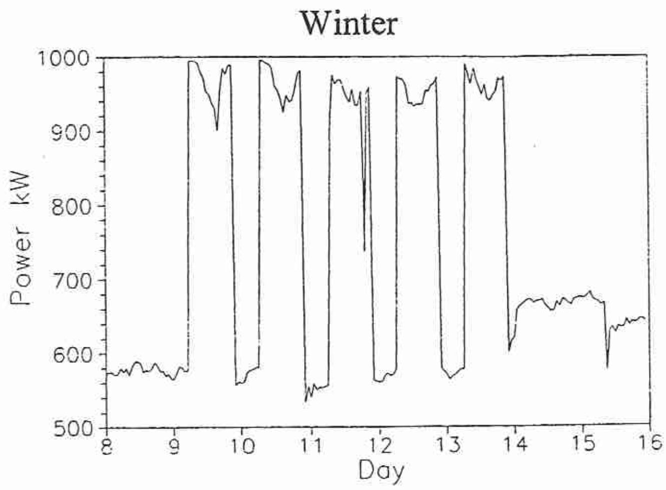

FIGURE 10 : Turbine Power Output

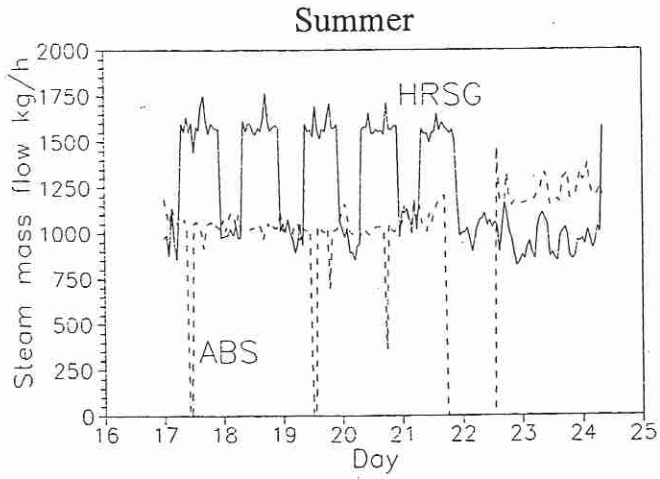

FIGURE 9 : Absorber And HRSG Steam Flow Rate

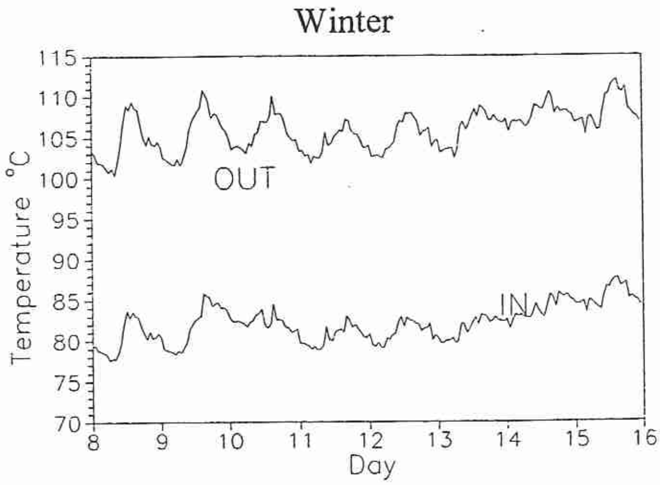

FIGURE 11 : Superheated Water Temperatures

Summer Period

During the summer week selected the plant operated at almost full load during day time and at about 60% at night and during the weekend, thus penalizing efficiency and thermal energy recovery.

Figure 6 shows a fall in electrical production during the middle hours of the day caused by air temperature increase (De Lucia et al., 1994, 1996, Ebeling et al, 1992, 1994).

This behavior is due to the reduction of air density that can also be seen in figure 7 showing the turbine exhaust mass flow rate.

In figure 8 it can be seen that the interrefrigeration water temperature inlet remains practically constant, while the exit temperature varies slightly.

The HRSG produces steam up to 1700 kg/h during day time, going down to 1000 kg/h at night (figure 9).

Auxiliary steam generation operation can be worked out by comparing the steam flow rate from HRSG with that required by the absorption unit. If the former is in excess, some of the steam produced is conveyed to internal factory uses; the auxiliary generator thus remains inactive. In other circumstances, the latter adds to the steam produced by the HRSG.

Winter Period

For reasons of space the winter period graphs will not be commented on. They can be seen in figures 10,11,12 for purposes of comparison. Only the most obvious differences will be commented on.

A fall in all mean temperatures is obviously expected. Almost all the temperatures related to the turbine are on average 10°C lower, while HRSG and others are more or less constant.

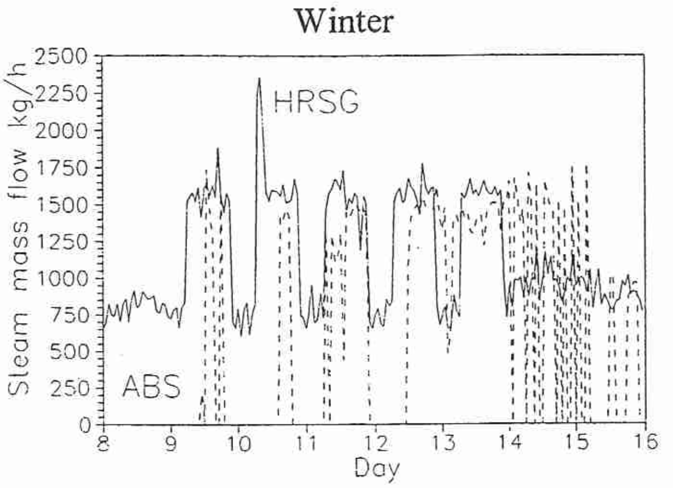

FIGURE 12 : Absorber And HRSG Steam Flow Rate

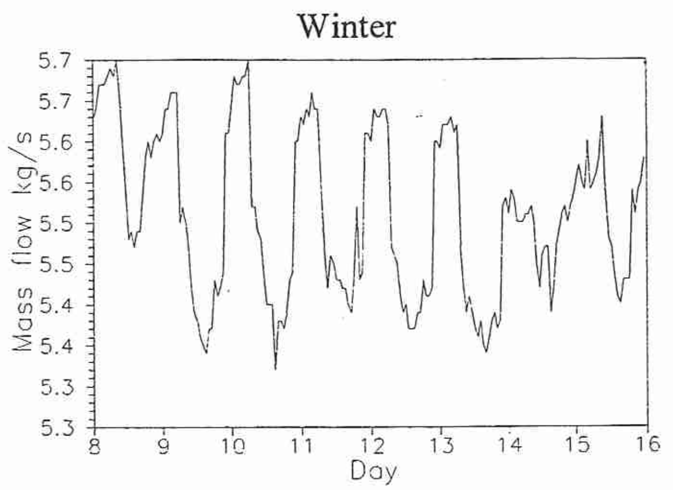

FIGURE 13 : Turbine Exhaust Gas Mass Flow Rate

It is the mean air temperature drop that justifies the increase in turbine power up to 1000 kW (figure 10).

PROBLEMS AND SOLUTIONS

Initially, there were great difficulties in defining the absorber unit cooling system. For each cooling kW produced, about 1.86 kW must be dissipate. The maximum inlet working temperature of the absorber group on the condenser side is 34°C. It would be very difficult in a Mediterranean country to imagine cooling such a system with air. It is therefore necessary to consider a cooling system which uses the latent heat of water evaporation to prevent it from reaching a crisis level during the summer months (during which, however, there is a greater need for cooling energy).

The first problem is the amount of available water. Traditional solutions for cooling condensed water call for the use of a cooling tower. This solution requires constant use of the cooling tower, even when air condensing would be sufficient (for example, during the night).

The idea of building an air cooling system was only considered in cases of real need (test > 26°C). This solution would have to use special batteries (made of zinc-plated steel with a distance between the fins of at least 6 mm) which would make it easy to periodically clean off the encrustation caused by the spraying of water. Such a choice would mean a greater cost and therefore would not be a practical alternative. A similar solution would be to use both the cooling tower and an air cooling system of standard production with batteries with copper or aluminum fins, but this makes the system much more complicated. Finally, the idea of using a cooling tower with finned tubes was taken into account. However, in this case, too, the ratio of costs to benefits forced us to fall back on the traditional solution. Consumption could, in fact, be reduced noticeably only during the winter, when there are no particular problems of water availability (this could, however, be an interesting idea for a different climate than the one where the plant is situated).

Because it was impossible to reduce the consumption of water by using a different cooling system, the possibility was considered of recovering some of the waste water inside the factory, and of reducing the discharge of water from the cooling tower as much as possible. Recovering the thawed water from the freeze dryer and the waste water from the phial washers, could considerably reduce the water taken from the well. After being treated, mixed, and collected, water is used for cooling. To accomplish this, a tank was built at the base of the tower. A system for recirculating the water from the cooling tower was also studied. Discharge is regulated by a system which measures its conductivity and further reduces water consumption.

Using these solutions, it is possible not only to rationalize the use of water, but also to noticeably reduce energy consumption.

The energy needed to work the fans in the cooling tower is not negligible. Initially the idea was to use double motorization and dissipative regulation of the air flow with moving dampers. Later on, it was decided to use motorization controlled by an inverter. This saves further energy, with only a slight cost increase with respect to the double motorization solution. In this case, it was necessary to install some servo-fans to avoid overheating the low-speed motors.

The HRSG do not appear to be as good as the other parts. Exhaust temperature T_{10} is still too high at about 200°C. The problem was partially solved by installing an auxiliary heat exchanger at the HRSG exit producing water for sanitary and ambient heating.

A last observation must be made about the use of water & glycol in the circuit (this use is fairly frequent in cooled water circuits, either to avoid possible freezing of the circuit during the winter, or for low working temperatures, or because of the necessity of using the circuit to fill ice storage units). By using this system, the output of the absorber unit is slightly penalized. However, the effect is more substantial for the overall output machine. In order to produce the same amount of cooling energy, it would be necessary to equip the absorber units with exchangers that were larger and cost about 20% more.

It was found that the absorber, planned to operate at between 7 (exit) and 12°C (input), due to a problem in the plant piping, was

	SUMMER				WINTER			
	max	min	mean		max	min	mean	
			day	night			day	night
We	920	520	750	580	990	540	802	593
m_c	297	229	267	230	306	213	270	225
m_{caux}	18.3	0	1.5	0.01	53	0	3.6	9.9
m_{wa}	1246	750	1070	1100	1738	661	1362	1412
η_{GT}	23.3	18.7	21.5	19.4	24.3	19.1	22	19.7
HR	5.353	4.294	4.51	5.15	5.24	4.11	5.44	5.08
Q_F	697	419	572	543	821	64	482	680
COP	1.145	0.731	.797	0.757	.958	0.11	0.54	0.80
Q_{sc}	383	331	356	369	392	362	377	373
T_3	168	150	159	155	146	131	141	137
T_{9b}	115	113	114	114	119	102	107	104
T_4	62	50	56	51	52.2	44	47.6	46.4
T_2	225	209	217	214	215	201	206	204
T_7	454	347	405	348	440	322	396	333
T_{10}	202	177	191	179	200	187	196	192
T_{11}	149	103	123	105	130	118	124	121
T_{16}	10.1	6.2	7.67	8.39	9.5	6.3	8.5	8.5
T_{17}	7.5	4.7	5.92	5.15	7.3	4.9	6.5	6.5
T_a	175	172	174	173	175	173	173	173
T_c	103	98	101	100	101	96	98	97
T_{13}	82	80	80.5	81	83	81	81.3	81.7
η_{CHP}	64.82 %				61.02%			

TABLE 1 : Summer and Winter Main Data

only operating at the lower temperature of 7 to 5°C. Thus the COP was heavily penalized. Only on one occasion were best working conditions recorded, for one hour the COP reached the nominal value (1.14). During the rest of the period under study mean values were about 0.8 (c. 30% less then expected). So the absorber is shown to be an extremely sensitive component and does not allow high flexibility in regulation in terms of working temperature. This means that particular care must be taken at the planing stage when the data have been established and the machine has been ordered. The absorber must be made to work as close as possible to temperature design conditions in order to avoid considerable reduction of output. This means that the conditions of steam, cooled water, and condensation water supply must be exactly defined before ordering it. This is not straightforward, given that this was a pilot project in which all of the inter-relationships between the various parts had to be defined.

PERFORMANCE EVALUATION

The technology which was used could be applied to pharmaceutical or foodstuff companies, or to any civil or industrial use which requires electrical, thermal, and/or cooling energy simultaneously.

This project was also intended to include a carefully documented batch of experiments which would furnish reference data for any future use of the technology. The experimental phase, which provided a large amount of data during its 5000 hours of operation (in table 1 the main data are reported), made it possible to confront the new system, under actual operating conditions, with the previously-existing cooling power station and its compression units, and with a traditional power plant.

On the basis of annual consumption, evaluation shows that the system developed is able to furnish an energy recovery in line with the prudent predicted results. The level of energy saved, on an annual basis, was 363.3 TEP/Y. However, the operation of the system is substantially influenced by the management of the cooling units.

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. For more details of performance analysis see (De Lucia et al., 1997, Hill J.P., 1992) which presents a thermoeconomic evaluation.

From an economic point of view, the length of time needed for a return on an investment was calculated in 5 or 6 years, on the basis of an estimated plant cost of 2,400 million Italian Lira (c. 1,600,000 US$). It must be said that the earnings from this project were negatively influenced by the considerable increase in natural gas prices in the last few years in Italy. During the same time, the price of electrical energy, which is the principal output of the proposed plant, has had a much smaller percentage growth.

Compressor cooling power station performance under real working conditions can be summarized as follow:

During the hottest part of summer days, when cooling requirements are higher, even if the compressors are working at full speed the cooling power output declines. These units are not able to supply more than 60-70% of the design output. In this period the maximum electric consumption is recorded and the COP decreases markedly (25-30%) up to values of 1.9 in the worst condition.

During the fall and winter the compressor cooling units operate at their best climatic conditions and the mean COP increases by up to c. 2.5-3.

On the whole, the behavior of the absorption cooling system can be considered reasonably satisfactory in view of the fact that the real COP for compressor units is c. 50% less than that expected.

CONCLUSIONS

The evaluations of the annual consumption make it possible to conclude that the plant described here is able to furnish a level of energy recovery along the predicted lines.

However, the operation is heavily influenced by the management of the cooling power unit, and by the sensitivity of the absorber to variations in the inlet and exit coolant temperatures and by the steam conditions, which markedly reduce its performance level. Solving these problems, which is already being researched for this particular project, will allow future plants to operate at levels closer to design ones.

From an economic point of view, the time needed for a return on an investment, which is estimated at 5 or 6 years, is heavily influenced by the substantial price increases in natural gas which have occurred in Italy in the last few years.

The best location for a plant of this type is in a factory which has a great need for thermal energy at high temperatures, such as steam, and where the energy requirements of the factory are clearly focused on thermal needs as opposed to electrical ones. Furthermore, in any industrial application, the estimated energy recovery is always far more than the actual, obtainable one because of:

- lower thermal energy needs during the summer;
- low coincidence of steam consumption.

Such problems are usually more evident for plants with low ratings and are such that they would exclude the use of such a cogeneration typology in pharmaceutical companies.

The project that was carried out aimed at solving these basic problems by a differentiated use of the heat coming out of a cogeneration plant. At the end of the project, it can definitely be affirmed that the use of steam produced by the thermal recovery of the exhaust gases of the turbine, both directly and as energy for the production of coolant fluids, allows the system great flexibility in operation, increasing the coincidence factor of the thermal requirements. The real capacity of the system to balance the heat recovery requirements of the cogeneration plant both in winter (when there is a great need for heating) and in summer (when there is a greater need for cooling) has been pointed out.

Finally, absorber units are extremely rigid machines, in term of temperature, and do not allow high flexibility in regulation. Once the design data have been established, the absorber must be made to work as close as possible to temperature design conditions, to avoid substantial penalties in its output. This means that the conditions of the of steam, cooled water, and condensation water supply must be exactly defined before ordering it.

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