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A COGENERATION PLANT BASED ON A STEAM INJECTION GAS TURBINE WITH RECOVERY OF THE WATER INJECTED: DESIGN CRITERIA AND INITIAL OPERATING EXPERIENCE

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ABSTRACT

The idea of re-injecting into a gas turbine cycle the steam generated by the heat recovery steam generator (HRSG) is a well-established practice, especially in small-medium size cogeneration plants operating under variable heat demand. Power augmentation, electrical efficiency increase, NO_x reduction and operating flexibility are the most obvious advantages brought about by steam injection. On the other hand, the discharge to the ambient of the injected steam has two major drawbacks: (i) a relevant water consumption and (ii) the large thermal loss related to the latent heat of steam. The addition of a recuperator downstream of the HRSG, whereby steam condensation takes place, can solve both problems, by achieving very high first-law efficiencies (over 100%, if reference is made to the lower heating value) and the integral recovery of water. The present paper describes the design philosophy and the operational experience of a cogeneration plant where such a condensation is accomplished. To the Authors's knowledge, it is the first time in the world that this is achieved with gas turbine exhausts. The plant is located inside the "CARROZZERIA BERTONE", a car manufacturing factory near Turin, Italy. It was designed to fulfill all the energy needs of the factory: it supplies all the electricity, steam and hot water required by the industrial process and during peaking hours, sells excess electricity to the national grid, at special increased tariffs offered to energy-saving plants in Italy. The plant erection (including the recuperator/ condenser) was completed in December 1992; commercial operation began in February 1993.

INTRODUCTION

Advantages and disadvantages of small gas turbines in cogeneration applications

Gas turbines are, in many ways, suitable for small/medium-sized cogeneration applications: they are highly reliable, easy to run and, if opportune measures (like water or steam injection, dry-low NO_x burners and/or SCRs) are taken, capable of satisfying the strictest limits of environmental regulations. Furthermore, owing to the high temperature of the exhaust gases they are able to produce "precious" heat with characteristics that are compatible with the heat vector distribution network (steam, superheated water, diathermic oil) traditionally used in industrial plants. On the other hand, they have a relatively limited electric conversion efficiency both in terms of nominal as well as, above all, reduced loads. Added to these limitations (in terms of thermodynamic performance) there is, for non-direct heat use[1] applications, also the disadvantage of a first-law efficiency which is several points lower than the values that can be obtained with a pure back-pressure steam cogeneration system or with reciprocating "total energy" gas operated engines[2]. Unless special construction features, like variable-geometry turbomachinery (rarely used in small-medium sized machines) are adopted, gas turbines are basically cogeneration systems with "one degree of freedom", where the thermal power cogenerated is strictly linked to the electrical power: this is a serious drawback both in terms of energy consumption and saving, the reason being that where most cogeneration applications are in question, both in the industrial as well as services sector, the heat demand is subjected to extremely variable temporal trends which are not necessarily correlated to the electricity demand trend.

The two most obvious alternatives aimed at preventing excess heat dissipation (when the heat demand is lower than that generated by the turbine): (i) heat accumulation and (ii) the installation of a supplementary burner, are both unsatisfactory. Although advanced proposals are nowadays being considered (Somasundaram and al., 1993), heat storage is economically feasible only in a few, favorable circumstances with short storage time while the introduction of supplementary firing leads to an

[1] Such as drying processes, whereby the heat of exhaust gases can be integrally recovered.

[2] Due to turbine inlet temperature limitations, gas turbines operate with great excesses of air. Hence, for the same amount of fuel and the same exhaust gas temperature, the heat released to the ambient by the exhaust gases of a gas turbine are typically 3-4 times larger than that of a boiler operating close to stoichiometric conditions.

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"under-sizing" of the gas turbine potential for a given maximum heat demand, in a power field where economy of scale plays an important role. Under these circumstances, the small cogeneration gas turbine has serious limitations in terms of achieving significant primary energy savings for most application typologies: as shown in fig.1, modern "energy saving" legislations require rather demanding year-round performance to a cogeneration plant. With reference to the Italian situation, a cogeneration plant generating electricity (E) and heat (H) on a yearly basis by consuming a fuel energy (F - based on LHV) should have a PER (Primary Energy Ratio) larger than unity. PER is defined as:

(primary energy required to generate E and H by "state-of-the art" plant separately) / (primary energy used by the cogeneration plant) = $(E/0.51 + H/0.90)/F = \eta_{el}/0.51 + \eta_{th}/0.90$

In other words, a cogeneration plant should use less fuel than the combination of a "state-of-the-art" electric plant having a net average electric efficiency as high as 51% and of a "state-of-the-art", 90% efficient boiler generating the same amount of electricity and heat. As shown in fig.1, it is difficult to fulfill this requirement with a simple cycle gas turbine, unless the plant operates most of the time at full load with full heat recovery.

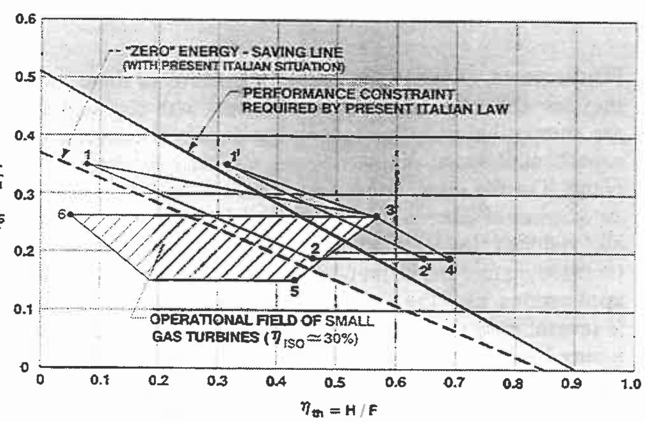

Fig. 1 *Operating points of small cogeneration gas turbines in the electric efficiency-thermal efficiency plane. Variations in the heat output capacity from nominal value (point 3) can be obtained by means of a by-pass vent (line 3-6) or by regulating the electrical output (line 3-5): both adjustments involve a considerable reduction of the yearly average PER. The operative lines of the steam injection solution, with (line 1'-2') and without (line 1-2) heat condensation recovery are shown. The significance of the points indicated in the figure will be explained further on in this paper (see fig. 5).*

Fully steam-injected plants

In larger plants (with a capacity of over 10/15 MW_{el}) most of the aforementioned problems are easily resolved by the use of a combined gas/steam cycle with an extraction/condensing steam turbine. The thermodynamic performance offered by this typology are excellent, resulting in the saving of considerable quantities of primary energy regardless of the variations in the heat demand (a modern combined cycle, even when operating under pure condensation, uses natural gas much more efficiently than most current thermoelectric power plants using the same fuel). For smaller plants, the combined-cycle option is not very interesting. This is due not only to the plant engineering and conduction problems that it involves but also to the limited efficiency characteristic of small steam turbines. On the contrary, the full steam injection solution[3] is more interesting. Compared to simple cycle solutions, this offers considerable advantages both in terms of electric efficiency (fig. 2) as well as specific cost (fig. 3). Furthermore, the massive injection of steam introduced into the combustor permits the attainment of extremely good emission levels without having to use special (and costly) "Dry-Low NOx" burners or catalytic denitrifiers (fig. 4).

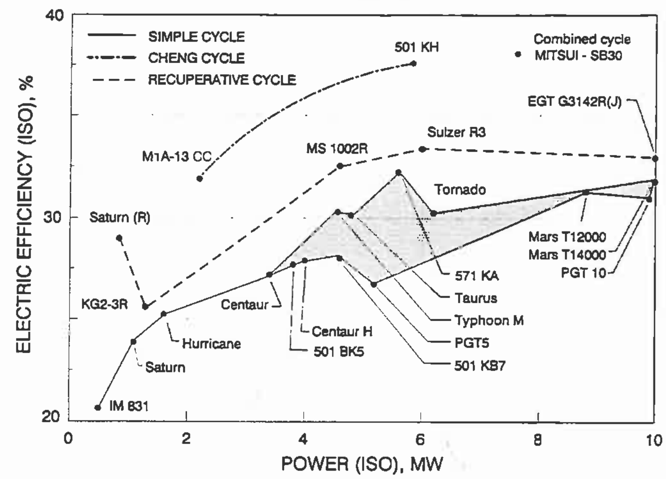

Fig. 2 *"ISO" efficiencies of medium/small gas turbines available on the market, according to "Gas Turbine World" (1992/3) data. N.B. The values shown here should not be confused with the ordinate in fig. 1. Apart from the possible "commercial" increase that numerous manufacturers add to the catalogue values, the values in fig. 1 are penalized to keep account of: (i) load losses in the inlet and outlet ducts, (ii) auxiliary consumption (in particular, gas compressor), (iii) fouling and (iv) ageing factors as well as an efficiency decline during actual operating (start-ups, stops, partial loads with respect to the "nominal" point at maximum load).*

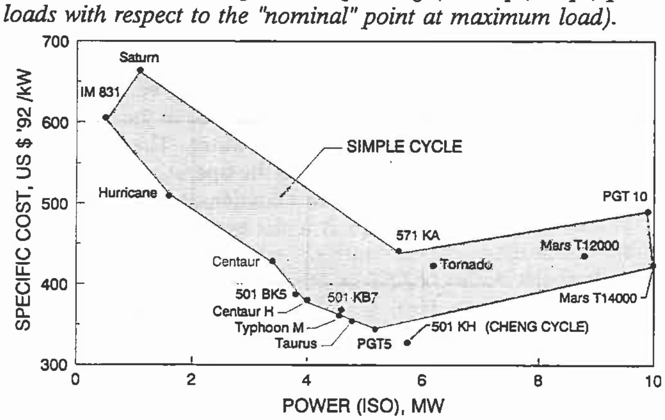

Fig.3 *"Overnight" specific costs of medium/small size gas turbine packages according to "Gas Turbine World" (1992/3) data (according to the authors' experience, these data are often considerably optimistic).*

Full steam injection is a practice widely dealt with in technical literature (from earlier work of Cheng(1978) to the three papers recently published by Rice (1993)). Although this indubitably has beneficial effects on electrical efficiency it does not, however, solve the problem of first-law efficiency nor that of primary energy saving criteria (Fig. 1). The serious disadvantage of the injection solution is the discharge to the ambient of all the

[3] Full steam-injected gas turbine plants are sometimes called "Cheng cycles". General Electric calls "STIG" the steam injected versions of its aero-derivative gas turbines.

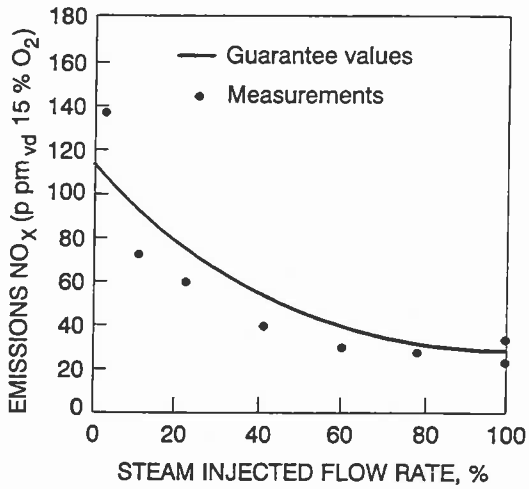

Fig.4 *Comparison of guaranteed and experimental values of NO_x emissions as a function of steam injected flow rate for the present plant. At full steam injection (9000 kg/h) the measured values are about 1/3 of the limit required by local Authorities (150 mg/mc at 15%O_2).*

steam injected. Besides the environmental problem caused by the "dissipation" of a precious material such as water, due to the high specific evaporation heat of water this also generates a large heat loss. Added to this there is the economic waste (although limited) linked to the reintegration of good quality demineralized water into the cycle.

The condensation of the steam contained in the exhaust gases and its re-use in the cycle (the best solution to the above mentioned problems) will constitute the main theme of this paper. The technical and economic feasibility of this solution will be discussed by the authors who will then go on to describe the results obtained during the experience currently underway at the Cogeneration Plant installed at the Carozzeria Bertone, the first example in the world where a solution of this type is operative on an industrial level.

TECHNICAL-ECONOMIC ANALYSIS METHODOLOGY OF COGENERATION PLANTS

A cogeneration plant is forced to operate under conditions that tend to vary in time. For example electrical and/or heat demands, environmental conditions and tariff conditions are capable of changing from hour to hour. For each set of these conditions there is an "optimum" engine functioning point. This coincides with attainment of the maximum instantaneous profit IP (evaluated in relation to a conventional system whereby the demanded heat H_{dem} is generated by means of a boiler with a prescribed efficiency and the demanded electrical energy E_{dem} is taken from the network), which can be expressed by the equation:

$$IP = (F1 + F2 + F3) - (F4 + F5 + F6) \quad (1)$$

where the first three terms refer to the conventional alternative:

F1 = is the cost of fuel used by the "reference" boiler to generate H_{dem}
F2 = is the cost of electricity E_{dem} taken from the network
F3 = is the sum of O&M expenses (excluding fuel, but including costs relative to electricity consumption of the auxiliaries) of the traditional plant

while the second three represent the same items relative to the cogeneration plant and are, unlike the first, dependent on the functioning point of the plant:

F4 = is the cost of the fuel used by the cogeneration engine (including eventual consumption of supplementary burner and integration boiler)
F5 = is the cost of electricity taken (or, when introduced into the equation with a negative sign, the profit following transfer); in this term the element relative to economic losses generated by a lack of availability of the plant (purchased by ENEL at rescue rates) is present for periods of time evaluated according to statistical bases.
F6 = is the sum of O&M expenses of the cogeneration plant, excluding fuel and auxiliary electricity consumption costs, which are directly deducted from the gross output furnished by the engine.

The situation is complicated by the presence, in equation (1), of terms that do not depend solely on instantaneous situations (e.g. the selling rate to the network depends on the "regularity" of the supply, on PER, etc.) or by the need to satisfy numerous obligations, also linked to plant behaviour throughout the year (e.g. compliance with the energy assimilation index, power commitments in the various categories, etc.). Therefore, in order to accurately evaluate the economic benefits obtainable from a cogeneration plant it is necessary to use relatively sophisticated and complex analysis methodologies (Consonni et al (89)). The COGEN simulation/ optimization code developed at the Department of Energy at the Milan Politecnico (Bertino and Bruni (1993)) was used for this analysis (for a more in-depth description of the above please refer to the paper cited above). The main steps of the calculation methodology will now be outlined:

(i) plant operation is divided up into a large number of intervals characterized by constant boundary conditions (electricity and heat demand, ambient temperature, electric and fuel tariff,etc);
(ii) the optimum contractual conditions (electric and methane supply) of the traditional reference solution are optimized;
(iii) the characteristic curves (both for one-degree and two-degrees of freedom systems, under average fouling conditions) of the cogeneration engines are defined. A different engine can be associated at each time interval: in this way the functioning of the same engine subjected to different environmental conditions (the influence of the ambient temperature on gas turbine output capacity is known to all) and/or a change in user heat characteristics (in this case, the return temperature of the water from the process) can be simulated;
(iv) first attempt values of the variables that depend on annual functioning and not on a single time interval (e.g. power commitments in different categories, type of rate, energy index, regularity premiums, etc.) are assumed;
(v) search for each of the time intervals of the optimum working condition of the cogenerating plant (see for example fig. 5 and table 1);
(vi) control, a posteriori, of the hypotheses outlined in point (iii) and complete recycling of the procedure until convergence is reached;
(vii) calculation of principal economic investment indicators (IRR,NPV, PBP).

Table 1 *Operative points (see fig. 5) whereby the COGEN code verifies instantaneous profit IP by means of equation (1). The point of maximum IP is not necessarily the final choice: other restraints such as the limit on the value of energy saving index, only verifiable in external recycling, can come into play.(pm = prime mover; sf= supplementary firing; bl = boiler; dem = demand).*

Operative point and method of functioning	E_{pm} kW	H_{pm} kW	H_{sf} kW	H_{bl} kW	IP \$/h	PER -
0 = engine at standstill	0	0	0	6000	0	-
1 = full steam injection, maximum output capacity	5292	1234	0	4766	229	0.782
2 = full steam injection, maximum supplementary firing	5292	1234	11918	0	128	0.619
3 = simple cycle, maximum output capacity	3588	7605	0	0	178	1.005
4 = simple cycle, maximum output capacity and maximum supplementary firing	3588	7605	6697	0	160	0.660
5 = simple cycle, minimum output capacity	655	1892	0	4108	21	0.775
6 = simple cycle, minimum output capacity and maximum supplementary firing	655	1892	12014	0	-76	0.450
7 = minimum output capacity, heat load follows with supplementary firing	655	1892	4108	0	20	0.885
8 = electric load and heat load follows with supplementary firing	2000	4512	1488	0	99	1.039
9 = heat load follows	2764	6000	0	0	146	1.095
10 = full output capacity, heat load with steam injection follows	4017	6000	0	0	205	1.040
11 = full steam injection, heat load with supplementary firing follows	5292	1234	4766	0	232	0.851
12 = electric load follows, auxiliary boiler connected	2000	4512	0	1488	99	1.037
13 = electric load follows, maximum supplementary firing	2000	4512	9576	0	2	0.552

THE COGENERATION PLANT INSTALLED AT CARROZZERIA BERTONE

Electricity and heat requirements

The Carrozzeria Bertone plant, where the cogeneration plant is installed, is situated in Grugliasco (Turin). Current production processes are characterized, from an energy point of view, by the daily diagrams showing electricity and heat absorption. These

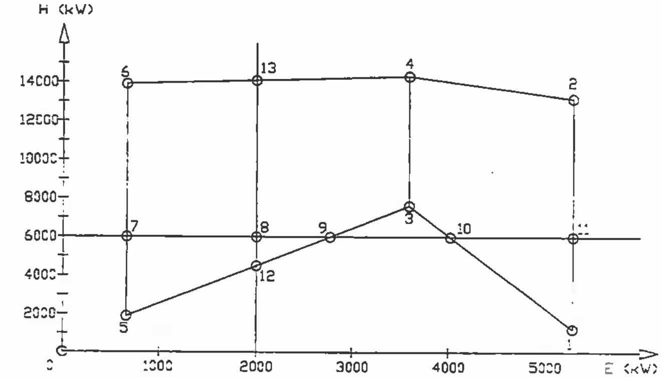

Fig.5 *Mode of operation of the code COGEN: the plant net profit IP is computed for all 14 points indicated in the figure. The example refers to E_{dem}=2000 an H_{dem}=6000 respectively. The results are given in tab.1.*

diagrams are illustrated in figures 6 and 7[4]: while the electric load trend is more or less independent of the seasonal conditions and concentrated, for the most part, during the "peak hours", the overall heat loads are heavily dependent on the external temperature and are distributed more uniformly, above all during the winter season, throughout the whole day. Despite the continuity of heat demand varying throughout the year, in order to simplify matters, the authors will refer to the three "typical" situations shown in fig. 7. These will be identified respectively as "Winter" (from November 15th to February 15th), "Summer" (from June 1st to October 15th) and "mid-season" days (from February 16th to April 30th and from October 10th to November 14th).

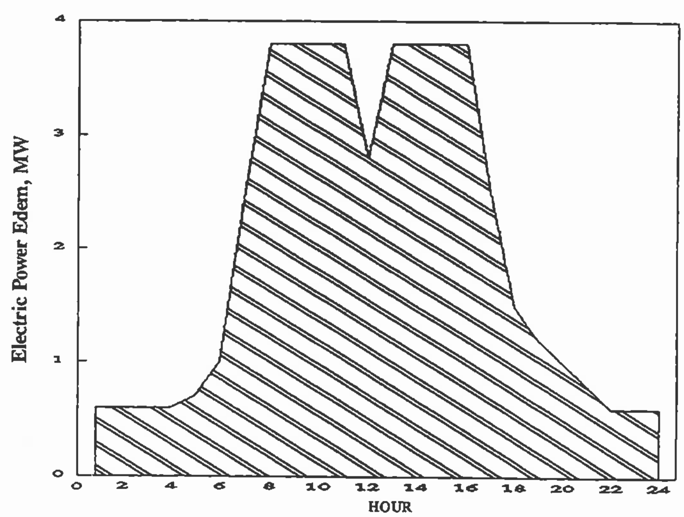

Fig. 6 *Net electricity demand trend of the Bertone plant during a working day. The electricity demand is only marginally affected by the seasonal trend.*

It is important to point out that before the construction of the cogeneration station, all heat demand, including low temperature

[4]These diagrams refer to the present situation, whereby most of the working activity is conducted during one shift. When the cogeneration plant project was originally drafted, the hypothesis of operating a double shift was put forward. This hypothesis increases the economic and energy advantages of cogeneration considerably.

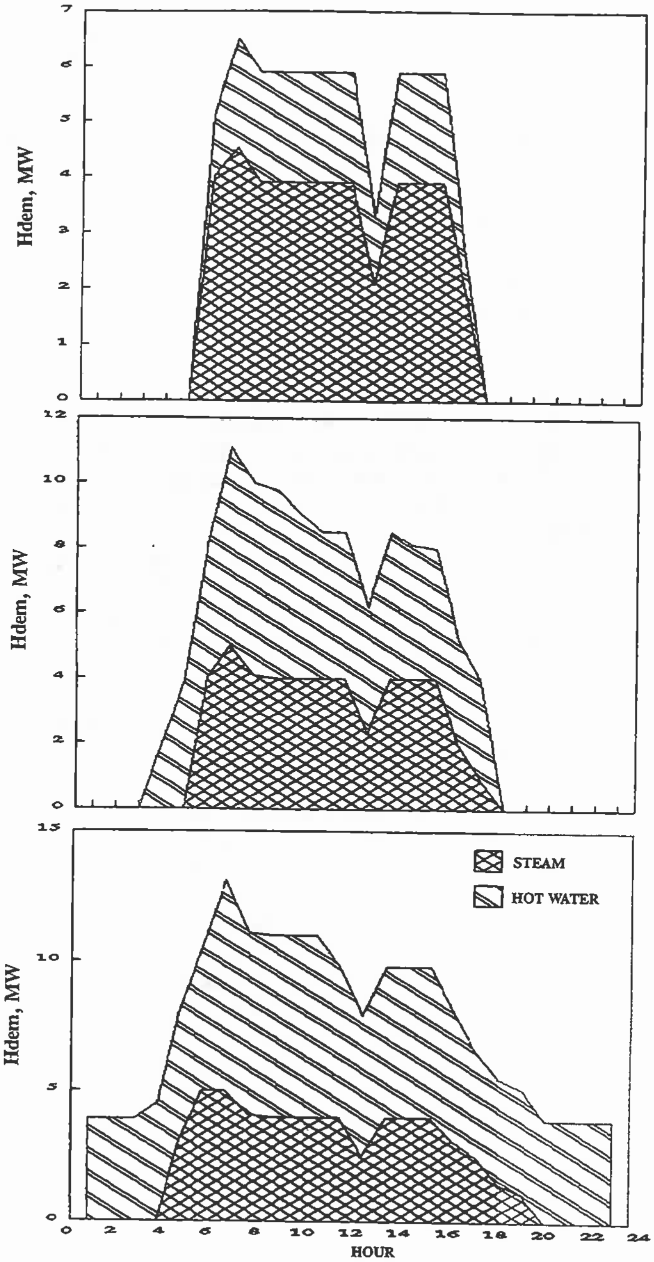

Fig. 7 *Heat demand trend of the Bertone plant during working days. The three typical situations refer respectively to the Summer, Winter, Spring/Autumn seasons. The heat loads are divided up into demand for steam and hot water. When the performance of the conventional solution is examined the two heat demands can be totalled. This is due to the fact that the plant is not capable of "taking advantage of" the low energy value of the heat output capacity of the hot water.*

demand, was satisfied by conventional boilers, generating high pressure steam. The cogeneration plant was originally designed with the idea of maintaining the pre-existing heat distribution network, so that only the sum of the heat output capacities, and not their temperature, was important.

Comparison between the simple cycle and the CHENG cycle

Two alternative systems will be examined within this framework: the simple cycle gas turbine with heat recovery and supplementary firing and the Cheng cycle. With regard to the above mentioned "typical" days, the cogeneration plants operate according to the methods (optimized by means of the above mentioned COGEN methodology) shown in figures 8 (Summer's day) and 9 (Winter's day). In short:

- during peak hours in the Summer season (fig. 8), the simple cycle gas turbine operates at full load. A good part of the heat cogenerated is however dissipated. Furthermore, the plant is incapable of satisfying the electricity demand (it is

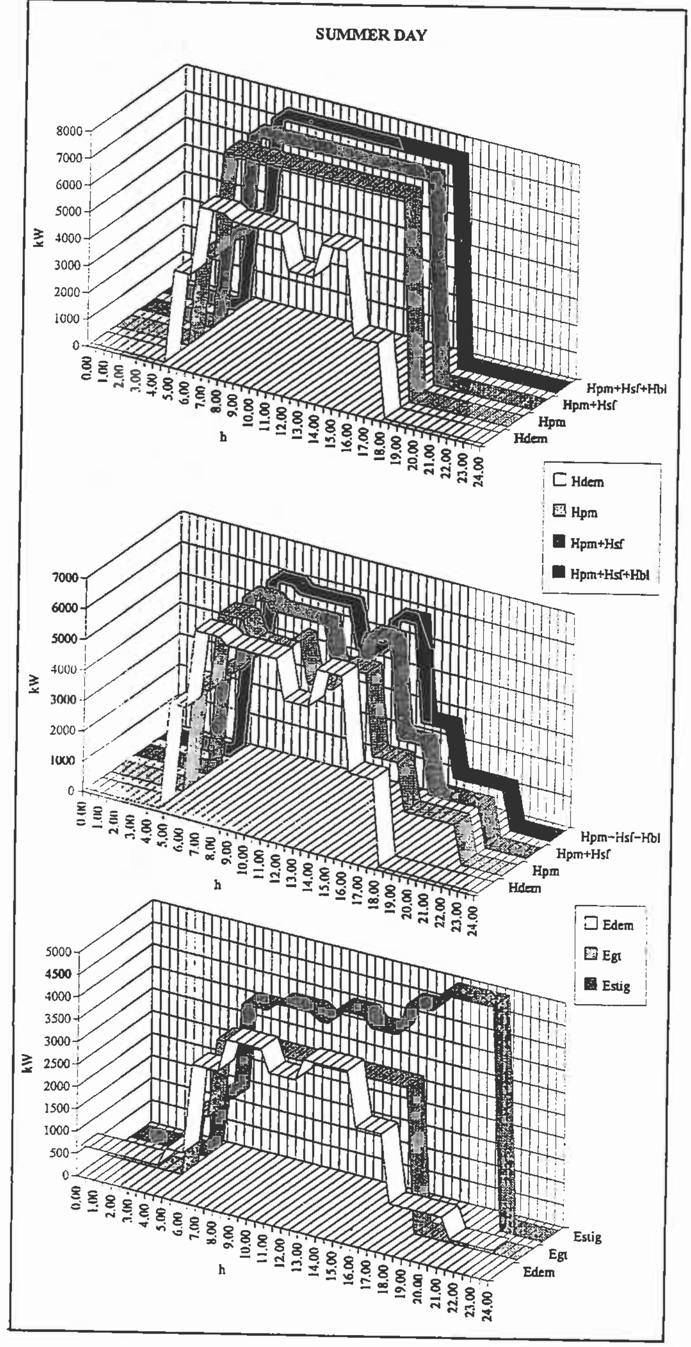

Fig. 8 *Optimized operating method of two cogeneration plants on a "typical" Summer's day.*
a) thermal power (gas turbine)
b) thermal power (Cheng cycle)
c) electricity (gas turbine and Cheng cycle)

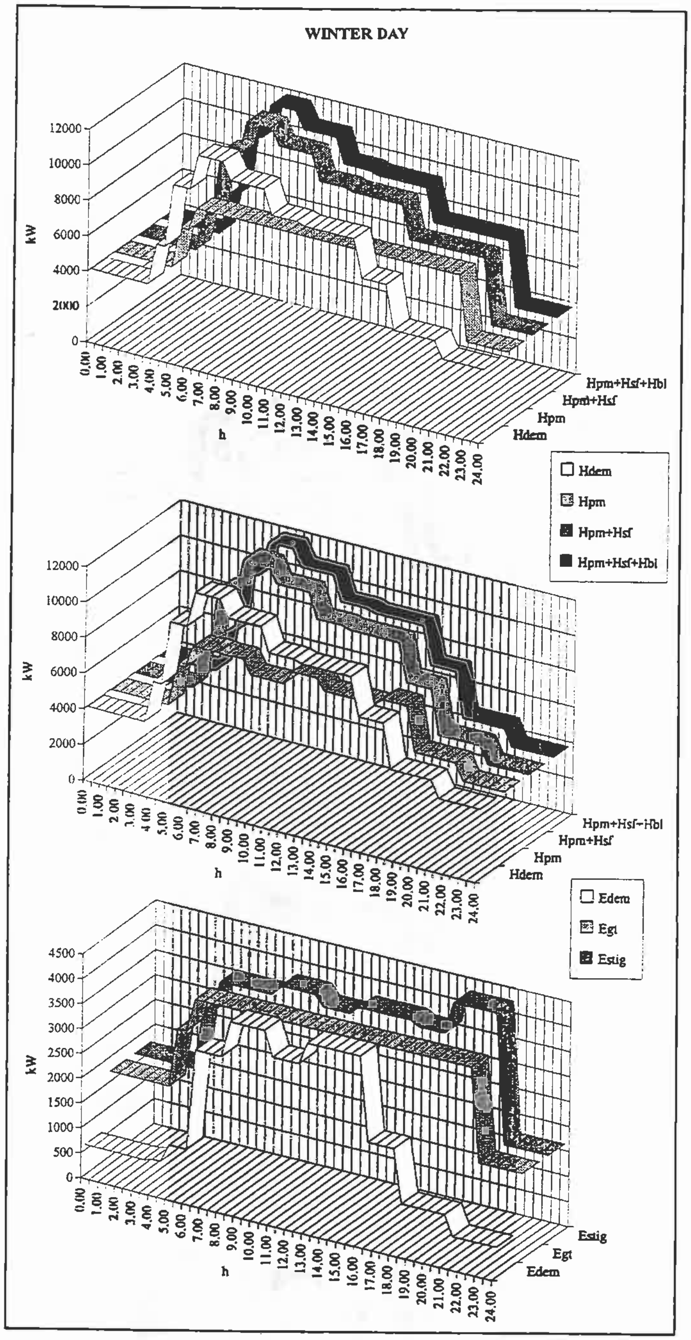

Fig. 9 *Optimized operating method of two cogeneration plants on a "typical" Winter's day.*
a) thermal power (gas turbine)
b) thermal power (Cheng cycle)
c) electricity (gas turbine and Cheng cycle)

therefore necessary to buy high-tariff power from the grid). The Cheng solution, on the other hand, operates according to operative line 1-3, by modulating the steam injection according to the heat demand. Both solutions foresee the stopping of the machine during the no-load hours.

- during the Winter season (fig. 9), the simple cycle gas turbine operates at full load during peak hours, modulating the supplementary firing to satisfy the heat loads during the early hours of the day and dissipating part of the heat during the late afternoon; it operates at partial load (heat follws) during the off-peak hours. On the other hand, during the peak hours, and depending on the quantity of heat load, the Cheng cycle follows the electric load by modulating the heat with supplementary firing, or the heat load by modulating the electrical energy yielded with the injection of steam. During the off-peak hours it does not inject steam and follows the heat load.

The economic results of the two solutions, on an annual basis, are summarized in table 2, where the items pertaining to the non-cogeneration solutions are also listed[5].

As can be seen, both solutions permit considerable operational savings. Apart from the superiority shown in table 1 (an annual saving of 190,000 US$ compared with the simple cycle gas turbine solution, in the face of an additional investment estimated at 950,000 US$), the Cheng solution was given preference in view of foreseeable future increases of self-consumed electrical energy due to the planned connection of other users to the Plant and/or a different management of the plants (two-shifts); if one assumes an average increase of about 20% of self-consumed energy, the difference between the savings obtainable with the two plant engineering typologies exceeds 300,000 US$/year.

RECOVERY OF HEAT CONDENSATION FROM STEAM INJECTION CYCLES

The idea of heat recovery by condensating exhaust gases is definitely not new: "condensation boilers" are a commercial reality, at least for small output capacities in the domestic sector, and the use of a recuperator/condenser downstream of mixed gas steam cycles has been considered by various authors, both for Cheng (Nguyen and Den Otten (1992)) as well as HAT (Day and Rao (1993)) and "Vapour Pump" (Guillet, 1992) cycles. As far as the authors know, the plant described here is the first in the world to render the process operative on an industrial scale.[6]

The amount of heat that could be recovered by cooling the exhaust gases from a steam injection cycle, at temperatures lower than the limits usually adopted, is remarkable, as long as it is possible to activate the condensation mechanism. A conservative prediction of this is shown in fig. 10, where the heat that can be

[5] The Plant was successfully tested in February 1993. The calculations made in this paper refer to the performance obtained during testing, less the average consumption of the auxiliaries encountered during the subsequent year and penalized on the basis of an "average" fouling of the machine. During testing, the energy budgets of the plant were determined in four basic positions. With table 1 terminology:

- point 1: full steam injection cycle, with "unfired" boiler recovery, maximum output capacity
- point 2: full steam injection cycle, with "fired" boiler, maximum output capacity
- point 3: simple cycle with "unfired" recovery boiler, maximum output capacity
- point 4: simple cycle with "fired" recovery boiler, maximum electricity and heat output capacity.

[6] Although the recuperator/condenser was installed prior to testing of the plant (February 1993), modifications to the heat distribution plant, aimed at giving low temperature users an autonomous network, were only completed during the summer break (August 1993). The plant has been fully operative since September 1993.

Tab.2 Summary, on an annual basis, of the economic results of the two cogeneration plants examined

	PLANT ARRANGEMENT			
	conventional (boiler+grid)	gas turbine simple-cycle	Cheng cycle (without condensation)	Cheng cycle (with condensation)
COST ITEMS ($/year)				
Fuel	467,500	690,000	772,000	812,000
Electricity	1,179,000	140,000	-154,000	-336,000
O&M	25,000	47,000	59,000	69,000
Grid black-out	62,500	-	-	-
Reintegration of demineralized water	-	-	11,000	8,000
Total	1,734,000	887,000	688,000	553,000
Operational savings	-	857,000	1,046,000	1,181,000

recovered by cooling the exhaust gases in the gas turbine at the BERTONE Plant is represented in the four operative limit (points 1-4) situations outlined in fig. 5. As can be seen, in the cases where steam injection (points 1 and 2) is used, recovery is fairly large: for example, by cooling the gases from 100 to 50 °C, approx. 35% of the heat introduced with the fuel is recovered, with double the efficiency of the first law. The condensation phase begins at temperatures of approx. 65-70 °C, although, in order to obtain a recovery equal to the capacity of the water injected, it is necessary to lower temperatures to approx. 40°C (point 1) and 50°C (point 2: with supplementary firing the content of steam is higher). By cooling the exhaust gases to even lower values, part of the water generated during the combustion reactions is also recovered and the plant "produces" water. In order to reach such low gas exhaust temperatures it is necessary to avail of a suitable cold source. It is also necessary to size the heat exchanger accordingly. With regard to the cogeneration plant described here, the only possibility of making use of the heat provided by the condensation process of steam was to separate the "privileged" (steam) heat users from the "non-privileged" (hot water) ones: a careful examination of heat consumptions enabled us to identify large heat consumers capacities capable of returning the water to the recuperator/condenser at temperatures of approx. 50 °C during the entire winter period, even at the expense of a complex and costly reorganization of the heat distribution network.

The typical technology used in air dehumidification processes was adopted to size the heat exchanger: continuous fin coils with copper tubes and aluminum fins. It should be noted that the secondary exchange surface chosen (a thin aluminum fin) permits generous sizing of the exchanger. In fact, the specific costs of this solution are considerably lower than the typologies normally adopted in recovery boiler exchangers (steel pipes): this is due to the fact that it is active on both sides, that the material has a lower density and, above all, the fact that it is not as thick. Furthermore, the condensation phenomenon considerably augments up the heat exchange, resulting in overall exchange coefficients which are significantly higher than those of a "dry" recovery boiler, while maintaining low speeds and modest head losses. A forecast of the heat exchange performances obtained by varying the sizing of the exchanger is shown in fig. 11, for different sets of inlet/outlet temperatures of water.

It should be pointed out that the curves shown in fig. 10 were taken from simple thermodynamic balances and are, therefore, extremely conservative. In reality, part of the condensation heat is recovered even for bulk temperatures of exhaust gases which are higher than the dew point temperature: condensation takes place when temperatures lower than the dew point are reached in the boundary layer which comes into contact with the cold surfaces. This phenomenon is dealt with correctly in the heat

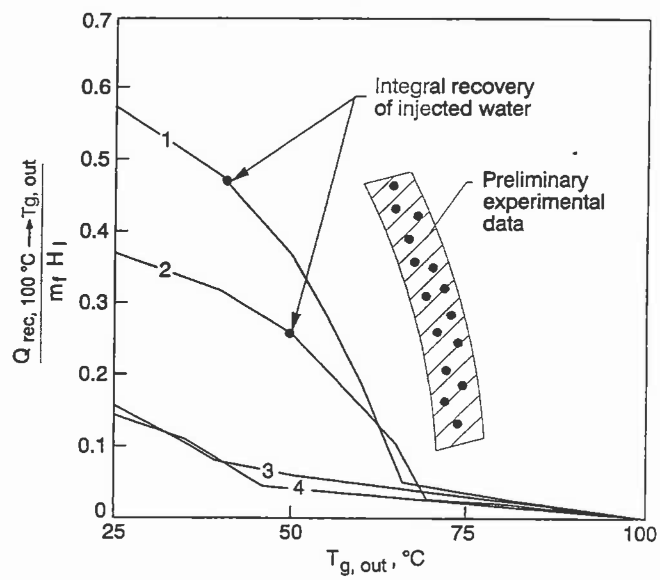

Fig.10 *Influence of the exhaust gas temperature $T_{g,out}$ on the relationship between the heat that can be recovered by cooling the exhaust gases from 100 °C to Tg,out and the heat introduced into the cycle with the fuel (reference: LHV). As outlined in the text, the heat output capacities are extremely conservative, since do not account for condensation occurring in the boundary layer on the fins, which is at temperature much lower than bulk temperature. This explains the discrepancy with the experimental results, which refer to operating point 1 (full steam injection), with return water temperature of about 60 °C. During Winter, with return water temperature of about 50 °C, it is expected the obtain the full recovery of the injected steam.*

exchange analysis in fig. 11, where much larger heat recovery (and therefore greater amount of water condensation) than predicted by the curves of fig. 10 are found, especially for generous heat transfer surfaces. The preliminary experimental results obtained in the real plant confirm this statement, as shown in fig. 10: a significant amount of steam is condensed even with exhaust gas temperature higher than dew point.

As shown in fig. 12, as the temperature at which the exchange gases are cooled varies, the operating curves of the engine are modified[7]. By assuming $T_{g,out}$ = 60°C, unlike what occurs in the absence of condensation, it can be noted how the amount of recoverable heat increases with the increase in the amount of steam injected (and consequently electrical output capacity). This is especially true in the presence of afterburner.

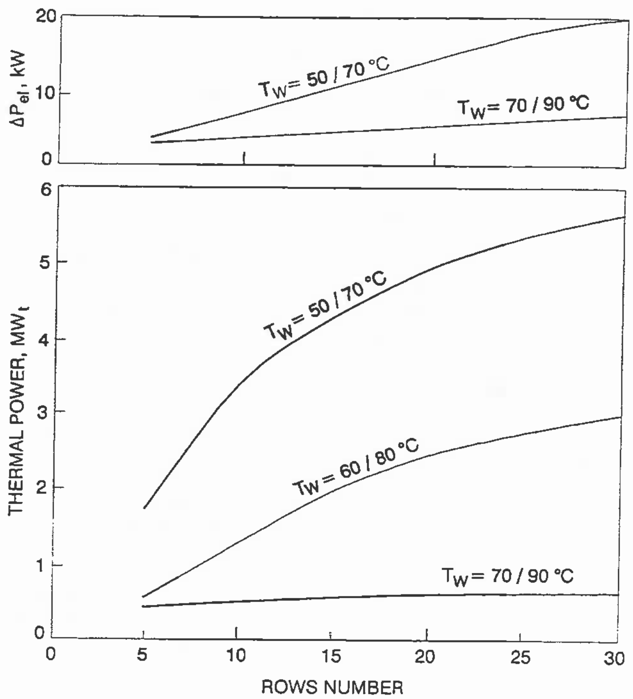

Fig.11 *Performance of the recuperator/condenser subject to the change in the number of rows of the finned battery (point of functioning 1) and the inlet and outlet temperatures of water. These calculations were made using the calculation code developed by Macchi and Lozza (1989). They refer to an exchange matrix with an external/internal surface ratio of approx. 10.3. Each row has a heat transfer surface of 162 square meters and the frontal speed of gas is around 2.1 m/s. The power losses in the gas turbine caused by the additional head losses related to condensation occurring in the exchanger are shown in the upper diagram.*

As shown in fig. 13, the influence of these improved thermodynamic performances results in a drastic change in the optimum use of the engine during the winter season (the only season during which a return temperature lower than 50°C can be maintained). It also ensures an excellent economic return (last column in table 2) of the additional investment (estimated in the order of approx. 300,000 US$): compared to a modest increase of fuel consumption, the plant is able to generate more electricity.

[7]The curves in fig.13 are calculated with the same conservative assumptions of fig.11. In reality, the recovered heat is larger.

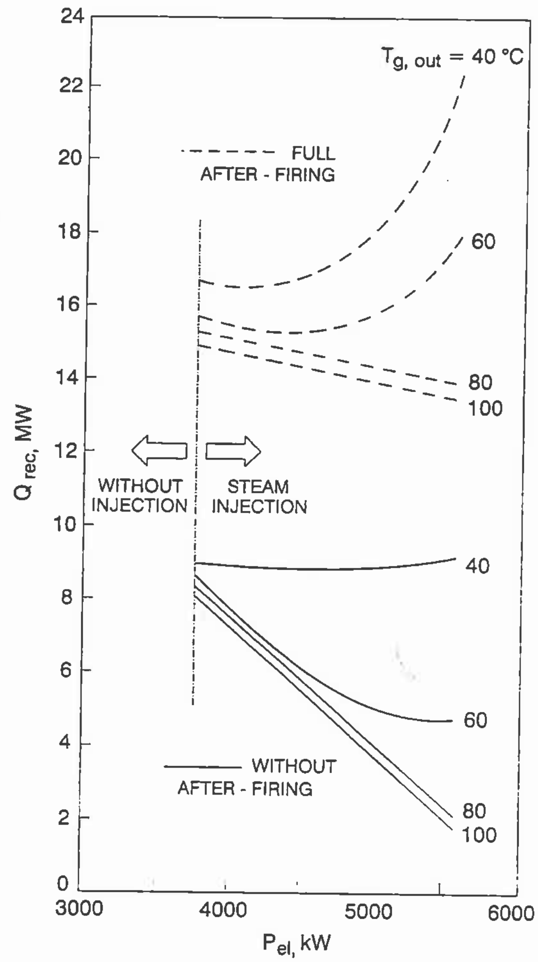

Fig.12 *Operating curves of the engine as the exhaust gas temperature changes at the outlet of the recuperator/economizer.*

Economics of the Project

The overall investment cost (US$) was as follows:

engineering &tests:	300,000
civil works:	175,000
gas compressor(including connections to natural gas pipeline):	350,000
main equipment (gas turbine, HRSG, instrumentations and controls):	3,000,000
additional recuperator/condenser (heat transfer surfaces&structures):	290,000
balance of plant (including connections to existing heat distribution network & electrical connections):	415,000
Total:	4,530,000

According to predictions, this investment ensures an excellent economic return: the annual energy saving is estimated (last column in table 2) to be about 1,200,000 US $/year. However, like all new initiatives, the addition of a condensation system to the Cheng cycle described here, presented some unknowns. In fact, the following still have to be verified: (i) the heat exchange performances of the system under unconventional operating

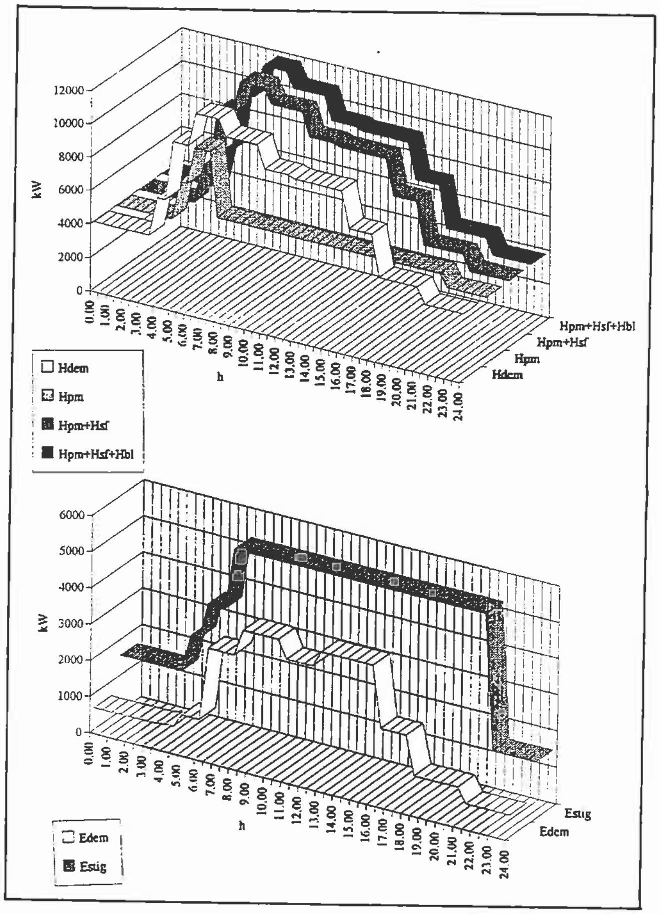

Fig.13 *Optimized operating method of cogeneration plant (including condensation) on a "typical" Winter's day: a) thermal power; b) electricity. A comparison with the results of fig. 9 shows the increased amount of electricity generated.*

conditions (Bombarda,1994), (ii) the quality of the water recovered (from the first samples taken this shows a certain acidity (pH= 4.1), mostly due to the presence of CO_2, but it appears that it can be recycled at an intermediate stage of the demineralization process) and (iii) the compatibility/duration of the low-cost (copper-aluminum) materials chosen for the exchanger,that could affect its life. At the time of writing this report (September 1993) the first experimental results pertaining to the performance of the recuperator /condenser system are coming to the fore. These are, however, only partial results due to the environmental temperature which rarely reaches values low enough to start condensation. It is therefore not possible to draw definitive conclusions regarding the merits of the project.

CONCLUSIONS

The history of the cogeneration plant installed at the Carozzeria Bertone (fig.14) demonstrates how corporate awareness of energy and environmental problems, together with a broadminded view to innovative technical solutions, can result in the construction, in a short time of plants offering considerable advantages not only in technical but also economic-managerial terms:

- towards the end of 1989 the energy manager of Carozzeria Bertone began a feasibility study for the installation of a cogeneration plant. Once the validity of the initiative had been verified, the possible plant typologies were assessed.
- in 1991 the "Cheng" cycle was chosen. This was considered the only system (for the size in question) capable of following, both from technical as well as economic point of view, the specific temporal variations inherent in the heat and electricity demands of a manufacturing company.
- in September 1991 the plant order was issued.
- in Autumn 1991 the innovative condensation system was designed for the production of hot water and simultaneous recovery of the water injected into the turbine (Poggio, 1992).
- in December 1992 the condensation stage was installed in the cogeneration plant.
- in February 1993 the plant became commercially operative and in September 1993 so did the condensation stage, which feeds the hot water network constructed "ad hoc".

Even with the uncertainties already discussed, there can be no doubts about the thermodynamic quality of the project which, added to an already efficient cogeneration solution, ensures considerable advantages both in terms of savings as well as efficiency. The plant is a good demonstration that, in order to achieve real optimization of a cogeneration plant project, it is essential to critically evaluate not only the "quantity", but also the "quality" of heat required by the end users. It is therefore necessary to carefully examine the possibility of modifying the heat distribution network and, where appropriate, the production processes themselves in order to combine, as effectively as

Fig.14 *View of the plant. It took some effort to convince the local community that the large plume coming out of the condenser was not polluting the ambient.*

possible, the characteristics of the cogeneration system with user demands. Obviously, the plant process requires (i) the availability of natural gas and (ii) the presence of low-temperature users. However, with proper sizing and management of heat distribution network and utilization terminals, these conditions can be met quite often, in commercial as well as in industrial applications.

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