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UP-RATED REHEAT GAS TURBINE FOR THE REPOWERING OF STEAM POWER PLANTS

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ABSTRACT

In the present paper, a thermoeconomic analysis of combined cycles derived from existing steam power plants is performed. The gas turbine employed is a reheat gas turbine. The increase of the two combustor outlet temperatures was also investigated.

The study reveals that the transformation of old conventional fossil fuel power plants in combined cycle power plants with reheat gas turbine supplies a cost per kWh lower than that of a new combined cycle power plant, also equipped with reheat gas turbine. This occurs for all the repowered plants analyzed.

Moreover, the solution of increasing the two combustor outlet temperatures resulted a strategy to pursue, leading, in particular, to a lower cost per kWh, Pay Back Period and to a greater Internal Rate of Return.

INTRODUCTION

The transformation of existing steam power plants (SPP) is a matter of current interest, mainly due to the widespread availability of aged steam power plants. Repowering and different repowered lay-outs have been investigated by many authors (Walters et al., 1988, Pace and Walters, 1996, and Bazzini, 1996).

In particular, in Negri di Montenegro et al. 1998a and 1998b, the conversion of SPP in combined cycle plants (CC) is carried out.

This conversion is performed by adding a gas turbine (GT) as topper, removing the steam generator and maintaining the steam turbine and condenser. Moreover, the heat recovery boiler is fed by exhaust gas discharged from the gas turbine. It should be observed that, in this way, the steam turbine and condenser work in off-design working conditions because of the steam mass flow rates that differ from those of the design case.

This type of repowering may be attractive for the higher efficiency achievable than in the existing SPP and the availability of the old site. This permits the realization of the repowered plant in a short period, also thanks to the simplified bureaucratic procedures (local/state permits).

The present study reports a thermoeconomic analysis related to the conversion of existing steam power plants in three pressure level reheat combined cycle plants, employing, in particular, a reheat gas turbine (RHGT) as topper.

Firstly, a thermodynamic analysis is performed for the different repowered plants. An increase in the two combustor outlet temperatures of the reheat gas turbine was also considered and the performance of the repowered plants with these new gas turbine working conditions was evaluated.

Subsequently, the economic analysis of the repowered plants is carried out by determining the cost per kWh, as well as other economic parameters, such as the Pay Back Period and the Internal Rate of Return. These economic variables for each of the combined cycles deriving from existing steam power plants, are then compared with those relating to a new reference combined cycle power plant, equipped with the same RHGT.

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NOMENCLATURE

CE	thermal power, referred to the hot gas mass flow rate coming from the combustor outlet, that the hot gas itself exchanges with the cooling flow, in the node n
COT	combustor outlet temperature
h	enthalpy
IRR	Internal Rate of Return
LHV	Lower Heating value
m	mass flow rate
NFW	Number of Feedwater preheaters
NPV	Net Present Value
p	pressure
P	plant electric power
PBP	Pay Back Period
T	temperature
S	heat recovery steam generator overall heat exchange surface
W	specific gas turbine power (referred to the air mass flow rate at the gas turbine inlet)
β	GT pressure ratio
η	plant efficiency (LHV)

Subscripts

c	cooling flow
cc	combined cycle power plant
g	hot gas
GT	gas turbine
n	relative to the physical state obtained by mixing the hot gas at the combustor outlet with the cooling flow. Starting point of the expansion process
r	repowered plant
s	steam
S	relative to the two combustors
sh	superheater
SPP	Steam Power Plant
st	stack
1	first gas turbine expander / combustor
1&2	first and second combustor
2	second gas turbine expander / combustor
4	gas turbine outlet

Acronyms

CC	Combined Cycle power plant
GT	Gas Turbine
HRSG	Heat Recovery Steam Generator
n	mixing node before the expander inlet
RHGT	Reheat Gas Turbine
SPP	Steam Power Plant

THERMODYNAMIC ANALYSIS

The Reheat Gas Turbine

The employed gas turbine is a reheat gas turbine whose schematic lay-out is reported in Fig.1. The reheat gas turbine cycle has been performed considering, in particular, three cooling flows for the most thermally stressed parts. They are relative to:

- the first expander (the cooling flow comes from the compressor outlet);
- the second combustor (here the cooling flow comes from an appropriate compressor pressure);
- the second expander (the cooling flow comes from the same compressor pressure of the second combustor cooling flow).

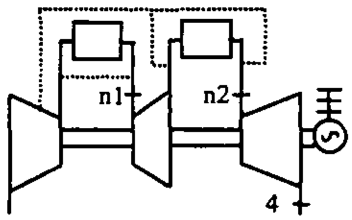

Fig.1 RHGT schematic lay-out

The expansion process, in the first and second expander, was modeled supposing to mix the cooling flow at the expander inlet (before entering the expander itself) with the hot gas coming from the combustor. After the mixing, the resulting physical state (n_1 or n_2) is the starting point for the expansion process.

Moreover, the cooling flow, for the second combustor, was assumed to be mixed with the hot gas coming from the first expander outlet, before entering the second combustor itself.

The main assumed thermodynamic specifications are listed in Table I. The values reported in the table may be considered realistic for a gas turbine with this kind of cycle.

Table I RHGT Main Specifications

β	30
COT_1 [°C]	1300
COT_2 [°C]	1300
T_4 [°C]	610
W_{GT} [kJ/kg]	450
η_{GT} [%]	38

The polytropic expansion efficiencies were set to 0.87 for both expanders, thus taking into account the penalization due to the presence of cooling mass flow rate in the expansion process (Hofstädter et al., 1998). The polytropic compression efficiency was assumed equal to 0.91, state of the art value for this gas turbine size. The expansion pressure ratio of the high pressure turbine was assumed equal to 2.

On the basis of all these assumed parameters, the cooling mass flow rates (referred to the mass flow rate at the compressor inlet), were then evaluated and resulted equal to 23% and 15% for the first and the second expander (included the cooling flows for the second combustor), respectively.

The power output was then evaluated for a mass flow rate at the gas turbine inlet of 530 kg/s and resulted equal to 240 MW.

All the calculations, for the reheat gas turbine and for the repowered power plants, were carried out with a commercial computational code (Gate Cycle, release 4.1).

The Up-Rated Reheat Gas Turbine

It must be noted that $COT_{1\&2}$ values are lower than in the case of a Brayton cycle gas turbine of the same size.

Since an increase in the $COT_{1\&2}$ values could supply greater gas turbine performance, even taking into account the greater cooling mass flow rate and off design working condition, an up-rating of the RHGT, obtained by an increasing of $COT_{1\&2}$ values, was analyzed.

In order to achieve that, the "cooling effect" (CE) was utilized. This parameter, introduced by the Authors (Moro et al., 1990), derives from the following considerations.

If n (n1 or n2) is the physical state of the gas at the expander inlet, after the mixing with the cooling flow (Fig.1), the thermal power balance in that node must give:

$$m_c\left(h_{nc}-h_c\right)=m_g\left(h_g-h_{ng}\right) \tag{1}$$

where m_c and m_g are the cooling and the hot gas mass flow rates mixing in the node n, respectively. Moreover, h_{nc} and h_c represent the cooling flow enthalpies after and before the mixing with the hot gas (in node n), respectively. Instead, h_g and h_{ng}, represent the hot gas enthalpies, before and after the mixing with cooling air (always in node n), respectively.

Then, the "cooling effect" may be considered as the thermal power, referred to the hot gas mass flow rate coming from the combustor outlet, that the hot gas itself exchanges with the cooling flow, in the node n. From Eq.(1) it may be written:

$$CE=\frac{m_c\left(h_{nc}-h_c\right)}{m_g} \tag{2}$$

or :

$$CE=h_g-h_{ng} \tag{3}$$

Of course, h_{nc} and h_{ng} are enthalpies of the two compounds (cooling air and hot gas) of the same mixture and, therefore, at the same temperature (T_n, resulted from the mixing of the hot gas with the cooling flow in the node n).

To foresee the off-design performance of RHGT previously introduced when an increase in the $COT_{1\&2}$ values occurs, a methodology regarding the evaluation of the cooling flows for different $COT_{1\&2}$ values has been necessary.

To perform this, it was supposed, for each expander, to vary the "cooling effect" (CE) when an increase in the $COT_{1\&2}$ values is assumed. The variation law was derived by analyzing the trend of CE versus COT for different Brayton cycle gas turbines available on the market (Fig.2). In particular, it was assumed that this trend remains the same for the CE of the first and second expander (of the reheat gas turbine) when $COT_{1\&2}$ values are increased from the design.

The cooling mass flow rates relative to the first and second expander, were therefore evaluated with an iterative procedure by using the CE in Eq.(2) or (3), once the $COT_{1\&2}$ values were set.

The Authors then evaluated the CC part load performance (following the load variation of a reheat gas turbine expressed in Joos et al., 1998), assuming that the polytropic compression and expansion efficiencies remain constant. The results of this investigation well agree, in the whole range of power, with those reported in Hauenschild and Jury, 1995 and relative to a combined cycle with a reheat gas turbine as topper having the same cycle of the gas turbine considered in this study.

As a consequence, in the present investigation, the RHGT performance (obtained by varying $COT_{1\&2}$) were evaluated by maintaining the polytropic compression and expansion efficiencies constant.

The cooling mass flow rate relative to the second combustor was linearly increased with the ratio between the average temperature of first expander outlet and COT_2, and the design average temperature.

The performance of the RHGT were then evaluated for different temperature values at the first and second combustor outlet, considering the two expander inlets in choking condition. The $COT_{1\&2}$ values were increased up to those of F technology Brayton cycle gas turbine, with the constraint that the maximum temperature value at the second expander outlet be below 670 °C.

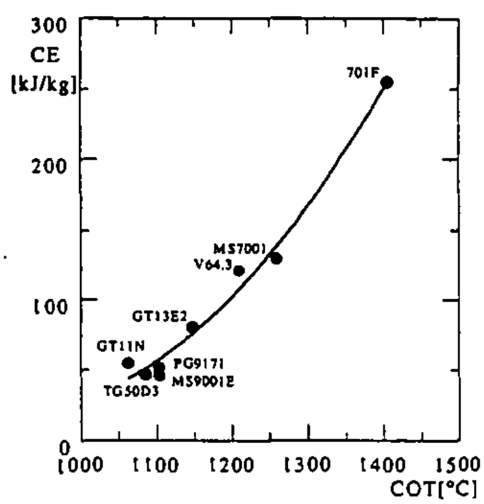

Fig.2 CE versus COT for some Brayton cycle gas turbines

The Repowered Plants

The SPP considered for repowering are 320 MW and 160 MW SPP, fueled by natural gas and oil. These sizes of conventional fossil fuel plants are particularly widespread, especially in Italy. In Table II, the main specifications for these plants, in design condition, are reported.

Table II SPP Main Specifications (design condition)

	160 MW	320 MW
p_{sh} [MPa]	13.8	16.6
T_{sh} [°C]	538	538
m_{sh} [kg/s]	132.0	290.5
NFW	7	7
P_{SPP} [MW]	145.5	307.8
η_{SPP} [%]	37.5	39.8

As far as the compatibility between the gas turbine and the existing steam turbine and condenser is concerned, it is known that the gas turbine exhaust gas temperature and mass flow rate influence the value of generated steam.

As discussed in previous papers (Negri di Montenegro et al., 1998a and 1998b), an upper limit exists for steam mass flow rate. This limit is caused by the pressure difference between the low pressure steam turbine last stage inlet and the condenser, which must never exceed the design value by more than +20% in order to assure that the mechanical stress in the last stage is within limits.

It was verified that the RHGT exhaust gas temperature and mass flow rate determine, for the two repowered plants, steam mass flow rates under the above mentioned maximum limit allowed. This occurs for all the $COT_{1\&2}$ values considered.

The assumptions adopted for the performance evaluations of the examined repowered plants were previously described in Negri di Montenegro et al., 1998a and Negri di Montenegro et al., 1998b. As mentioned, the calculus was carried out with Gate Cycle software. This code takes into account the main off-design aspects, besides those of the above mentioned gas turbine, also of the steam turbine (working in sliding pressure) and condenser. The governing stages of the high pressure steam turbine were replaced with new ones to obtain an inlet pressure maximizing the efficiency of the repowered plant.

The heat exchange surfaces inside the HRSG were determined with the NTU method. The relative overall heat exchange coefficient values and the minimum temperature difference were set thanks to data on the heat recovery boilers supplied to ENEL by HRSG manufacturers.

The values of performance, number of gas turbine units, stack temperature and HRSG surface are reported in Table III for the repowered plants. The results are relative to the employment of the above mentioned reheat gas turbine (with the main specifications in Table I).

The number of GT for each plant was chosen in such a way as to obtain the best efficiency of the repowered plant without making the repowered plant lay-out too complex.

The following Fig.3 and 4 report efficiency (LHV) versus power of the repowered plants, for different values of COT_1 and COT_2. The values of COT_2 are limited to 1360 °C, since higher values determine a T_4 greater than 670 °C.

It should be observed that, for both the repowered plants, the increase in COT_1 and COT_2 leads to an increase in power and efficiency. This occurs for all the values of COT_1 and COT_2 considered.

Moreover, in both cases, the increase in power (maximum 4.6%, in Fig.3, and maximum 4.2% in Fig.4) are remarkable, while those relative to efficiency are less significant.

Table III Repowered Plant Main Specifications

CC	CC derived from 160 MW SPP	CC derived from 320 MW SPP
Number of RHGT	1	2
P_r [MW]	344	700
η_r [%]	54.2	55.1
T_{st} [°C]	90	100
S [m^2]	187773	381332

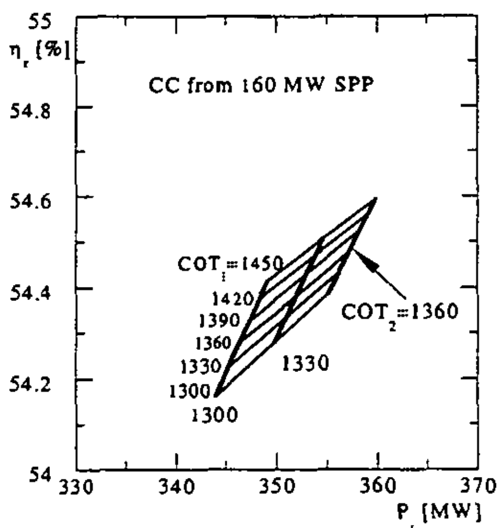

Fig.3 η_r versus P_r for different COT_1 and COT_2 values. (case of the CC derived from 160 MW SPP)

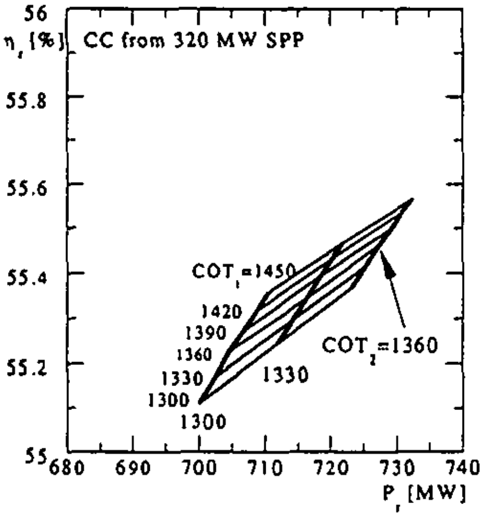

Fig.4 η_t versus P_t for different COT_1 and COT_2 values. (case of the CC derived from 320 MW SPP)

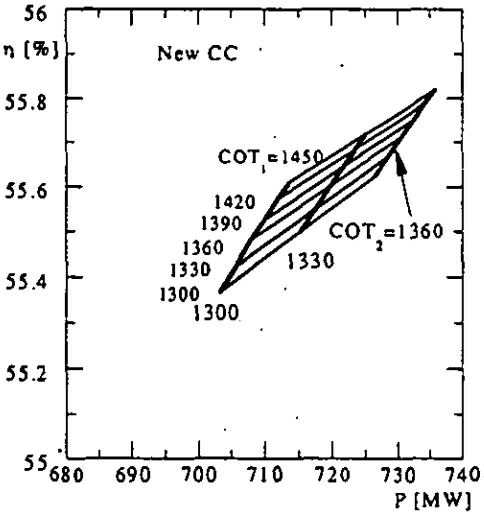

Fig.5 η versus P for different COT_1 and COT_2 values. (case of the new CC)

The new CC taken as reference

The new reference combined cycle power plant under study has the reheat gas turbine (with the main specifications in Table I) as topping cycle and a three pressure level reheat steam cycle as bottoming.

Obviously, in this case, all the components work in design working condition.

In the performance evaluation, the pressures in the vaporizers were varied to optimize the efficiency of the CC.

For every HRSG heat exchanger, the overall heat exchange coefficient and the minimum temperature difference values, were assumed as above.

Table IV reports the values of performance, number of gas turbine units, stack temperature and heat recovery boiler overall heat exchanger surface for this new combined cycle power plant.

Here, in order to investigate a proposed combined cycle power plant lay-out, two gas turbine units were utilized.

Table IV New CC Main Specifications

Number of RHGT	2
P [MW]	703
η [%]	55.4
T_{st} [°C]	90
S [m^2]	391185

In Fig.5, efficiency (LHV) versus power of the new CC, for different values of COT_1 and COT_2 are reported. Here again, the values of COT_2 are limited to 1360 °C, since higher values determine a T_4 greater than 670 °C.

Also in this case, the increase in COT_1 and COT_2 leads to an increase in power and efficiency. This occurs for all the values of COT_1 and COT_2 considered.

Furthermore, the increase in power (maximum 4.5%, in Fig.5) is much higher than the one relative to efficiency.

ECONOMIC ANALYSIS

The economic analysis was carried out by evaluating the cost per kWh, net present value (NPV), pay back period (PBP) and internal rate of return (IRR) for the plants analyzed. As far as the NPV is concerned, its evaluation was performed assuming a reference cost per kWh equal to 41.7 mills/kWh.

Obviously, the total cost per kWh produced by the plant is the sum of three different terms: the capital cost of the plant; the cost of fuel; the cost due to the plant employees, maintenance and operation.

In the case of transformation of the SPP to a combined cycle, the capital cost is mainly supplied by the cost of the gas turbines, the heat recovery steam generator and the replacement of the high pressure steam turbine governing stages. Moreover, in this cost, the costs relative to the revamping of the steam turbine (especially high and middle pressure), of the electric generator of steam turbine and of the condenser are included. For the new combined cycle power plant, the capital cost is relative to the purchasing of all the components and the construction of the new plant.

The Repowered Plants with the Reheat Gas Turbine in design condition

The evaluation of all the previously introduced parameters was performed for the combined cycles derived from the existing two SPP, along with the new combined cycle power plant. For these calculations, an annual interest rate equal to 12%, a plant working period of 6000 hours/year and an economic life of the plant equal to 15 years, were assumed. The 6000h/y are equivalent hours of a plant working at the maximum continuos rate.

The price of the RHGT was considered to be 172 \$/kW. As far as the price of the electric generator is concerned, this was included in the capital costs. Moreover, it was assumed that all the repowered plants are fed with natural gas, whose LHV is 34750 kJ/Nm^3 and cost is 0.111 \$/Nm^3.

The main thermoeconomic results of these calculations are presented in Table V, where the cost per kWh relative to employees, maintenance and operation is indicated as "other costs".

It should be noted that the cost per kWh for the two repowered plants turns out to be lower than that of the new CC (35.59 mills/kWh).

In particular, the cost per kWh of the CC derived from the 320 MW SPP results to be about 29.65 mills/kWh, the PBP turns out to be around 3.2 years, against 7.2 years for the new CC. The IRR is about 39.7%, against 20.2% for the new CC.

Furthermore, the cost per kWh for the CC derived from 160 MW SPP is about 30.78 mills/kWh. The PBP and IRR are around 3.5 years and about 36.2%, respectively.

Therefore, the two repowered plants prove to be more economically advantageous than the new CC.

Table V Main Thermoeconomic Results

		CC from 160 MW SPP	CC from 320 MW SPP	New CC
Net steam power	[MW]	104	220	223
Net gas turbine power	[MW]	240	480	480
TOTAL POWER	[MW]	**344**	**700**	**703**
TOTAL CAPITAL COST	[M\$]	**102.7**	**199.9**	**374.3**
Fuel plant consumption	[Nm^3/h]	65898.2	31796.6	131796.6
Cost per kWh due to capital	[mills/kWh]	7.31	6.99	13.03
Cost per kWh due to fuel	[mills/kWh]	21.29	20.92	20.82
Cost per kWh due to others costs	[mills/kWh]	2.19	1.75	1.75
TOTAL COST per kWh	[mills/kWh]	**30.78**	**29.65**	**35.59**
NPV	[M\$]	153.0	343.7	174.6
PBP with 12%	[y]	3.5	3.2	7.2
IRR in 15 years	[%]	36.2	39.7	20.2

The Repowered Plants with the Up-Rated Reheat Gas Turbine

In order to evaluate the influence of the $COT_{1\&2}$ increase on the plants' economic parameters, the cost per kWh, PBP and IRR were evaluated for each plant, for different values of COT_1 and COT_2. The percentage variations of these economic parameters (referred to the values reported in Table V) are plotted in the following Fig. 6, 7 and 8, versus COT_1 and for different COT_2 values.

It should be pointed out that the increase in $COT_{1\&2}$ (and in power), obtained by means of some modifications on the RHGT unit (widening the cooling flow pipeline area, for example), leads to increasing the total cost of the RHGT itself. As a consequence, the cost per kW of this turbine was assumed constant (with $COT_{1\&2}$ variations) and equal to that of the RHGT in design condition (172 \$/kW).

It must be noted that both COT_1 and COT_2 increases have a positive influence on the cost per kWh, PBP and IRR. This occurs for all the three plants examined. Moreover, for assigned COT_1 and COT_2 values, the beneficial effect is the highest for the new combined cycle, even though the new combined cycle has the highest cost per kWh with respect to the repowered plants considered (Table V).

The cost per kWh, PBP and IRR variations are weakly influenced by the type of repowered plant considered (320 or 160 MW SPP).

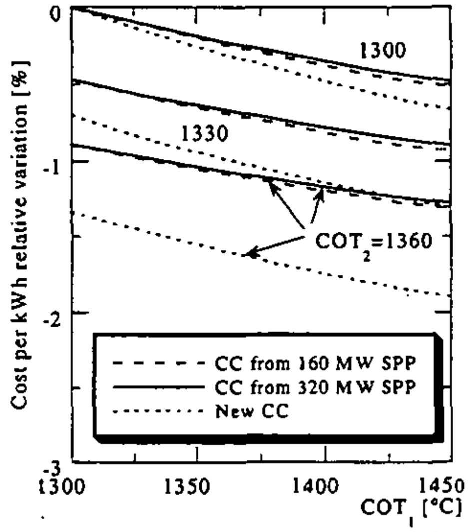

Fig.6 Cost per kWh relative variation [%] versus COT_1 for different COT_2 values

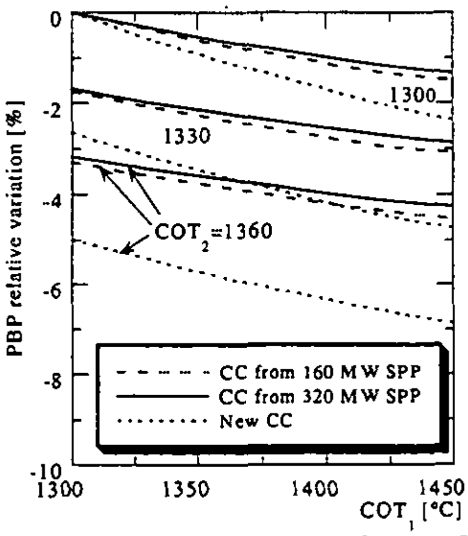

Fig.7 PBP relative variation [%] versus COT_1 for different COT_2 values

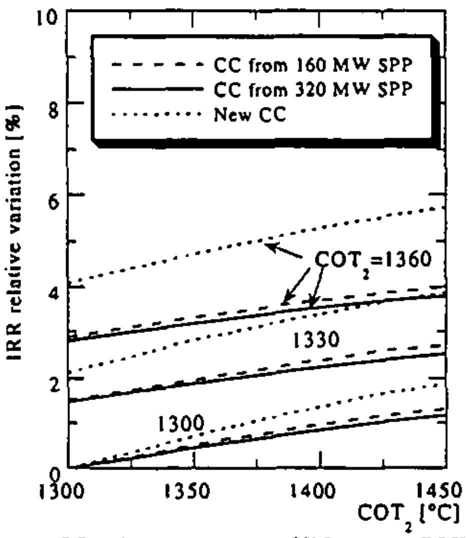

Fig.8 IRR relative variation [%] versus COT$_1$ for different COT$_2$ values

CONCLUSIONS

The present paper has highlighted that the transformation of conventional steam power plants in combined cycle is more economically advantageous (in terms of cost per kWh, PBP and IRR) than building a new combined cycle power plant, when a reheat gas turbine is employed.

In particular, among the repowered plants investigated, the cost per kWh turned out to be the lowest for the repowered plants derived from 320 MW steam power plant. In fact, the cost was about 30 mills/kWh against about 31 mills/kWh for the CC derived from 160 MW SPP. Instead, the cost per kWh of the new combined cycle power plant, also equipped with reheat gas turbine, resulted in about 36 mills/kWh.

Finally, the influence of the $COT_{1\&2}$ increase on the plants' economic parameters (cost per kWh, PBP and IRR) was also evaluated for each plant considered. These economic parameters are positively affected by an increase in $COT_{1\&2}$. In particular, for the CC derived from 320 MW SPP, when a COT_1 of 1450 °C and COT_2 of 1360 °C are considered, the cost per kWh decreases (with respect to the case when both $COT_{1\&2}$ are at 1300 °C) by about 1.2 %, the PBP decreases by about 4.2 %, while the IRR increases by about 3.2 %.

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