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REPOWERING OF A RANKINE CYCLE POWER PLANT BY MEANS OF CONCENTRATING SOLAR COLLECTORS

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ABSTRACT

Repowering of an existing steam power plant by means of solar concentrating collectors is examined as a viable option to decrease CO2 emissions and increase electric power production during daytime peak load energy demanding hours. In the proposed scheme, the regenerative bleeds from the steam turbine are partially intercepted while boiler feed-water is heated by means of parabolic solar trough concentrating collectors. It is considered that fossil boiler, steam turbine and condenser are the same of the original plant, without modifications.

After choosing a steam cycle reproducing an existing power plant, the scheme of solar repowering is examined and the efficiency of conversion of the solar energy is optimized in relation to the characteristics of the solar collector. The paper shows that the modified scheme produces very little effects on the working conditions of the existing components, either at full load or partial load, and does not reduce the conversion efficiency of the fossil fuel.

In comparison with solar thermal power plants with heat storage and only solar energy as thermal input, the proposed scheme is expected to have comparable efficiency but lower costs per kWh produced, as a consequence of the fact that there is no need for steam turbine, condenser, cooling tower and auxiliary boiler. Moreover it is expected that personnel and maintenance costs will be lower.

NOMENCLATURE

A	= area of solar collectors
c_p	= specific heat at constant pressure
e_x	= exergy per mass unit
$\dot{E}_x$	= exergy flux
I	= direct normal irradiation (DNI)
$\dot{m}$	= mass flow rate
q	= heat per mass unit
$\dot{Q}$	= heat flux
T	= temperature
w	= work per mass unit
$\dot{W}$	= power output

Acronyms

DNI	=direct normal irradiance
EOC	= Exergy Output Coefficient

Greeks

η	= efficiency

Subscripts

c	= collector
f	= oil flowing in the solar collectors
g	= global
0	= ambient condition
t	= thermal
w	= water

INTRODUCTION

The commitments of European Union (EU) for complying with the goals of increasing production of electricity from renewable sources, will lead the EU governments to promote the search for technical solutions of renewable power generation that complement the current system of power generation based on large power plants fuelled by fossil fuels. In the last years, solar thermal power systems have had new impetus with research and demonstration projects, both in Italy and other European countries. The solar thermal schemes that are currently proposed by researchers and private companies [1-2] are mainly two: (a) small steam cycle power plants with concentrating collectors, thermal storage tanks and emergency boiler; (b) solar installations integrating a combined cycle gas turbine. The first type shows high costs and technical concerns with heat storage and, additionally, low LHV efficiency when the conventional boiler is operated. The second plant type shows decayed performance of the bottoming steam cycle when integrated with the solar system, due to the higher mass flow rate flowing through the steam turbine.

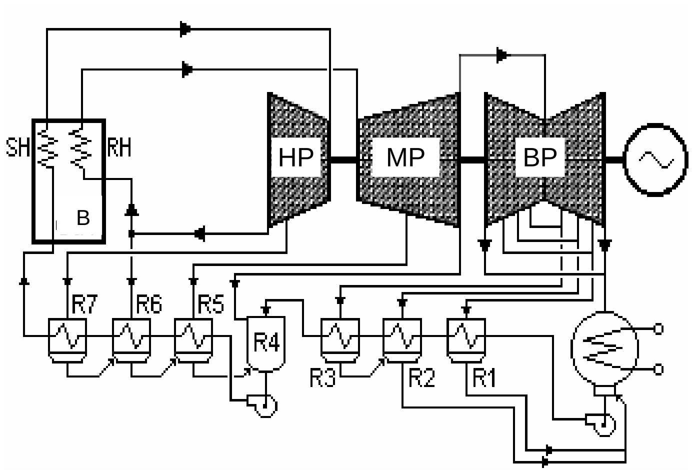

Figure 1 – Examined Steam plant

The present work relies on the consideration that there are large conventional coal-fuelled steam cycle power plants that are expected to continue to be built and operated in the coming years due to the low cost of coal in comparison with other fossil fuels. Electric power utility companies may have interest to integrate their fossil fuelled power plant with renewable energy sources in order to decrease CO2 emissions per MWh and meet regulations concerning the percentage of energy produced from renewable sources.

The scheme proposed here is basically a feed-water repowering scheme of an existing steam plant by partially intercepting regenerative bleeds from the steam turbine. Feed-water is then partially or totally preheated by the heat produced by solar collectors instead of being heated by the regenerative bleeds.

This solution has several advantages with respect to the above mentioned conventional solar thermal steam plants:

1) solar energy is not used for generating steam to be expanded in turbine but only hot water that is produced in simple counter-current heat exchangers: this arrangement leads to higher thermal efficiency of the solar collectors due to the lower temperature of the working fluid in the receiver;
2) the irreversibility losses in heat transfer from fluid operating in solar collectors to liquid water circulating in the steam plant, can be minimized by maintaining a low temperature difference between the two fluids, along all the heat exchanger. In conventional solar steam power plant, instead, the fluid operating in the solar collector, is used for vaporizing and superheating: in this case the temperature difference between hot and cold fluid is necessarily variable along the exchanger and, as a consequence, produces relevant exergy losses.
3) heat storage is not necessary and, in the absence of solar radiation, the plant continues to produce power at the original LHV efficiency without detrimental effects;
4) when the solar system is running, the conversion efficiency of primary energy of fossil fuel remains unchanged;
5) the effects of the proposed modifications are similar to those of the well-known interventions of feed-water repowering operated with a gas turbine. The main disadvantage, as it will be discussed later, is that the maximum temperature at which the water can be heated by solar energy is a little lower than the value that can be reached in plants with solar concentrating systems only.

The advantages of the use of solar heating in regenerative steam plants was already examined by Ying and Hu [3] but they did not consider that solar collectors efficiency is strongly influenced by the temperature at which solar energy is supplied.

The present paper after examining the solar collector efficiency, from first and second law points of view, analyzes the effects of the proposed solution on the performance of an existing 320MW steam plant. Simulations have been conducted by means of the Gate Cycle® code [4], in order to obtain accurate descriptions of the behavior of the various components when they are operated at off-design in consequence of the solar repowering intervention. The efficiency of solar energy conversion, expressed as ratio of the surplus power produced thanks to the solar input divided by the total solar radiation collected by the panels, is examined.

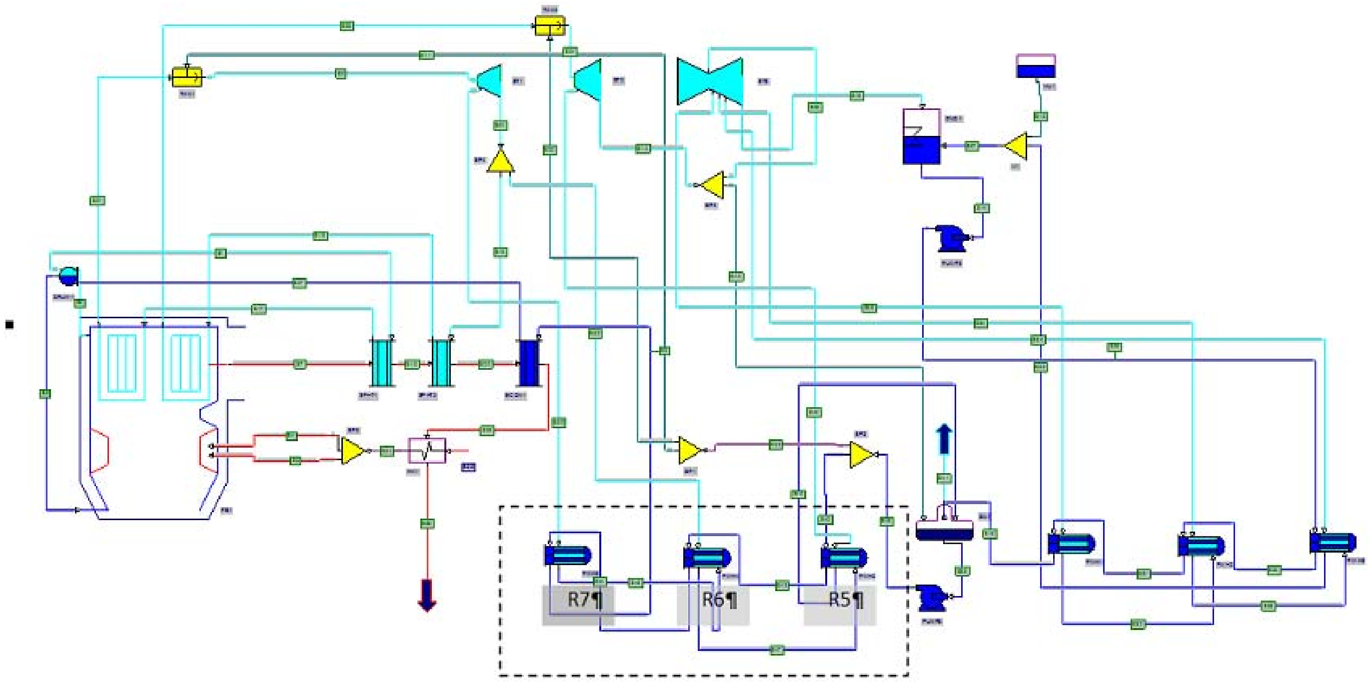

Figure 2 - Plant lay-out of the existing steam plant simulated within the Gate-Cycle environment. The dotted line rectangle indicates the bypassed feed-waters

THERMODYNAMIC CYCLE OF THE EXISTING STEAM PLANT

Since the work presented here aims at repowering of existing plants, it was decided to choose a power facility that well represents what is the situation today in the Italian park of steam plants. Among the various types of plants still in operation, a simple and widespread scheme is the unified plant with a nominal power of 320 MW operated in Italy by ENEL. Its main characteristics are given in Table 1. Although it was possible to gather detailed data on characteristics of this plant, for the purposes of this work, we preferred to perform design calculations in order to have all the necessary data for the subsequent calculations of the off-design performance of the steam plant after repowering by means of the solar plant.

Figure 1 shows the scheme of the steam power plant. We note the presence of three turbine groups, high, medium and low pressure, and seven regenerative feed-water heaters, distributed as follows:

- a group of three surface heat exchangers where low pressure water is preheated before entering the deaerator;
- an intermediate open regenerator that serves also as deaerator to remove non-condensable gases from the feedwater;
- a group of three regenerators where feed water, after being pressurized to the maximum cycle pressure, is heated before entering the economizer.

Figure 2 shows the scheme developed for simulation within the Gate-Cycle visual interface environment. Particular care was devoted to designing superheater and reheater that were split in radiation and convection units, in order to maintain about constant, at partial load, the steam temperature at the turbine inlet [7]. As far as concerning the regenerative feed water heaters, steam bleeds were taken at pressure levels so that enthalpy increase of feed water was about equally split up among all the heaters, in order to maximize the overall thermal efficiency of the steam plant. The design of the regenerative exchangers is based on the method of TTD (Terminal Temperature Difference), that is the difference between the saturation temperature of steam and temperature of the water leaving the regenerator. For all the regenerators TTD has been set at 5 °C which represents a good compromise between efficiency and costs for the exchanger surface.

According to this design procedure, temperature at exit of the feed water heaters were obtained as given in Table 2. These temperatures will be used later for designing the scheme of the solar feedwater heating system. The design performance resulted from simulation under the above assumptions are given in Table 3.

CHARACTERISTICS OF THE ADOPTED SOLAR CONCENTRATING COLLECTOR

The solar plant was designed by adopting the solar EuroTrough collector model ET150 with a length of 150 m and opening of 5.76 m. The collectors are fed with mineral oil

Table 2 – FWH water exit temperature

Feedwater Number	1	2	3	DA	4	5	6
Water exit temp.(°C)	63	86	122	165	198	240	286

***Table 3**. Results of the simulation for the original steam plant*

Electric power output	319.9 MW
Thermal load (LHV)	730.1 MW
Overall net electrical efficiency	43.8%

that is suitable to reach high temperature (up to 400°C) and allows for off working time period (during night time, e.g.) when the temperature can drop to very low values without causing problems at restart. Main characteristics and performance obtained during experimental tests can be found in [8] while, for performance evaluation, several simulation codes, developed in the last years, are able to take into account accurately optical and thermal losses. However, for the purposes of this work, the efficiency of the EuroTrough ET150 collectors was evaluated by means of the following empirical formula [6] that takes into account the direct normal irradiance (DNI) I and difference of temperature, ΔT, between the mean temperature T_{mean} of the oil flowing in the collector and the ambient temperature T_0

$$\eta_c=\frac{\dot{Q}_c}{A_c \times I} = 0.75 - 4.5\times10^{-5}\Delta T - 3.9\times10^{-2}\,\Delta T/\mathrm{I} - 3.0\times10^{-4}\,\Delta T^2/\mathrm{I} \qquad (1)$$

with ΔT in [K] and I in [W/m^2]. The above expression gives results in reasonable agreement with the experimental and simulation results given in [9]. It can be seen from the above expression, that the collector efficiency decreases with increasing T_{mean}, due to the increase of heat losses from the receiver to the ambient.

EXERGY OUTPUT OF THE SOLAR FIELD

For designing the solar field it is necessary to identify how many and which of the feed water heaters should be bypassed when water is heated by solar energy. A second law analysis of the exergy efficiency of the solar field may be helpful to choose the best feedwater configuration, since it is able to evaluate the quality of the heat produced by solar plant, in terms of the maximum work (or potential work) available from the stream of the heat transfer fluid circulating in the solar collector. The maximum work $\mathrm{d}w_{\max}$ the heat $\mathrm{d}q$ at the temperature T can generate, or the availability (exergy) of the heat is

$$\mathrm{d}e_x = \mathrm{d}w_{\max} = \mathrm{d}q(1-\frac{T_0}{T})\,, \qquad (2)$$

where T_0 is the absolute ambient temperature. If the temperature of the heat transfer fluid is increased from T_1 to T_2 in the solar collector, the exergy (or “steady flow availability”) delivered to the heat transfer fluid per mass unit is

$$\Delta e_x = \int_1^2 (1-\frac{T_0}{T})\mathrm{d}q = q - T_0\int_1^2 c_p\frac{\mathrm{d}T}{T} = = q - T_0 c_p \ln(T_2/T_1) \qquad (3)$$

where q is the heat supplied to mass unit of fluid and c_p is specific heat (supposed constant). Taking into account that q is equal to the enthalpy change of the fluid

$$q = c_p(T_2 - T_1) \qquad (4)$$

the exergy change can be related to the heat supplied by

$$\Delta e_x = q(1-\frac{T_0}{(T_2-T_1)/\ln(T_2/T_1)}) = q(1-\frac{T_0}{T_{\text{log}-\text{mean}}})\,. \qquad (5)$$

The log-mean value of the temperature of the fluid can be approximated by a simple arithmetical mean,

$$T_{\text{log}-\text{mean}} \cong \frac{T_1+T_2}{2} = T_{\text{mean}} \qquad (6)$$

that is same mean temperature used in the empirical formula of the collector efficiency given in Eq.(1).

If $\dot{m}_{oil}$ is the mass flow rate of oil circulating in the collectors, the ratio between exergy flux $\dot{E}_{x,c}$ and heat flux $\dot{Q}_c$ generated by the solar collectors, is

$$\frac{\dot{E}_{x,c}}{\dot{Q}_c} = \frac{\dot{m}_{oil}\Delta e_x}{\dot{m}_{oil}\,q} = \frac{\Delta e_x}{q} \cong 1-\frac{T_0}{T_{\text{mean}}} \qquad (7)$$

that is the so called “Carnot factor”, evaluated at the mean temperature at which the heat transfer process is performed. The exergy flux $\dot{E}_{x,c}$ represents the maximum (potential) work that could be extracted per time unit from the fluid, in a fully reversible energy conversion process.

In order to maximize the exergy output of the solar collectors, for an assigned collector area A and solar radiation I, let’s examine the Exergy Output Coefficient (EOC)

$$\text{EOC}=\left(\frac{\dot{E}_{x,c}}{A_c\times I}\right) = \frac{\dot{E}_{x,c}}{\dot{Q}_c}\times\frac{\dot{Q}_c}{A_c\times I} \cong \left(1-\frac{T_0}{T_{\text{mean}}}\right)\times\eta_c \qquad (8)$$

representing the ratio of maximum work that can be obtained, $\dot{E}_{x,c}$, to the collected solar energy, $(A_c \times I)$.

It is worth noting that the EOC defined in Eq.(8) cannot be formally considered as exergy efficiency, because $(A_c \times I)$

is not the input exergy rate. However, since the exergy input rate is about 95% of the solar energy flow $(A_c \times I)$, due to the high temperature (5800K) at which the solar radiation is emitted, the EOC can be considered approximately as exergy conversion efficiency of the solar input.

It can be observed that in Eq.(8) the "Carnot coefficient", $(1 - T_0 / T_{mean})$ increases with increasing T_{mean}, while the collector efficiency η_c decreases with increasing T_{mean}. It is possible, therefore, to find a mean temperature T_{mean} that maximizes the exergy output. On the basis of Eqs.(1) and (8), it is possible to evaluate the collector efficiency and the exergy output coefficient, as a function of the mean temperature of the heat transfer fluid, for different values of the direct solar radiation. The results are shown in Figure 3.

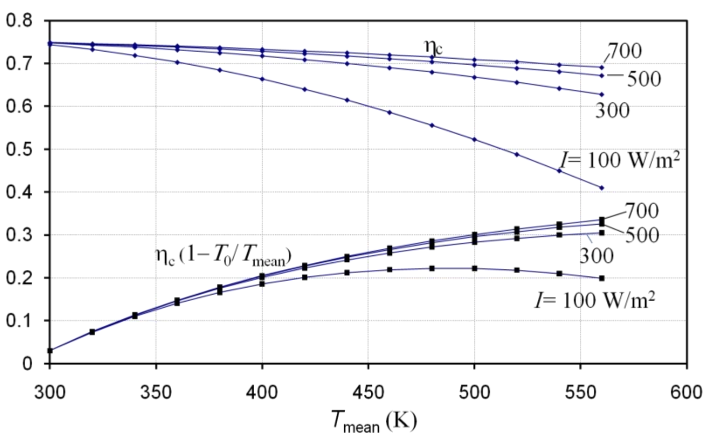

Figure 3 - Collector efficiency and exergy output coefficient (T_0 =290K)

It appears that the collector efficiency decreases slowly with increasing mean temperature when the solar radiation is higher than 300 W/m^2, while under low radiation conditions, the efficiency falls rapidly with increasing T_{mean}. As far as concerning the exergy output, at low irradiation, due to the effects of low collector efficiency, the exergy efficiency shows a maximum at T_{mean} =480 K, while, for higher irradiation levels, the exergy output coefficient always increases with increasing T_{mean} in the observed temperature range.

Since the irradiation is variable during the hours of the day and during the seasons, a statistic optimization procedure is necessary to identify the configuration able to maximize the annual exergy output.

It is worth noting that, the exergy output is not influenced by the difference between inlet and outlet temperature, T_1 and T_2. Increasing the difference (T_2-T_1), while keeping constant T_{mean}, a decrease of the mass flow rate of heat transfer fluid can be obtained, with less work needed to circulate the fluid. However, the temperature T_2 should be less than the maximum allowable for the oil used as heating fluid.

In this work, the chosen configuration was obtained bypassing three feedwater heaters (see Figure 2) that make use of steam extracted from high pressure bleeds of the steam turbines. This choice leads to a good exergy efficiency of the solar collectors at about all the working conditions, while assuring a significant increase of the overall power output thanks to the increased steam mass flow in the steam turbine.

Fig. 4 shows a detail of the repowered lay-out where the three regenerators are partly by-passed by using a heat exchanger fed by oil coming from the solar collectors.

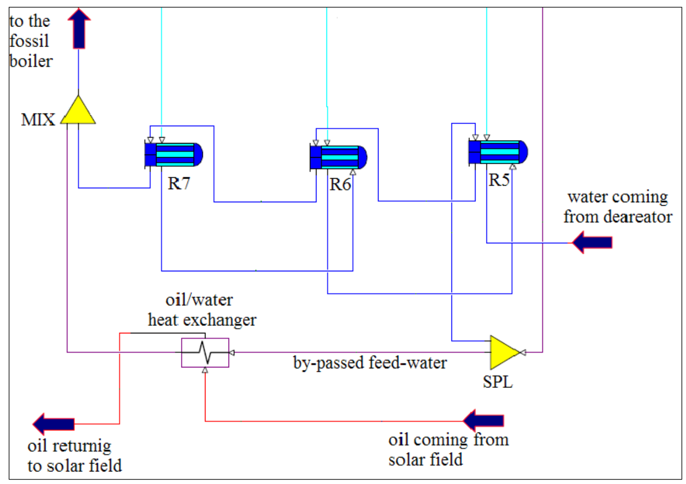

Figure 4 - Detail of the by-passed high pressure regenerators.

DESIGN OF THE SOLAR FIELD

The surface of the solar field was evaluated by assuming that, under the maximum solar irradiation, solar collectors are able to supply 90% of the heat rate originally supplied by by-passed regenerators. Under these hypotheses, even during hours with maximum irradiation, the regenerative heat exchangers are not completely deactivated. The design of the solar field was carried out assuming that, at the design point, the thermal load of the fossil boiler is equal to the thermal load of the original steam plant.

The design of the solar field was carried out for the location of Brindisi (Italy) , latitude 40°37'59" Nord, with a maximum value of direct solar radiation on the horizontal plane equal to 630 W/m^2. The collectors were supposed to be oriented along the East-West direction, and a best tracking technique able to maximize the solar capture was assumed. Under such hypothesis the maximum DNI, in a summer day, on the plane of the collectors was equal to 680 W/m^2.

Inlet and outlet temperature of the heat transfer fluid (oil) circulating in the collectors can be determined from the temperatures at inlet of R5 and outlet of R7, respectively. Assuming that the solar system should be able to supply water at the same temperature of the original regenerative feedwaters, oil temperature at the terminals of the water/oil heat exchanger can be determined after assuming the efficiency of such heat exchanger. The heat transfer surface of this heat exchanger increases with the exchanger efficiency and therefore, increasing the oil temperature profile above the water temperature profile, leads to a significant reduction of the heat transfer surface. Preliminary tests (not given here for the sake of brevity) were carried out in order to reach a good compromise between investment and increase of energy output. From such tests, the design was carried out assuming that oil and water have the same capacity rate

$$c_{p,oil}\,\dot{m}_{oil} = c_{p,w}\,\dot{m}_w \tag{9}$$

in order to maintain the difference of temperature between hot and cold fluids constant along all the heat exchanger. Oil is delivered from the solar collector exit to the oil/water heat exchanger at the maximum temperature of 347°C (620K) whereas it comes back to the collector inlet at a temperature of 200°C. The corresponding collector efficiency was 69.8% while the exergy output coefficient was 32.0% . After assuming the desired heat flow $\dot{Q}_c$ supplied by solar collectors and evaluating the collector efficiency η_c , the collecting area A_c can be evaluated from

$$A_c = \frac{\dot{Q}_c}{\eta_c \times I}, \tag{10}$$

where I the value of DNI at the design point for the selected location of the power plant.

The performance of the repowered plant was evaluated making use of the simulation code Gate-Cycle ®, after assuming that all the components of the original plant work under "off-design" conditions, in order to take into account the effects of the modifications of the working conditions on the performance of each component. Since solar collectors are not included in the component library of the code "Gate-cycle" , it was necessary to make use of home-made macros in order to model the solar collector and to evaluate its efficiency, flow rate of heat transfer fluid and collecting area.

Table 2 gives the main data concerning the design-point performance of the steam plant repowered by the solar field. The electric power output is increased by 43.8 MW from 320 to 363.8 MW (about 14% more) thanks to the increased steam flow in MP and LP turbine. The thermal power supplied by the solar field is 129 MW, obtained by means of a surface collecting area of 294700 m^2. Considering the necessary distance between the lines of collectors, the overall ground surface is about 600000 m^2.

The steam flow rates in evaporator and superheater remain about unchanged as well as in the re-heater, due to the hypothesis that thermal load in the fossil boiler is unchanged. Consequently, the flow rates in HP and MP turbines remain about unchanged: actually there is only a little decrease of flow rate in HP turbine and a little increase in the MP turbine, due to the variation of the steam bleed at the exit of the HP turbine. There are also modifications in operating conditions of condenser and not-bypassed feedwater heaters, due to the increased circulating water. For the condenser, it was assumed that the cooling water flow rate remains the same of the original steam plant: as a consequence, the pressure of the condenser becomes higher in order to account for the increased heat rate.

As already told above, at the design point, the thermal load of the fossil boiler in the repowered plant is unchanged with respect to the original plant. Then, the energy conversion efficiency $\eta_{t,sol}$ of the thermal energy produced by the solar collectors can be evaluated as the ratio between the extra electric power output

$$\Delta\dot{W}_{out} = \dot{W}_{\text{repowered plant}} - \dot{W}_{\text{original plant}}, \tag{11}$$

and the thermal power $\dot{Q}_c$ produced by the solar field

$$\eta_{t,sol.} = \frac{\Delta\dot{W}_{out}}{\dot{Q}_c} = \frac{\Delta\dot{W}_{out}}{\eta_c \times A_c \times I}. \tag{12}$$

At the design point, $\eta_{t,sol.}$ is equal 33.9 % that is an appreciable value, taking into account the relatively low temperature at which the heat flux $\dot{Q}_c$ is supplied to the plant. The overall (global) solar energy efficiency,

$$\eta_{g,sol.} = \frac{\Delta\dot{W}_{out}}{A_c \times I} = \eta_c \times \eta_{t,sol.} \tag{13}$$

takes into account the overall process of conversion of the solar energy from collecting the solar radiation to the electric power. At the design point, $\eta_{g,sol.}$ = 23.5%.

This result can be compared with the exergy output coefficient of the collector, given by Eq.(8): it can be recognized that

$$\frac{\Delta\dot{W}_{out}}{\dot{E}_{x,c}} = \frac{\eta_{g,sol}}{(EOC)_c} \tag{14}$$

is the ratio of the actual additional power output and the ideal maximum work that can be obtained from the heat transfer fluid. This ratio, at the design point is equal to 23.5/32.0=0.73. The exergy losses, responsible for the lower power output with respect to the maximum, can be found mainly in

a) exergy losses in the heat exchanger between oil and feedwater;
b) exergy losses in expansion of steam not used for feedwater heating ;
c) additional losses in the steam condenser (it should be taken in mind that, in the condenser, heat is rejected at 306K (the saturation pressure is assumed equal to 5 kPa) while T_0=290 K)

On the other hand, exergy losses due to irreversibilities in the original feedwaters that are by-passed through the use of solar energy, are avoided. In order to quantify such exergy recovery, it can be observed that, in a feedwater heat exchanger, neglecting desuperheating process, heat is transferred from condensing steam at the saturation temperature T_{sat}, to liquid water at increasing temperature that can reach, at maximum, the saturation temperature T_{sat} . In this case, lost work W_{lost} in the original feedwater, due to the irreversibilities in the heat transfer process is related to heat exchanged Q by

$$W_{lost} = Q\left(\frac{T_0}{T_{ml}} - \frac{T_0}{T_{sat}}\right), \qquad (15)$$

where T_{ml} is the logarithmic mean temperature between min and max temperature of liquid water. Such W_{lost} is avoided in the repowered plant, so contributing to the extra power $\Delta\dot{W}_{out}$ produced by the plant.

***Table 4**- Performance of the repowered steam plant under maximum solar radiation*

Net electric power output	363.8 MW
Overall LHV fuel efficiency	49.8%
Thermal energy conversion efficiency of the solar energy, $\eta_{t,sol}$	33.9 %
Solar radiation conversion efficiency, $\eta_{g,sol}$	23.5%

The overall fuel efficiency, expressed as a ratio between the electric power output and the LHV fuel heat rate is increased from 43.8% to about 50%. This index, even if it does not take into account the heat input supplied by the solar plant, is useful for economical analysis of the costs of energy production, as well as, for evaluating CO2 emissions. It appears that, thanks to the aid of solar energy, the plant fuel efficiency is increased to values not far from levels that can be reached only in gas-steam combined cycle plant.

In order to evaluate the contribution of solar energy on annual basis, the performance of the plant under different values of solar radiation were examined by means of the simulation of the whole plant under off-design working conditions, with various values of direct normal irradiance (DNI) and electric load. On the basis of such results and, known the data of solar radiation in the chosen location (Brindisi, Italy), the primary energy saving was estimated about equal to 16000 TOE per year.

CONCLUSIONS

In the proposed scheme, at the maximum solar irradiance, the reduction of fuel consumption per MWh is about 8%. The solution is therefore technologically feasible to reduce consumption of fossil fuels, as well as profitable taking into account that the peak production may occur in central hours of the day and during summer when the selling price of electricity is higher.

The repowering of a steam cycle power plant using solar concentrating collectors appears potentially competitive with the other solutions since it maintains high the conversion efficiency of the solar energy for the following reasons: lower operating temperature of the fluid used in the solar collectors, leading to fewer thermal losses; high efficiency of steam cycle, as it is optimized for large power plants; lack of irreversibility losses in the thermal storage systems. The thermodynamic analysis has highlighted that energy conversion process of the solar energy presents three kinds of exergy losses (see points a) to c) in the preceding paragraph) that are not avoidable in the proposed scheme as well as in a conventional steam plant with only solar energy as thermal input. The proposed scheme, instead, allows that those exergy losses that, in the original steam plant, were caused by heat exchange in the bypassed feedwaters are avoided in the repowered scheme, so contributing to increasing power output.

Finally, it is to be considered that the costs of the plant are expected also to be competitive thanks to the reuse of many components of the existing fossil fuelled steam plant.

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