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Thermo economic Analysis of Different Configurations of combined Cycle Coupled with a Parabolic Trough Solar Plant

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ABSTRACT
The parabolic trough technology is currently one of the most widespread solar thermal systems forelectricity production. This paper is a thermo-economic study of an Integrated Solar Combined Cycle (ISCC) with Direct Steam Generation (DSG) wherein the solar field is part of the economizer of the heat recovery steam generator (HRSG).Two configurations were analyzed: both included two pressure levels without reheater; in the first one the parabolic trough plant was the high pressure economizer and in the second one the low pressure superheater of the HRSG (heat recovery steam generator). A Euro Trough (ET100) concentrator was considered in this study, the working fluid was water with direct steam generation. Evaporation in the absorber was not an issue since the solar plant was the economizer of the HRSG and an approach point greater than 3°C was considered. The main objective was to obtain the optimum design of the different sections of the boiler and the size of the parabolic field. Optimization was achieved using a Genetic Algorithm developed in previous works by the authors with good results. The method was applied here to configurations that included the parabolic trough plant. As a result, a thermo-economically optimum design for the parabolic trough plant as a section of the HRSG was obtained. The results showed that the solar field increased the power and efficiency of the combined-cycle plant during the operation and made it less susceptible to climate conditions.
Keywords: Parabolic trough, solar plant, thermo economic study, integrated combined cycle solar plant

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NOMENCLATURE

A_A	Solar collector area (m^2)
A	Heat transfer area (m^2)
B	Cash flow (US dollar /year)
C	Concentration ratio
C	Cost
$C_{a\text{-}inv}$	Annual amortization cost (US dollar /year)
C_{kWh}	Generation cost (US dollar /kWh)
C_p	Constant pressure specific heat (kJ/kg K)
Ec	Economizer
Ev	Evaporator
Sh	Superheater
f	Optimization function
F'	Absorber efficiency factor
F_R	Heat removal factor
G_b	Direct radiation (W/m^2)
h	Enthalpy (kJ/kg)
h	Annual operation hours
HRSG	Heat recovery steam generator
I_{Tot}	Total income
m_s	Steam mass flow (kg/s)
N_m	Number of parabolic trough modules of each row
N_r	Number of rows of the parabolic trough
PTC	Parabolic trough collector
Sh	Super heater
ST	Steam turbine
T	Temperature (K)
U	Global heat transfer coefficient ($Wm^{-2}K^{-1}$)
$\overline{W}$	Mean annual power (kW)
W_{GT}	Gas turbine power (MW)
W_{ST}	Steam turbine power (MW)
X	Moisture content

Subindex

a/A	ambient/Absorber
E	Exterior
Ec	Economizer
Ev	Evaporator
Exh	Exhaust
f	Fuel
fitness	Fitness
HRSG	Heat recovery steam generator
I	Interior
Inl	Inlet conditions
Out	Outlet conditions
om	Operation and maintenance
Pen	Penalization
R	Reflector

Tot	Total
GT	Gas turbine
ST	Steam Turbine
u	Useful
Greek letters	
α	Absorptivity
ε	Emissivity
ρ	Reflectivity
σ	Stefan-Boltzmann constant
η	Efficiency

I. INTRODUCTION

Parabolic troughs are currently one of the best proven solar thermoelectric technologies, and the one having demonstrated long-term business development. This is due to its short implementation time and long operation period (over 30 years). Currently there are around 30 plants in operation and more than 1220 MWe installed, which corresponds to 96% of all of all systems installedCSP(Concentrating Solar Power)[1,2].

There isample research on the combination of a parabolic trough solar plant with other technologies,likeLentz and Almanza [3] which combines a parabolic trough plant with a geothermal one. On the specific combination of combined cycle and parabolic trough plant there are works like Montes *et al*. [4] and Nezammahalleh *et al.* [5]. There are somethermal power plants are under construction with the ISCC scheme in Egypt and Algeria [6] aiming at showing the large-scale viability of this technology.

Nezammahalleh *et al*. [5] highlights the advantages of ISCC with DSG (Direct Steam Generation)when solar energy is used to supplement the energy produced by the gas turbine. This leads to a better exploitation of the energy, increases the generation of power in the steam turbine, and compensates for the power decrease of the gas turbine in certain environmental conditions. Most papers consider the solar field as the economizer of the steam generator [4] while in other cases, the parabolic trough plant produces all the thermal energy for the steam cycle and the boiler acts as an auxiliary energy system. Others propose that the solar field be the economizer and the superheater of the plant, while the boiler acts as the evaporator[5].Works like Zarza [7] show that DSG is feasible within different pressure ranges

Among the thermal analyses of the solar plant, Bakos *et al.*[8] have shown the variation of the parabolic trough collector efficiency as a function of the heat transfer fluid; Montes [9,10] compares the direct steam generation with other work fluids like *Therminol VP-1* and shows that DSG presents higher energetic and exergetic efficiency because there is no need for a heat exchanger, thus considerably lowering thermal losses.Finally, works like Tyeagi *et al.*[11] are concerned with second law analyses of this kind of systems. However, there are few works related with the thermo-economic analysis and optimization of the systems.

Considering the above, the objective of this paper was to thermo-economically optimize an Integrated Solar Combined Cycle (ISCC) power plant with Direct Steam Generation (DSG), and particularly the design parameters of the heat recovery steam generator, including the size of the solar field. DSG was retained because of the aforementioned benefits of the solar field as part of the economizer of the heat recovery steam generator (HRSG), so there would not be a two-phase flow in the receiver of the parabolic trough.

In previous works, authors of this paper developed a Genetic Algorithm thermo-economic optimization model applied to the analysis of Combined-Cycle Gas Turbine (CCGT) power plants. This paper proposes the application of the same methodology [12, 13] to the thermo-economic optimization of Integrated Solar Combined Cycles (ISCC).

This paper is divided into 3 sections. The first one describes the configurations analyzed and the design parameters of the ISCC. The second explains the thermal and optimization models. The third presents the results of the optimization and sensitivity analyses.

II. PLANT CONFIGURATION

2.1. Plant Layout

Due to the large quantity of HRSG design parameters that can be taken into account (e.g. number of pressure levels, distribution of economizers, evaporators and superheaters in the HRSG, introduction of reheaters or preheaters), there are many different design configurations for combined-cycle power plants. Nevertheless, in this paper the optimization model was applied to a two pressure level without reheater plant as the one shown in fig. 1. This kind of plant includes for the low pressure level: one economizer, evaporator and superheater and for the high pressure level two economizer one evaporator and one superheater.

The options of the solar field coupling are the following.

A. Two pressure levels without reheater, without solar field (Fig. 1)
B. Two pressure levels without reheater. Solar field as the high pressure level economizer (Fig. 2).
C. Two pressure levels without reheater. Solar field as the low pressure level superheater (Fig. 3).

2.2. Design Parameters

The HRSG thermodynamic design parameters and the solar field size were the independent variables in the optimization problem. As said, the parameters of the gas cycle were excluded of the optimization since a small commercial gas turbine was selected. Its design parameters are shown in Table 1. The design parameters of the steam turbine were also excluded from the independent variables: they were considered fixed values during the simulation of the cycle. These values are also shown in Table 1.

The variation limitsconsidered in the Genetic Algorithm for the design variables are shown in Table 2 where the pinch point (PP) is the difference between the steam temperature at the evaporator entrance and the gas outlet temperature in the same section. This parameter mostly determines the HRSG area and cost. The approach point (AP) is the difference between the steam outlet temperature at the economizer (in this case the solar field) and the saturation temperature at the drum pressure. This parameter is very important and its value is suggested to be greater than 3°C in order to avoid evaporation at the solar trough plant (two-phase flow) [12].

The temperature difference at superheater determines its area;it is the difference between the inlet gas temperature and the outlet steam temperature at the superheater. The optimization of the solar field is made considering the geographic conditions of Cerro Prieto, Baja California, Mexico (Table 3). The solar collector used was the commercial model collector *Eurotrough ET-100*[14]. A North-South orientation of the solar field and multiple arrangements of the solar troughs were considered.

III. OPTIMIZATION MODEL

3.1. Thermodynamic Analysis

a) Solar trough plant

In the thermodynamic model, the efficiency of the parabolic trough is a function of the heat removal factor of the collector (F_R) [15].

$$\eta = F_R\left[\eta_o - U_L\left(\frac{T_i - T_a}{G_b C}\right)\right] \quad (1)$$

where G_b is the direct radiation, U_L is the overall heat loss coefficient, C is the concentration ratio, $\eta_o$is the optical efficiency, T_i is the inlet temperature of the collector and T_a is the ambient temperature. In this equation F_R is defined as a magnitude that relates the actual useful energy gain of the collector to the useful gain if the whole collector surface was at the fluid inlet temperature, and is obtained with the equation:

$$F_R = \frac{mC_p}{A_A U_L}\left[1 - e^{-\left(\frac{A_A U_L F'}{mC_p}\right)}\right] \quad (2)$$

The collector efficiency determined by Eq. 1 was considered to determine energy absorbed at the economizer and superheater sections. This factor determined the area of the solar field during the optimization procedure.The value of U_L in Eq. 2 is obtained considering the following thermal losses

- Heat transfer between the absorber and the fluid [8]
- Conduction heat transfer through the tube wall [9]
- Convection and radiation heat transfer to the glass cover [9]
- Convection and radiation from the glass cover to the atmosphere [9]
- Heat transfer losses through the holders and junctions [9].

b) Combined cycle

To simulate the ISCC, a Visual Basic program was developed which applied the “cash flow and cost” model proposed by Rovira [16]. This model includes a simulation of the gas cycle operating in design conditions applying the model described in Muñoz *et al.*, [17] and Facchini and Stecco [18].

Regarding the HRSG and the steam cycle, the simulation was achieved applying the correlations of the IAPSW (the International Association for the Properties of Water and Steam). The thermodynamic model applied to the Combined Cycle Power Plant(CCPP) was validated comparing the results of simulation with aninstalled plant.More information about the CCPP model and its application can be found in [12,13].

3.2. Description of the Thermoeconomic Model

Based on the optimization model proposed by Duran [16], two optimization criteria were applied:

a) Maximization of the annual cash flow:

$$f(x_j) = B = I_{Tot} - C_{Tot} \quad (3)$$

where I_{tot} is the total income of the generation plant and C_{tot} is the generation cost that includes operation and maintenance costs (of the whole plant including solar field and total fuel costs)as well as amortization cost. Details about this model can be found in [9].

b) Minimization of the generation cost: is the mean annual energy output divided by the total generation cost per year

$$f(x_j) = \frac{\overline{W}.h}{C_{Tot}} = 1/C_{kWh} \quad (4)$$

where $\overline{W}$ is the mean annual output of the plant and *h* is the total working hours per annual operation period. This paper considers 7000 operation hours per year for the whole plant. This period of operation is normally used for CCPP [16].The total cost is a function of the amortization cost as follows:

$$C_{Tot} = C_{a-inv} + C_{om} + C_f \quad \textbf{(5)}$$

C_{a-inv} is the amortization cost and includes the cost of the gas turbine, steam turbine, HRSG and solar field; C_{om} is the operation and maintenance cost and C_f is the fuel cost.

The cost functions considered in the present paper were taken from Duran [19] for the combined cycle and from Montes *et. al.* [9]for the solar plant. The equations that describe the cost model are displayed in Table 4. This paper involves the minimization of generation cost criteria. Optimization by genetic algorithms yields accurate results as shown by Toffolo [20] and Valdés *et al.* [12]. The genetic algorithm optimization model is described in the Appendix.

3.3. Description of the Design and Optimization Program

The Visual Basic optimizationprogram employed for the analysis presented in this paper includes the following modules:

1. Gas turbine simulation: This module simulated the gas turbine cycle in order to calculate its outlets at part and full load. The design of the gas turbine was not part of the optimization model.
2. Simulation of the ISCC: It was used to calculate the operational variables (mass flow, efficiency, moisture content, etc.). This module included all the equations that govern the performance of the different components of the system. It comprised three sub-modules:
   a. Thermal simulation of the solar plant,using the equations for the thermal analysis of the solar plant(Section 3).
   b. Thermal simulation of the HRSG for the different sections of the boiler, considering the values of the variables generated by the genetic algorithm.
   c. Steam turbine simulation. Considers the results of the solar plant and HRSG simulationsto obtain the power and efficiency of the cycle.
3. Optimization of the CCGT power plant: The genetic algorithm optimization tool (described in the Appendix) optimized the cycle. The "fitness" (healthfunction) of each individual was found using the results of the above modules.Fig. 4 shows a schema of the optimization program.

IV. RESULTS

The optimization of the selected configurations focused on the HRSG and the solar field was achieved. In all iterations the same gas turbine design parameters were considered, while the steam turbine variables were obtained during the optimization of the boiler. The optimization results are shown in Table 5. Configuration A is the CCPP without the solar field, conf. B is the one with the solar field coupled in the bottoming cycle and conf. C is the one which the solar field is the LP superheater. As it may be seen in the table configuration B had the lowest generation cost (even though the solar plant in this configuration was bigger and had 24 loops in total) and highest efficiency (This result can be observed more clearly in fig. 5)because the solar field coupled into the high pressure section, made more energy available to the low pressure level. This increased the steam mass flow in this last section and also increased the power generated.

As to configuration C, its steam mass flow increase was lowerthan thatpresented in the configuration whitthe solar field coupled in HRSG high pressure level. This is because the energy transferred in this section was smaller. Both configurations with integrated solar field had greater efficiency and lower generation cost than the configuration without solar field.

With the parameters used here, the percentage of solar energy contributedby each configurationdiffered (Fig. 6); the optimal solar energy contribution for a 2P level configuration was almost 20%when the solar energy was used in the bottoming cycle. This contribution was lower when the solar energy was used in the low pressure superheater.

Finally, a sensitivity analysis as a function of direct radiation for configuration B (Fig. 7) showed that the radiation had a large influence on the steam mass flow.Fig. 8 also shows the generation cost and efficiency variation as a function of the solar direct radiation.

V. CONCLUSIONS

The methodology developed in previous works for the optimization of combined cycle power plants was successfully applied to the optimization of an integrated solar combinedcycle power plant, despite scanty information regarding HRSG cost. Accurate models to predict the cost of this element is necessary to compare its cost to that of a solar trough plant.

Attending to the numerical results, the two ISCC yields were better than the configuration optimized without solar plant. Hence, the combination of systems seemeddesirable, and especially the integration of the solar trough plant at the high pressure level. Also worth noting is the absence of evaporation risk in this section when the solar plant is the economizer of the HRSG, because an approach point larger than 3° degrees was considered. With the solar field coupled into the economizer of the HRSG high pressure level there was more available energy in the low pressure level, leading to an increment in the HRSG efficiency.

Finally, a strong effect of the solar radiation in the generation cost and efficiency of the system was patent.

Further work should apply the optimization method used here to more complex combinedcycle integrated systems, such as two- or three-pressure levels with re-heater, and also integrate the solar field in more than one section of the HRSG.

APPENDIX
DESCRIPTION OF THE OPTIMIZATION METHOD

One of the objectives of this work was to set up a methodology to facilitate the design and optimization of the CCGT. While the general principles of genetic algorithms can be found in Goldberg [18 and 19] and Bentley [20], the algorithm applied here was based on Duran [16], as described below:

1. A population of a certain number of individuals is randomly generated. The individuals are identified by the values of the design variables. In the present paper the population was made up of 1000 individuals and the optimization variables are described in the table 6:
2. All the individuals are evaluated with the fitness function and they are classified according to this value. In this model the fitness function is composed by the total income (Eq. 3) and two penalization functions in the following way:

$$f_{fitness}(x_j) = f(x_j) - P_1 \cdot PenT_{exh-HRSG} - P_2 \cdot PenX \quad (6)$$

where $f_{fitness}(x_j)$ is the objective function defined by Eq. 3,

$P_1 \cdot PenT_{exh-HRSG}$ corresponds to the penalization that discards all the individuals (designs in this case) whose HRSG outlet temperature is less than 100°C:

$$PenT_{exh-HRSG} = \begin{cases} Abs\,(T_{exh-HRSG} - 100) & if\ T_{exh-HRSG} \le 100 \\ 0 & if\ T_{exh-HRSG} > 100 \end{cases} \quad (7)$$

$P_2 \cdot PenX$ corresponds to a penalization that discards all the individuals (designs) whose moisture content (*X*) in the last stage of the steam turbine is below 16%.

$$Pen\,X = \begin{cases} Abs(0.16 - X) & if\ X \ge 0.16 \\ 0 & if\ X < 0.16 \end{cases} \quad (8)$$

3. The healthiest individuals are selected as the parents of the following generation. Genetic operators (mutation and crossover) are applied to these selected individuals and a new generation is obtained. Each generation has the same population size.
4. The new generation is evaluated again with the fitness function. The hypothesis underlying this method is that the new generation is formed by healthier individuals than the previous one.

Finally, the process is repeated until a previously established number of generations is reached.

FIGURES
LIST OF FIGURES

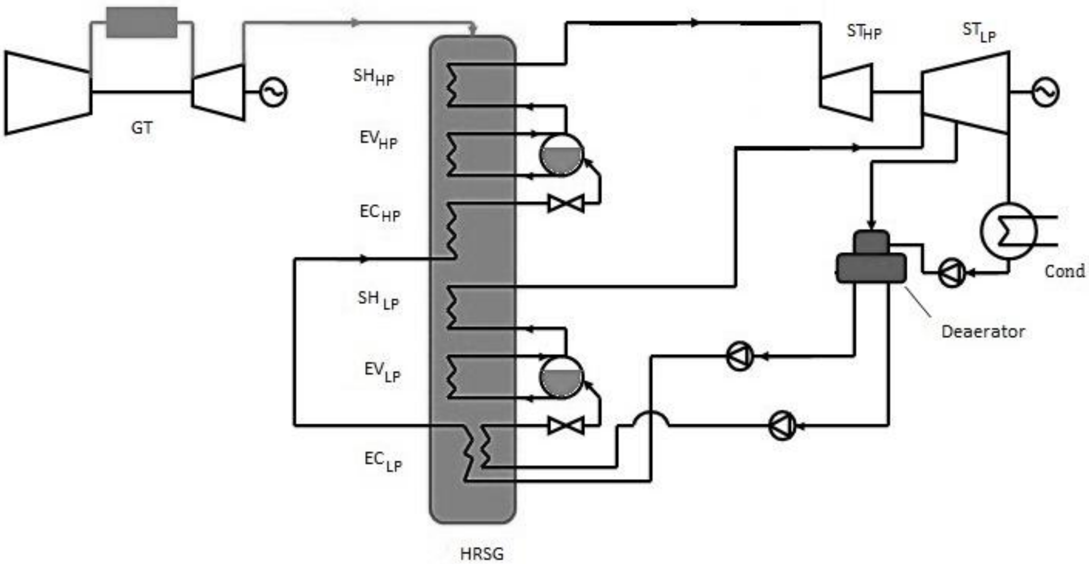

Fig. 1. CCPP with two pressure levels without reheater.

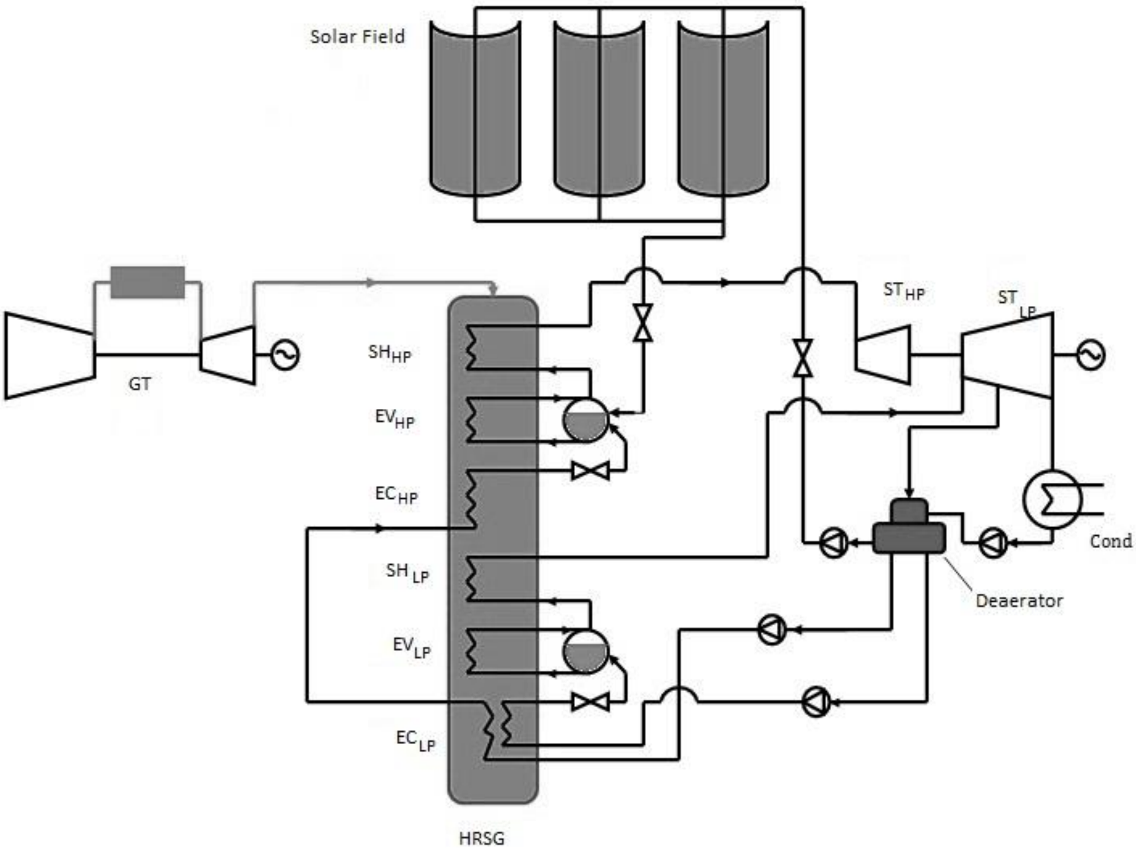

Fig. 2. Configuration A: CCPP with two pressure levels without reheater where the high pressure economizer is the solar field.

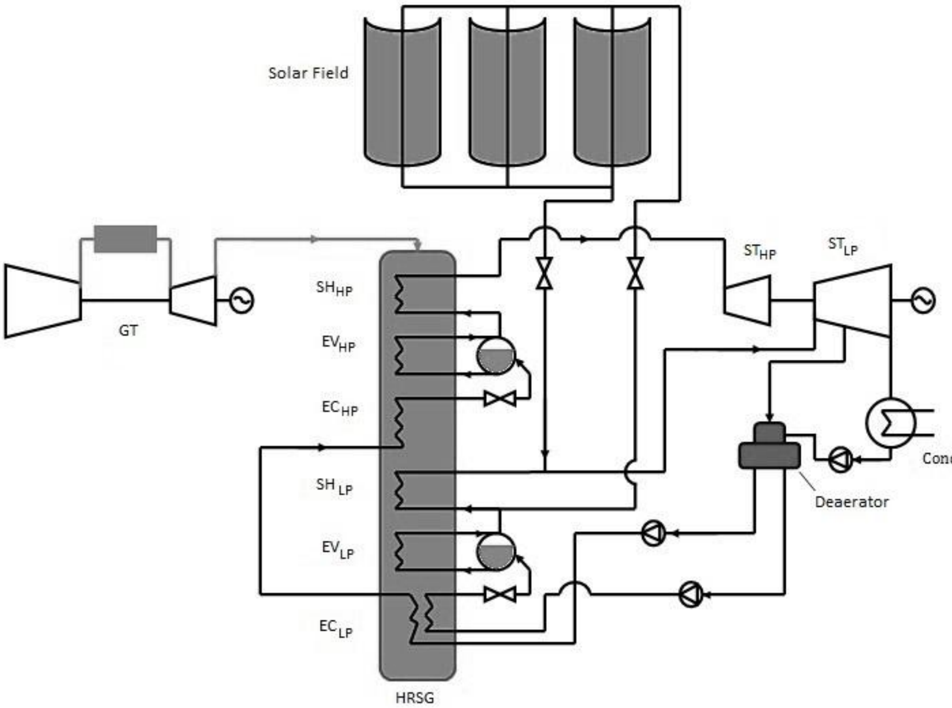

Fig. 3. Configuration B: CCPP with two pressure levels without reheater where the Low pressure superheater is the solar field.

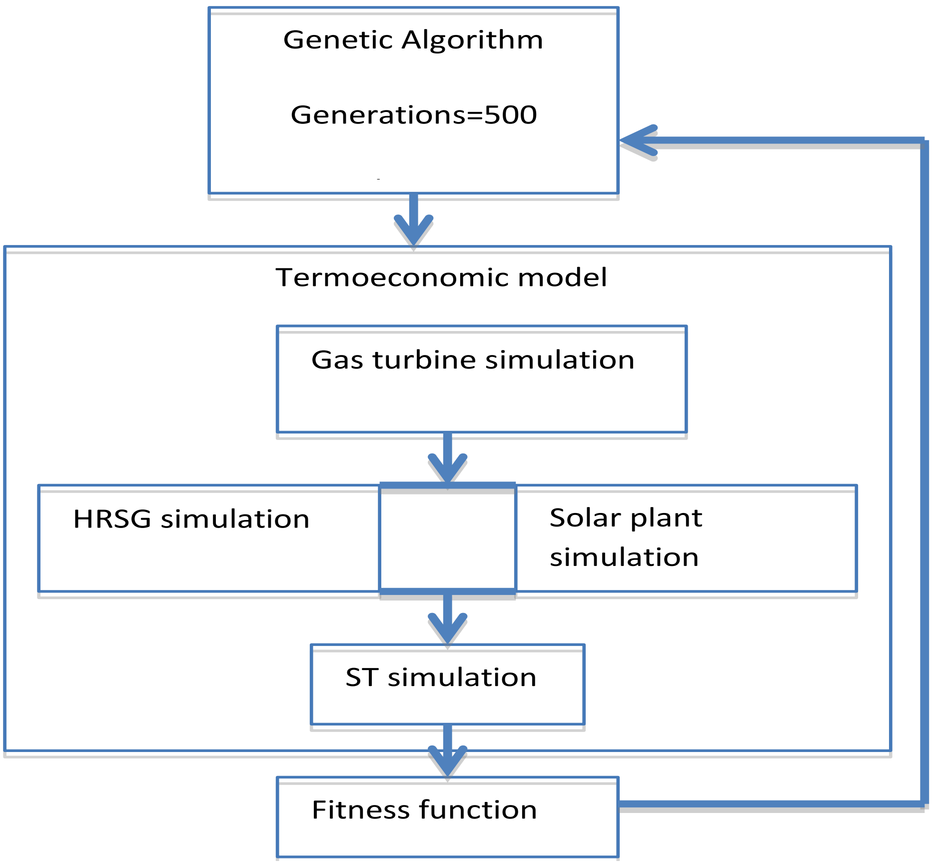

Fig. 4. Schema of the optimization program with all the simulation modules.

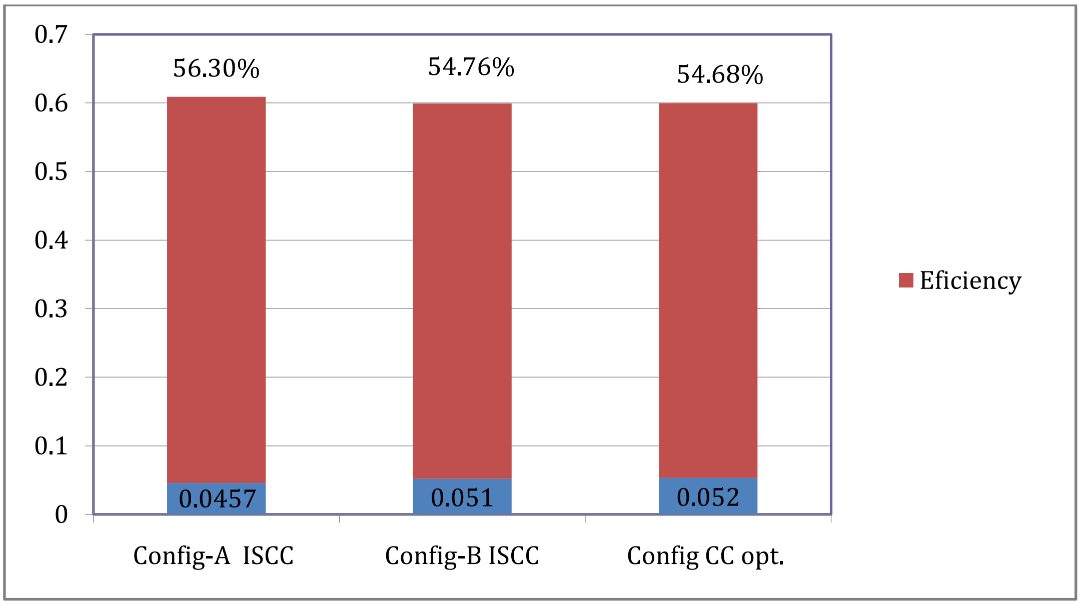

Fig 5. Comparison efficiency and generation cost of the configurations analyzed

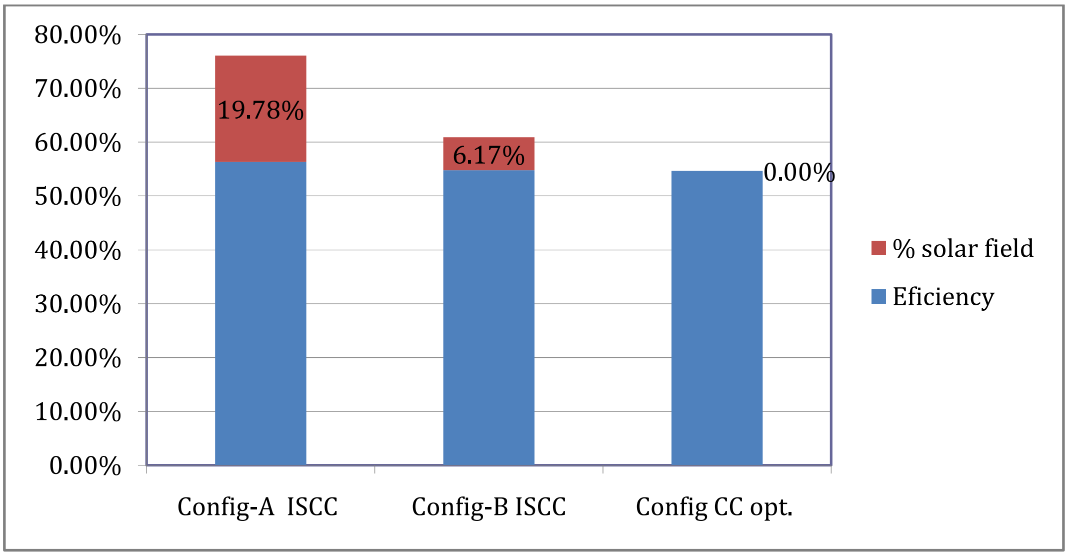

Fig. 6. Optimal energy contribution of the solar field: Efficiency.

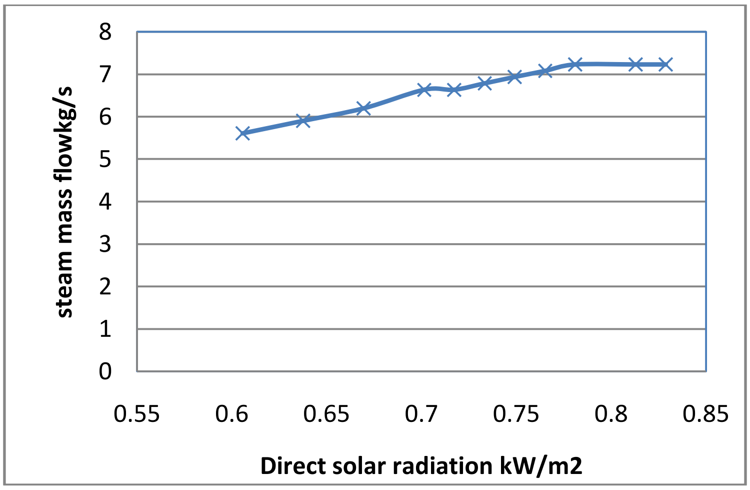

Fig. 7. Variation of the steam mass flow in the parabolic trough as a function of the solar radiation in the optimized configuration A.

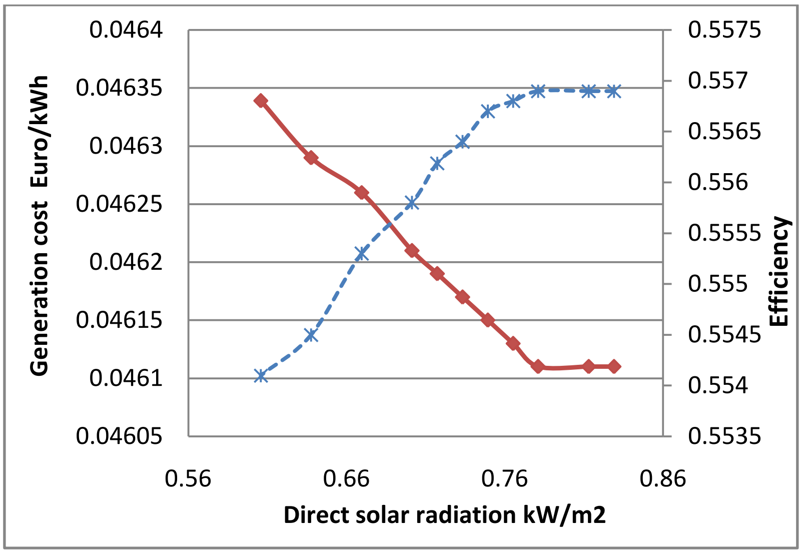

Fig. 8. Variation of the generation cost and efficiency of the ISCC as a function of the solar radiation in the optimized configuration A.

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Table 1. Gas and steam cycle design parameters

Gas cycle design parameters	**Value**
Compression ratio	30
Inlet temperature	1430 K
Gas turbine outlet temperature	710 K
Gas mass flow	120.2 kg/s
Nominal power	38.8 MWe
Steam cycle design parameters	**Value**
Pressure levels	2
Turbine isentropic efficiency	0,85
Condenser pressure	0.045 bar
Deareator pressure	0.2 bar

Table 2. Parameters for the thermoeconomic optimization

Design parameter	**Interval of variation**
Drum Low pressure	3bar-18bar
Low pressure Pinch Point	3°C-20°C
Low pressure Approach Point	3°C-20°C
Difference Temperature at the Low pressure superheater	20°C-85°C
Drum High pressure	50-100bar
High pressure Pinch Point	3°C-20°C
High pressure Approach Point	3°C-20°C
Difference Temperature at the High pressure superheater	20°C-85°C

Table 3. Cerro Prieto, Baja California geographic data

Parameter	Value
Latitude	109,916 ° W
Longitude	23,0833° N
Ambient temperature	35°C
Average global irradiation	790 W/m^2

Table 4. Considerations for the economic model

Gas turbine cost	$C_{TG} = 0.1788\, W_{TG}\,(MW) + 3.0253$
Steam turbine cost	$C_{TV} = 0.115 W_{Tv}\,(MW) + 2.75$
HRSG section cost (€/Wm^2)	$C_{TG} = \sum_{sec} k_i (UA_i)^{0.8}$
Solar plant fixed cost (€/m^2)	200
Field cost (€/m^2)	2
Solar plant operation and maintenance cost (€/m^2)	9

k_i= Is a coefficient of the cost of UA unit in each HRSG section. The description of how this coefficient is obtained may be found in Duran [9] and Valdés [16].Cinv, Com, Cf (Eq. 5)?

Table 5. Results of the optimization on the integrated combined cycle solar plant.

Design parameters	Configurations		
	A	**B**	**Configuration withoutsolar plant**
Drum low pressure (bar)	4.42	3.71	3.2
Low pressure Pinch Point (K)	9.04	4.2	3.01
Low pressure Approach Point (K)	7.84	6.52	4.07
Low pressure temperature difference at superheater (K)	75.3	50.33	84.8
Low pressure steam mass flow (kg/s)	8,91	7,9	5,32
Drum high pressure (bar)	91.37	101.17	66.21
High pressure Pinch Point (K)	7.44	14.72	3.98
High pressure Approach Point (K)	6.16	7.94	7.6
High pressure mass flow (kg/s)	10,05	9.3	10.89
Generation cost (€/kwh)	0.04573	0.0518	0.053
Efficiency	56.3%	54.76%	54.68%
ISCC power (kW)	54772.66	53083.0	52986.82
Total solar plant loops number	24	7	0
Parallel loops number	7	7	0

Table 6. Optimization variables.

Drum pressure	P (bar)
Pinch Point	PP (°C)
Approach Point	AP (°C)
Temperature Difference at superheater	DT (°C)
Steam mass flow	m (kg/s)
Number of parabolic trough modules per row	Nm
Numbers of rows of the solar plant	Nr

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