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A simplified calculation method for the evaluation of the performance of a hybrid solar plant with linear parabolic collectors and Joule-Brayton air cycle

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ABSTRACT

A calculation method for the evaluation of the performance of a 50 MW hybrid solar plant with linear air collectors and an open Joule-Brayton air cycle, equipped with an inter-refrigerated compressor, a regenerator and a natural gas combustor has been developed.
If the plant is operated each day between sunrise and sunset for an entire year in conditions close to nominal ones, the net electricity produced in one year is 244,000 MWh, the fuel consumption is 34,575 tons of natural gas, the annual average efficiency of solar collectors is 49.8%, the average thermodynamic efficiency of power block is 42.1%, the average conversion efficiency of solar energy to electricity is 20.9%.
The saving of fossil fuel due to the employment of solar energy in the hybrid plant is 22%, the cost of electricity produced results to be 8.3 cents$/kWh. This value is competitive with that of other plants using renewable energy.

1. INTRODUCTION

In almost all thermodynamic solar plants provided with cylindrical parabolic collectors, synthetic oils are employed as heat transfer fluids (HTF) in the collectors with water steam evolving in a Rankine cycle. Nine plants were built in the Mojave Desert (California), starting from 1984, with electrical powers ranging between 13.8 and 80 MW [1]. The maximum temperature of the oil had to be limited to 390 °C owing to oil stability problems and this reduces the efficiency of the Rankine cycle. In 2007, the 64 MW Nevada Solar One plant started working in the USA [1], also using synthetic oil and water steam. These plants have the advantages of a tested and consolidated technology, but the disadvantages of a low temperature of the hot oil and a potential danger of possible fires due to the use of oil as a HTF and as a heat storage medium.

Recently, many 50 MW plants have been built in Spain [2]. Among these, are: Andasol 1, Andasol 2, Andasol 3, Arcosol 50, Arenales, Aste 1A, Aste 1B, Astexol II, Casablanca, Extresol-1, Extresol-2, Extresol-3, La Africana, La Dehesa, Manchasol-1, Manchasol-2, Termosol 50, Termosol 1, Termosol 2 and La Florida. In all these plants synthetic oil is used as a heat transfer fluid, molten salts as heat storage medium and water steam working in the Rankine cycle [3-6]. These plants combine the advantages of the tested use of oil as HTF and of the use of molten salts which have a higher heat capacity than oil. Rubbia [7], in the “Archimede Project”, proposed the use of molten salts both as heat transfer fluid and as heat storage medium. In this case, the minimum temperature of the salt cannot be lower than 240°C to avoid the solidification of the salt while the maximum fluid temperature is of about 550°C. A first 5 MW prototype of this plant was built in Sicily, Italy, at Priolo Gargallo [6, 8]. The main problem of such plants is the necessity to heat all the pipes containing the salts continuously to avoid solidification and this involves energy consumptions.

In a previous paper [9], the authors firstly proposed the use of an innovative solar plant operating with cylindrical parabolic collectors and atmospheric air as HTF (this is a completely free and completely safe fluid) evolving in an open Joule-Brayton cycle. In a subsequent paper [10], an improved model of the same plant was presented, while in ref. [11] the authors evaluated the performance of an integrated Brayton-Rankine combined cycle plant.

In the present paper, a simplified model is presented, able to evaluate the technical and economic performance of a 50 MW air hybrid parabolic trough solar plant utilizing both solar energy and natural gas, in order to operate the plant near the design conditions.

2. CALCULATION METHOD

Figure 1 shows the scheme of the plant with linear parabolic collectors and an open Joule-Brayton cycle.

The atmospheric air is compressed into a three-stage, inter-refrigerated compressor, then preheated in the regenerator and sent to the field of parabolic collectors, from which it comes out at a maximum temperature of 600 °C (a limit imposed by the stability of the selective coating of the receiver tube); downstream of the collectors there is a natural gas combustor that raises the fluid temperature to a value of or higher than 800 °C, so as to increase the thermodynamic efficiency of the Brayton cycle; the combustion gases operate a gas turbine connected to the electric generator; before discharging into the atmosphere, the gases pass through the regenerator to preheat the oxidizing air.

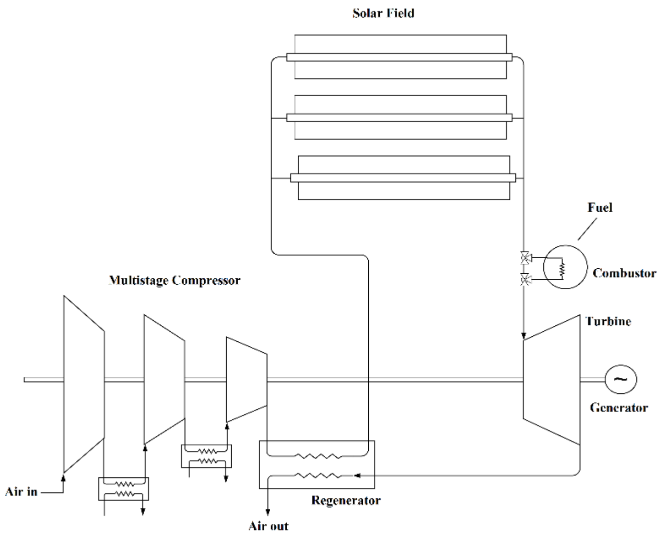

Figure 1. Scheme of the plant

In the operation of the plant, with the change of direct radiation (DNI), it was supposed to vary the fuel flow while maintaining the air flow constant to maintain the temperature of the gas at the turbine inlet at 800 °C.

The operation of the plant components was simulated with the THERMOFLEX [12] code, starting from the design conditions, and then performing some series of calculations in out-of-design conditions, varying the outside air temperature and the useful solar thermal power transferred to the fluid.

By means of the obtained results, some correlations of the thermal power supplied by the fuel as a function of the useful solar thermal power and a correlation of the turbine efficiency as a function of the outside air temperature were developed.

These correlations, together with a correlation of the useful thermal power provided by the parabolic collectors, obtained by the STS collector code [13], were implemented in a Matlab Simulink program called PATHERY (PArabolic TEHRmodynamic Hybrid).

By using the PATHERY program, starting from the direct solar irradiance values, the ambient air temperature and relative humidity of the location of the plant, it is possible to calculate: the hourly values of the electrical power generated by the plant, the hourly and annual values of solar thermal energy and of the thermal energy generated by fuel, the collector field and power block efficiencies, and the electricity produced in one year.

In the next section, the calculation model and correlations will be explained in detail.

3. SIMULATION OF THE SOLAR FIELD

The field of linear parabolic collectors was simulated by means of the STS stationary code, able to carry out the thermal analysis of a linear parabolic collector. The receiver tube of the collector is subdivided into an arbitrary number of axial steps, each described by five nodes, whereas the outside ambient, represented by the mirror, the air and the sky, is described by three nodes at fixed temperature. The experimentally validated code can consider oil, molten salts, carbon dioxide and air as heat transfer fluid. The calculation model implemented in the code and the solution technique of the heat balance equations are explained in detail in the reference [13-14]. Once established the geometric and optical data of the collector, the inlet temperature of the fluid, the ambient air temperature, the mass flow rate and the value of projected direct irradiance, the code calculates the outlet temperature and enthalpy of the fluid, the pressure drop, the useful thermal power to the fluid, and the efficiency of the collector.

The solar power absorbed from the receiver tube of a collector is calculable by the relation:

$$\mathrm{P_{s,abs}} = \mathrm{DNI} \cdot \mathrm{IAM} \cdot \eta_{\mathrm{opt}} \cdot \mathrm{A_{col}} \tag{1}$$

where DNI is the direct normal irradiance, IAM is the incidence angle modifier, η_{opt} is the normal optical efficiency and A_{col} is the area of the collector.

The parameter IAM takes into account the effect of the inclination of the normal direct irradiance on the irradiance projected on the collector and on the decreasing of the optical efficiency with the incidence angle. It is calculated by eq. (2) [15]:

$$\mathrm{IAM} = \cos(\mathrm{i}) - 2.859621 \cdot 10^{-5}\mathrm{i}^2 - 5.25097 \cdot 10^{-4}\mathrm{I} \tag{2}$$

The normal optical efficiency is defined as:

$$\eta_{\mathrm{opt}} = \rho \cdot \tau_{\mathrm{env}} \cdot \alpha_{\mathrm{rec}} \cdot \gamma \tag{3}$$

where ρ is the reflectivity of the parabolic mirror, τenv is the normal transmissivity of the glass cover, αrec is the absorptivity of the receiver tube and γ is the interception factor of the radiation due to the error of the sun tracking apparatus.

The useful thermal power Pu transmitted to the fluid in the collector is calculable by the equation:

$$P_u = m_A \cdot (h_0 - h_i) \tag{4}$$

where m_A is the flow rate, ho the outlet enthalpy and h_i the inlet enthalpy of the fluid.

Owing to the thermal losses due to radiation and convection, the useful power is lower than the absorbed power.

The geometrical and optical data of the collectors are reported in Table 1.

Table 1. Geometrical and optical data of collectors

Parameter	Value
$D_{i,rec}$ (m)	0.084
$D_{o,rec}$ (m)	0.090
$D_{i,env}$ (m)	0.128
$D_{o,env}$ (m)	0.134
a (m)	5.76
L (m)	100
τ_{env}	0.96
α_{env}	0.02
α_{rec}	0.95
ρ	0.94
γ	0.92
η_{opt}	0.788
$\varepsilon = 0.0406344 \quad (1.0022532)^{T}$	

The collectors are oriented with the focal axis along the North-South direction and they are rotated around this axis during the day in order to follow the sun.

By running the STS code in different conditions of the input parameters, such as inlet temperature to the collector, direct irradiance, mass flow rate, ambient air temperature, etc., several correlations of global efficiency of the collector, of outlet temperature of the fluid, of useful thermal power, and of other useful variables were obtained, as a function of the solar parameter x_s, defined as:

$$x_s = DNI \cdot IAM \cdot \eta_{opt} \quad (5)$$

This parameter has the physical meaning of solar energy absorbed at each instant from the receiver by unit area of collector aperture.

The correlations were obtained by the DATAFIT program [16].

The correlation of useful thermal power to the air, obtained for an inlet temperature of 350 °C and a mass flow rate ranging from 0.65 to 1.12 kg/s in a collector 100 m long, and an atmospheric air temperature of 25°C, is the following:

$$P_u = a\,x_s^3 + bx_s^2 + cx_s + d \quad (6)$$

$$a = a_1/(1 + a_2 m)$$

$$a_1 = -4.7140777 \cdot 10^{-7}, \qquad a_2 = -0.0846198$$

$$b = b_1/(1 + b_2 m)$$

$$b_1 = 3.11791 \cdot 10^{-4}, b_2 = -0.364378$$

$$c = c_1 + c_2 m$$

$$c_1 = 0.3594564, \qquad c_2 = -0.0191244$$

$$d = d_1 + d_2 m$$

$$d_1 = -2.1116633, \qquad d_2 = 0.096150322$$

Figure 2 shows the trend of Pu as a function of the solar parameter x_s for a mass flow rate of 1.04 kg/s. Obviously, Pu increases as the values of DNI and x_s increase.

The collector efficiency is defined as the ratio of useful thermal power to the solar power projected onto the collector area A_{col}:

$$\eta_{opt} = P_u/(DNI \cdot A_{col} \cdot \cos(i)) \quad (7)$$

In figure 3 the trend of collector efficiency as a function of the parameter x_s is shown.

The graph shows a point of maximum, since, for lower values of DNI the absorbed solar power by the receiver prevails on the heat losses, with an increase of the efficiency, while, for larger values of DNI, the heat losses instead prevail, with a decreasing of the collector efficiency.

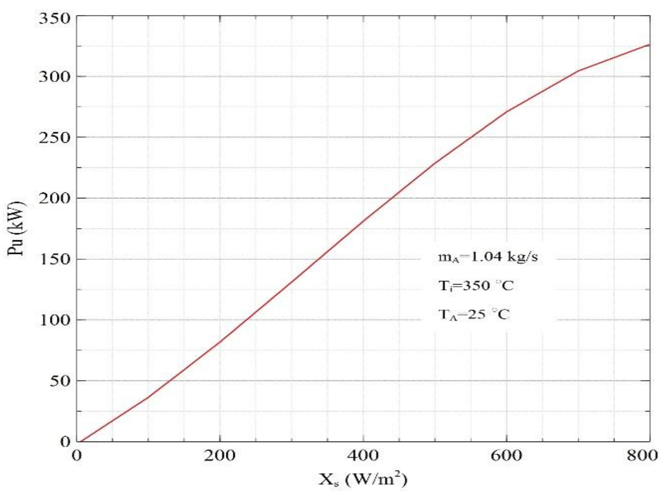

Figure 2. Useful solar power per collector as a function of x_s

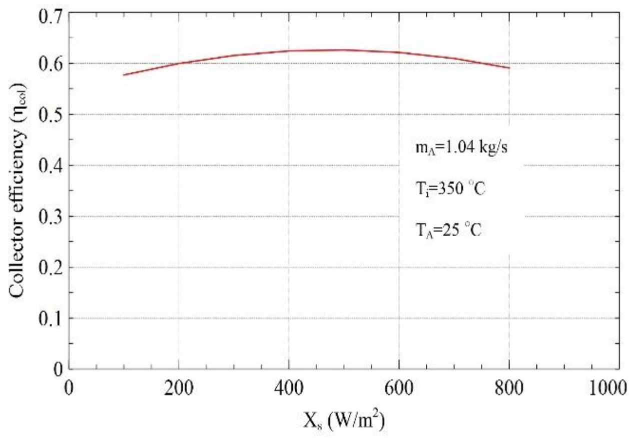

Figure 3. Collector efficiency as a function of solar parameter

4. SIMULATION OF THE WHOLE PLANT

Table 2 shows the plant design data, which is supposed to be located, for the high values of direct solar irradiation (2034 kWh/m^2 in a year) in Almeria, Spain (Lat.36.85 ° N, Long 2.38 ° W).

The main input data are: nominal electrical power of the plant, atmospheric air temperature, design DNI value, atmospheric air temperature, compressor and turbine pressure ratios and inlet gas temperature to the turbine.

The field of solar collectors is modeled as a generic heat generator upstream of the natural gas combustor. Once the type of the plant and the characteristics of the various components are defined, the THERMOFLEX code allows to calculate the thermodynamic coordinates in all plant points, the air flow rate, the thermal power supplied by the collectors to the air, the fossil fuel flow rate, the thermal power generated by the combustion, the gross and net electric power supplied by the gas turbine, the net thermodynamic efficiency of the cycle and many other quantities of interest.

Table 2. Design data of hybrid air joule-brayton solar plant

Nominal Net Electric Power	**50 MW**
Direct Normal Irradiance (DNI)	850 W/m^2
Zenith Angle	13.850^0
Azimuth Angle	-10.713^0
Total Area of Collectors	126,144m^2
Compressor Pressure Ratio (β_c)	9.6
Turbine Pressure Ratio (β_t)	8.5
Net Thermodynamic Efficiency of power block (η_t)	0.421
Atmospheric Air Temperature (T_A)	25^{0}C
Inlet fluid temperature to the gas turbine	800^{0}C
Air mass flow rate	228 kg/s
Fuel mass flow rate	1.2 kg/s

By using the THERMOFLEX code, after determining the above quantities in the design conditions (fixing the outlet air temperature from the solar heat generator at 600 °C and the temperature of the combustion gas at 800 °C), some series of calculations were performed in out-of-design conditions for three atmospheric air temperature values of 0 °C, 25 °C and 50 °C and some values of the useful solar thermal power, always maintaining at 800 °C the input temperature of the combustion gas to the turbine.

Using the results of the calculations, three correlations of the useful thermal power generated by the combustion of natural gas (Qf) were developed for the three external air temperature values, using the DATAFIT program, as a function of the useful solar thermal power (Qs), and a correlation of the net thermodynamic efficiency of the power block as a function of the external air temperature T_A.

The correlations obtained have the following form:

$$for\ T_A = 0\ ^0C, \quad Q_f = a\ + b\ Q_s + c\ Q_s^{0.5} \tag{8}$$

$$a = 134.2850244, \qquad b = -1.05112279$$

$$c = 5.44051789 \cdot 10^{-2}$$

$$for\ T_A = 25\ ^0C, \quad Q_f = a\ + b\ Q_s + c\ Q_s^{3} \tag{9}$$

$$a = 122.81659, \qquad b = -1.05112279$$

$$c = 5.80169199 \cdot 10^{-8}$$

$$for\ T_A = 50\ ^0C, \quad Q_f = a\ + b\ Q_s + c\ Q_s^{0.5} \tag{10}$$

$$a = 112.3064293, \qquad b = -1.03488953$$
$$c = 3.562151 \cdot 10^{-3}$$

$$\eta_t = a\ + b\ T_A^{2.5} \tag{11}$$

$$a = 0.4263534002, \qquad b = -1.882460353 \cdot 10^{-6}$$

The inlet air flow to the plant is related to the outdoor air temperature, in the range of 0-50 °C, by the equation:

$$m_A = -0.0042\ T_A - 1.1313333 \tag{12}$$

When the air temperature at the outlet of the collectors exceeds the value of 600 °C, it was supposed to defocus the collectors in such a way as to decrease the air temperature to a maximum value of 600 °C. In the calculation procedure, this constraint is respected by placing a maximum value on the solar parameter xs. This maximum value is related to the air flow rate m_A by the relation:

$$x_{s,max} = 250.5792 \cdot m_A^{2.4106596} \tag{13}$$

All correlations have been implemented in the PATHERY calculation program, which introduces the weather time data of the location of the plant (external air temperature, relative humidity, DNI, etc.) [17].

The size of the solar field is calculated, in the design conditions by the equation:

$$Q_s = N_{col} P_u \tag{14}$$

where N_{col} is the number of solar collectors and the quantity Qs is obtained from the output data of THERMOFLEX operated under design conditions. The value of Pu was calculated by eq. (6) for the solar parameter design value, being the inlet air temperature in the collectors equal to 347 °C, close to 350 °C.

The net electrical power Pel supplied by the hybrid system can be calculated with the equation:

$$P_{el} = \left(N_{col} P_u + m_f\ NHV\right) \eta_t \tag{15}$$

where m_f is the fuel flow, NHV is the net heating value of the fuel and the other quantities are known.

For a power of 50 MWe, one obtains: Ncol = 219, a total area of the collector field of 126,144 m^2, an air flow rate of 228 kg/s and a natural gas flow rate of 1.2 kg/s.

Using the PATHERY program, the hourly thermodynamic coordinates of the fluid at the inlet and outlet of all components, the useful solar thermal power, the fuel flow rate (NHV = 46.85 MJ/kg), the thermal power generated by combustion, the efficiency of the collectors, the net power of the power block, the overall net thermodynamic efficiency of the plant and the net electrical power have been obtained.

The calculation sequence is as follows: for each outdoor air temperature value, the flow rate per collector, using eq. (12), is calculated, then the useful power for the collector Pu is determined through eq. (6); multiplying Pu by Ncol, one obtains the useful solar thermal power Qs; interpolating eq. (8-10), the thermal power generated by fossil fuel Qf is obtained; finally, by multiplying the total thermal power (Qs + Qf) by the efficiency of the power block, eq. (11), one obtains the electrical power Pel supplied by the plant. Figures 4-7 show, for a clear day, the trends over time of the main quantities that characterize the plant.

In particular, figure 4 shows the trend of DNI as a function of time;

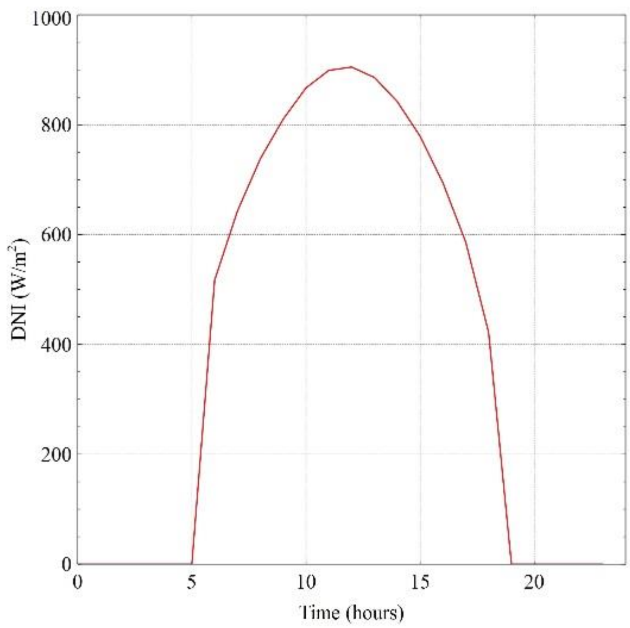

Figure 4. Direct Normal Irradiance as a function of time for a clear day

Figure 5 shows the time trend of useful solar power, thermal power provided by the fossil fuel and total thermal power;

Figure 6 shows the electric power and figure 7 shows the collectors, power block and overall plant efficiencies as a function of time.

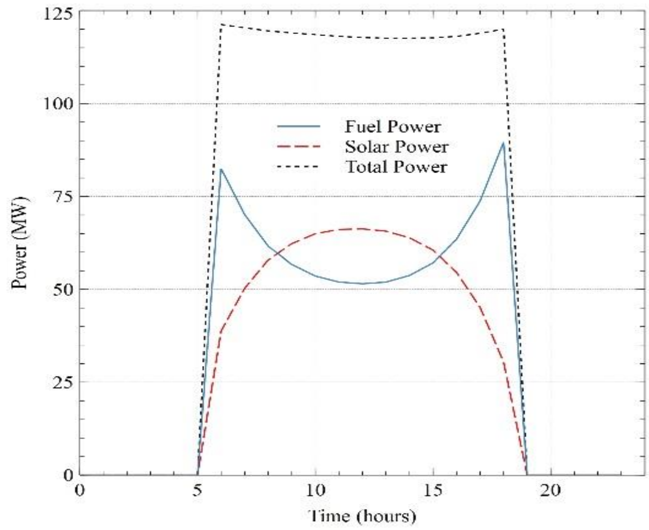

Figure 5. Useful solar power, fuel thermal power and total power as a function of time in a clear day

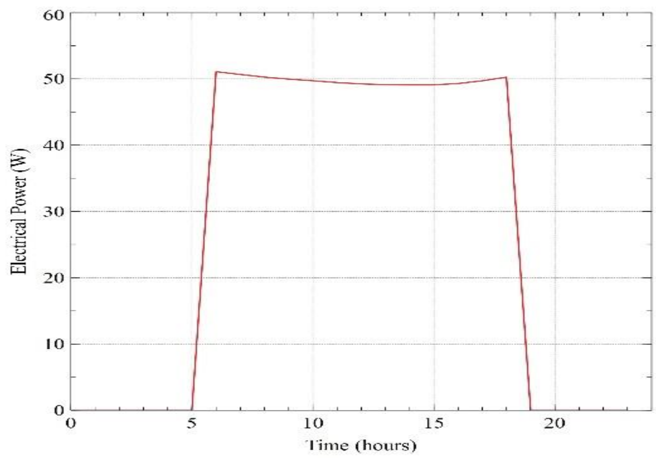

Figure 6. Net electrical power as a function of time in a clear day

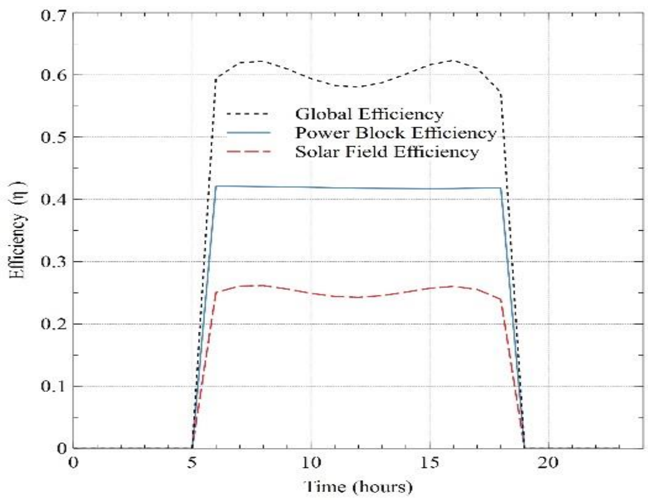

Figure 7. Solar field, Power block and global efficiencies as a function of time in a clear day

It was considered useful to evaluate the performance of the hybrid system over a whole year, exploiting the sun's supply throughout the year and assuming that the plant is switched on every day from sunrise to sunset under nominal power conditions.

By integration of hourly values, the annual values of net electricity Eel (MWhe), of useful solar energy ES (MWht), of thermal energy supplied by combustion Ef (MWht), of Heat Rate (ratio between the thermal energy generated by combustion and the electrical energy produced) of solar fraction FS (ratio of the thermal energy provided by the solar collectors and the total thermal energy supplied by both sun and fossil fuel), of average annual yield collectors and power block collectors were calculated. These values are shown in Table 3, for a solar multiple (SM) varying between 1 and 1.4.

Table 3. Annual data of hybrid Brayton cycle plant

SM	**1**	**1.2**	**1.4**
E_{el} (MWh_e)	244,000	243,705	243,428
E_s (MWh_t)	127,755	140,296	151,799
E_f (MWh_t)	450,914	437,663	425,251
Heat Rate (BTU/kWh)	6293	6128	5962
F_s	0.22	0.24	0.26
Solar Collector efficiency (η_{col})	0.498	0.455	0.422
Power Block Efficiency (η_t)	0.421	0.421	0.421
CF (%)	55	55	55

Solar multiple is defined as the ratio between the thermal power produced by the solar field at the design point and the thermal power required by the power block at nominal conditions [5].

The data in Table 3 shows that, for SM = 1, on an annual basis, solar energy supplies 22 % of total thermal energy, with a solar field efficiency of 49.8 % and a global conversion of solar energy to electricity of 20.9 %. Increasing the solar multiple, the useful solar input increases while the overall conversion efficiency decreases.

5. ECONOMIC EVALUATION

To establish the economic convenience of the plant, the calculation of the levelized cost of energy LCOE ($/kWh) was carried out, utilizing the algorithm of NREL [18].

LCOE is defined as [18], [19]:

$$LCOE = \frac{(CapitalCost \cdot CRF + Fixed\ O\&M\ Cost)}{8760 \cdot CF} + Fuel\ Cost \qquad (16)$$

Being

$$CFR = \frac{g\,(1+g)^n}{[(1+g)^n - 1]} \qquad (17)$$

In eq. (16), Capital Cost is the cost per kW ($/kW) of the plant, CRF is the capital recovery factor [18], Fixed O&M Cost is the fixed operations and maintenance cost in a year ($/kW y), Fuel Cost is the cost of the fossil fuel ($/MBTU), Heat Rate is the amount of heat developed by the combustion per unit of electricity produced (BTU/kWh), Variable O&M Cost is the variable operations and maintenance cost ($/kWh), 8760 is the number of hours in a year and CF is the capacity factor, ratio of the equivalent number of working hours of the plant in nominal conditions and the number of hours in a year.

In eq. (17), g is the discount rate and n the lifetime of the plant in number of years.

Table 4 reports, at varying of solar multiple between 1 and 1.4, the ground cost, the collectors cost, the power block cost, the total cost, the Unit Capital Cost and the LCOE value.

Table 4. Costs of plant and LCOE

SM	1	1.2	1.4
Ground Cost (M$)	8.70	9.53	10.36
Collectors Cost (M$)	36.70	44.04	51.39
Power Block Cost (M$)	45	45	45
Total Cost (M$)	90.40	98.57	106.75
Unit Capital Cost (M$)	1808	1971	2135
LCOE (cent$/kWh)	8.3	8.4	8.5

The results obtained are based on the assumptions summarized in Table 5 [20].

Table 5. Assumptions

g (%)	4
n	25
Fixed O&M Cost ($/kW)	27.5
Variable O&M Cost ($/kWh)	0.003
Fuel Cost ($/MBTU)	8
Installed collector cost ($/m^2)	291
Occupied ground area (km^2)	1.3 A_{col} +0.18
Ground cost (land cost and construction costs for building and roads) ($/m^2)	25.3
Power Block costs ($/kW)	900 = 650 + 250 of balance of the plant [21]

For Heat Rate and CF values, see Table3. For SM = 1, Table 4 shows a total plant cost of 90.4 M$, a unit cost of 1808 $/kW and a LCOE value of 8.3 cents$/kWh. As SM rises from 1 to 1.4, the cost of the plant increases by 18% and the cost of electricity produced increases by 2.4%.

6. CONCLUSIONS

A calculation method for the design and the evaluation of the annual performance of a 50 MW hybrid plant with linear parabolic solar collectors, using atmospheric air as a heat transfer fluid and natural gas as integrative fossil fuel has been developed.

In the system, the air evolves into an open-type Joule-Brayton cycle, equipped with a three-stage inter-refrigerated compressor and a regenerator that preheats the air before it enters into the solar collectors field; downstream of solar collectors, a natural gas combustor raises the temperature of the combustible gas to a predetermined value (800 °C in the present work); after running the turbine, the combustion gases are sent to the regenerator and then discharged into the atmospheric air.

The cost of the plant, located in Almeria (Spain), was estimated, for SM=1, to be 90.4 M$, with a unit cost of 1808 $/kW.

If the plant is operated each day between sunrise and sunset for an entire year in conditions close to nominal ones, the net electricity produced in one year is 244,000 MWhe, the fuel consumption is 34,575 t of natural gas, the annual average efficiency of solar collectors is 49.8 %, the average thermodynamic efficiency of power block is 42.1 %, the average conversion efficiency of solar energy to electricity is 20.9 %.

The saving of fossil fuel due to the employment of solar energy in the hybrid plant is 22 %, the cost of electricity produced results to be 8.3 cents$/kWh. This value is competitive with that of other plants using renewable energy.

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NOMENCLATURE

A	Area, m^2
a	Aperture of the collector, m^2
CF	Capacity factor
CRF	Cost recovery factor
D	Diameter, m
DNI	Direct normal irradiance, W m^{-2}
E	Annual energy, MWh
F	Solar fraction
g	Discount rate
h	Enthalpy, kJ kg^{-1}
i	Incidence angle
IAM	Incidence angle multiplier
L	Length of the collector, m
LCOE	Levelized Cost of Electricity, \$ kWh^{-1}
m	Mass flow rate, kg s^{-1}
P	Power, kW
Q	Thermal Power, kW
SM	Solar multiple
T	Temperature, K
x	Solar parameter, W m^{-2}

Greek symbols

α	Absorptivity
βc	Compressor pressure ratio
βt	Turbine pressure ratio
γ	Interception factor
ε	Emissivity of the receiver tube
η	Efficiency
ρ	Reflectivity
τ	Transmissivity

Subscripts

A	Air
abs	Absorbed
col	Collector
el	Electrical
env	Glass envelope
f	Fuel
i	Inlet
i, rec	Inside, receiver