INTEGRATING SOLAR POWER IN LARGE COMBINED-CYCLE POWER PLANTS

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Abstract:

Large Solar Thermal Electric Generation Systems are in rapid development in many areas of the world: many of them are based on Concentrated Solar Power technology, using either a central receiver or rows of large parabolic trough solar collectors. The solar thermal thermodynamic conversion appears interesting for integration with high-efficiency combined cycle power plants, running on natural gas and using the combination of gas and steam cycles with advanced matching of the two by accurate design of the Heat Recovery Steam Generator (HRSG). In most of the proposed approaches, however, integration of the solar section for producing steam is performed using a typical “parallel-boiler” scheme. This approach produces an advantage in terms of reducing the consumption of the primary fossil fuel (natural gas); however, the original efficiency of the combined cycle is negatively affected, as the performance of the HRSG decreases.
The idea here proposed is to follow design and off-design control guidelines which try to maintain a high efficiency of the HRSG (including operation with variable radiation conditions), and possibly to improve its performance, using the solar contribution to alleviate the pinch/heat capacity mismatching between the gas and the water/steam circuits. The fundamental idea is that of adding a parallel external solar evaporator to the specific heat transfer bundle located inside the HRSG. The application refers to a large HRSG with three pressure levels, using advanced parabolic trough solar collectors. Two innovative control strategies are proposed for correctly managing the off-design performance: control of the temperature increase across the collector in order to maximize its exergy efficiency (in converting the original energy from the sun) and implementing a dynamic reconfiguration for the solar field in the medium/high pressure sections, which use the same type of collector. The simulation over one representative year produced promising results.

Keywords:

Concentrated Solar Power, Integrated Solar Combined Cycle Power Plant, Parabolic Trough, Solar Collectors, Solar Energy.

1. Motivations

It can be predicted that the global economy will not obtain the desired emission level decrease of 20%, referring to the 1990, improving only energy systems using fossil fuels. Solar power systems are notably suitable as non-risk-creating technologies contributing to the reduction of greenhouse gases. According to the National Renewable Energy Laboratory [1], the installed global renewable electricity between the years of 2000 and 2011 increased from 748GW to 1285GW, which covers about 22% of the electricity generated all over the world in 2011. What is more, it is expected that the major part of electricity produced until 2035 will derive from natural gas or renewable energy, rather than from coal or lignite regarded as traditional fossil fuels. Being aware of existing international clauses restraining the emissions of greenhouse gases, it is an important task to introduce technological solutions allowing the production of electricity from fuels obtaining high efficiencies rates and low emission indicators. A plant combining renewable and conventional energy sources in a large-scale unit is an attractive concept, especially if it is possible to apply it easily to already existing installations. Taking into consideration the above-mentioned arguments, an Integrated Solar Combined Cycle System (ISCCS) is worth a dedicated analysis from many points of view: thermodynamics, economics, preservation of natural resources and mitigation of emissions as well.

2. Technology background – CSPP plants and possible hybridization

The system under consideration belongs to the category of solar-thermal energy conversion power cycles. In order to generate electricity in solar power plants it is essential to use concentrating devices, which allow for the increase of working fluid temperature to the necessary level. A comprehensive review of the status of Concentrated Solar Systems (CSS) can be found in [2]. Different schemes have been proposed and experimented: central receiver systems (CRS), such as Solar Tower and Solar Dish, or decentralized systems with parabolic trough collectors or Fresnel concentrators. In order to keep an uninterrupted power generation during low solar radiation, CSS plants can be equipped with a high-temperature heat storage system, which however complicates the circuit layout and has an impact on daily-averaged performance; the common alternative is a fossil-fuel backup heater. In this light, the idea to combine a concentrating solar field with an independent high-performance fossil-fuelled power block is attractive. The hybrid solar power plant [3] can be operated flexibly in the fuel saving or power boosting mode, and is always available as a back-up system in the eventual case of low solar radiation. Additionally, solar hybridization can be relatively easily applied adding a solar energy generating field to an already existing fossil fuel power plant. Various hybrid installations may be exemplified: PTC50 Alvarado (Acciona Energy Company) is a 50 MWe solar power plant integrating the hybridization of central receiver system with biomass and natural gas firing [2]. A second hybrid Concentrated Solar Power-Biomass plant is also operating in Spain (Borges) by Thermo solar, based on a parabolic-trough solar field [3]. The only coal-hybrid CSS Colorado Integrated Solar Project is not operating any more [4][4].

2.1. Integrated solar combined cycles with parabolic trough concentrators

The most common hybridization possibility is represented by integrated solar combined cycles with parabolic trough concentrators. Different schemes for solar field integration can be implemented. The general objective is to increase the steam generation capacity of the plant, determining a higher power output and improving the fossil fuel economy as well.

In general, using solar heat to preheat water is not very attractive in the CCGT environment, as there is already plenty of heat available in the exhaust gases, used for the steam-generating process with no need of additional support. The traditional option is to integrate the parabolic-trough solar field in parallel with a Heat Recovery Steam Generator (HRSG). The solar system is equipped with an independent steam generator, producing superheated steam. The solar steam is then mixed with the steam produced in the combined-cycle HRSG. The mixed stream drives the turbine and after expansion steam is condensed and subsequently split in two separate flows – the solar steam generator and the HRSG. This solution has been reviewed in [5] and [6], while [7] and [8] are examples of studies conducted on such systems.

The second option of integration - here discussed in some detail – implements, instead, a solar back-up of the evaporator, with a partial series arrangement of the solar fields loops to the HRSG. During sunny hours, as the heat transfer fluid reaches a proper temperature (~ 390°C) an increasing fraction of feed water flow rate, already pre-heated in the economizer, is withdrawn from the HRSG. It is then evaporated in the Solar Steam Generator and re-injected in the HRSG to be superheated using the exhaust gases. This solution increases the steam cycle power output, but can also provide an opportunity to achieve a lower stack temperature and an increased heat recovery from the flue gas stream. The solar evaporator overtakes a portion of saturated steam production and thus the heat transfer in the HRSG can be improved, modifying the pinch conditions in the HRSG.

The largest ISCCS parabolic-trough power plant is located in Indiantown, Florida, United States [10].

Other examples of solar combined cycles power plants using parabolic trough technology are Hassi R’mel ISCCS in Algieria (25 MWe generated via solar field), Ain Beni Mathar ISCC in Morocco (20

MWe), Kurayamat ISCCS in Egypt (20 MWe), YAZD ISCC in Iran (17 MWe) and a small research unit with a solar field's capacity of 5 MWe operated by ENEL in Priolo Gargallo, Sicily, Italy. [9]]

Existing researches ([11], [12]) take under consideration power plants with one of the evaporators supported by the solar field. The idea here proposed is to back-up evaporators at every pressure level in a three-pressure level HRSG.

3. Scope of the study

The main aim of this study is to analyse the possibilities and profits of integration of solar technology into a combined cycle power plant. The integration rests on adding additional, external solar steam generators to every of three evaporators operating at different pressure and temperature level inside the HRSG.

The present approach should be considered novel, since the available literature mainly discusses solar field back-up options limited to one evaporator in the HRSG. Baghernejad and Yaghoubi [12] presented an exergoeconomic analysis where the solar field supported only high pressure evaporators in parallel HRSG configuration, while Behar et al. [11] discussed the instantaneous performance of the first Integrated Solar Combined Cycle System in Algeria where parabolic trough collectors where linked with evaporator in only one pressure level HRSG. Review articles of CSPP technology [5] point as well at examples of solar field integration with only one evaporator. Moreover, the design and control guidelines suggest that it is more reasonable to generate more heat at lower temperature level, when the solar radiation conditions are poor. Hence, a novel idea of dynamic allocation of the solar collector field is proposed, and will be described in some detail. The model is also enriched by the maximum exergy control routine, not common in existing power plants, which is described in chapter 5.1.

The work consisted of 3 main stages: the combined cycle power plant model, the solar fields model, and the integration of these two components. The simulations were performed for the climate conditions of Southern Poland. It should be emphasized that the majority of the available analyses consider more climatically advantageous locations like Algieria [11] or Iran [12]. As the northern countries are not sufficiently sunlit, it may seem that concentrated solar power technology has no future in those locations. However, favourable conditions exist in Poland in terms of availability of land and of the possibility of external financing, so that a preliminary feasibility study was motivated.

4. Combined Cycle System model

A model of the Combined Cycle System (CCS) was built with the use of Equation Solver Modular System (ESMS), a simulation tool developed for complex power plant simulations [13]. The simulation code is based on a flexible computerized platform allowing to calculate parameters in a power cycle without limitations to Application-Specific systems. Although the user does not work with a graphical interface, specifically devoted files enable to define the links between components, to define the specific features of the component like efficiency or pressure loss coefficient, and finally to specify boundary conditions. Each power cycle component is represented by equations referring to the fundamental physical and thermodynamic principles. Hence the power plant under study is represented by a group of algebraic equations, data describing boundary conditions and continuity equations. The solution is found by an iterative process.

Input data for the model are based on a Polish power plant under construction. The mathematical model of the CCS consists of the three-pressure Heat Recovery Steam Generator (HRSG) and of the steam plant island. No detailed model of the gas turbine is included: the gas turbine is represented only by the values of temperature, mass flow rate and by the composition of the flue gases. The data refer to a GE 9f 5-series gas turbine. The characteristics of the CCS model are resumed in table 1Table 1. The value of the Pinch point temperature difference was set at 8K at every pressure stage.

Once the model has been prepared, it is possible to perform a whole year simulation considering the effect of variable environmental conditions. Under the off-design simulation step, the surface of every heat exchanger inside the HRSG has already been set in the design thermodynamic analysis. The change of the environmental conditions affects the parameters, temperature profiles, power outputs of the power plant. The simulation was conducted for the meteorological data of Cracow, available in the TRNSYS library [14]. Consuming over 720 Mm^3 (0,5 million tons) of natural gas, the CCS produces 3998 GWh net electric energy, with a year-average efficiency of 58,26%. The indicator of fuel chemical energy consumption equals then 6161 GJ per GWh energy produced.

Table 1 Characteristic parameters of the combined cycle power plant

Parameter			Value
Low pressure level	Pressure		0,34 MPa
	Temperature		534 K
	Steam mass flow rate		10,7 kg/s
	Low pressure turbine power output		70,547 MW
Intermediate pressure level	Pressure		2,8 MPa
	Temperature		814 K
	Steam mass flow rate		105,4 kg/s
	Intermediate pressure turbine power output		55,021 MW
High pressure level	Pressure		12,4 MPa
	Temperature		814 K
	Steam mass flow rate		94,7 kg/s
	High pressure turbine power output		33,565 MW
Stack temperature			363,45 K
Gas turbine		Net power output	292,470 MW
		Net efficiency	38,07 %
Auxiliaries (pumps)			1,387 MW
Overall combined cycle power plant net efficiency			58,6 %

The subsequent target was to incorporate heat flows from the solar fields into HRSG. At this stage of work it is possible to calculate the nominal heat duty of the evaporators. A substantial fraction of the evaporator heat duties should be provided by correctly-sized solar fields, thereby determining the number of solar collectors to be installed. The design considers the best solar radiation conditions possible in the site; the study includes the off-design analysis, which should be extended to conditions of stable work of the conventional power plant, when solar radiation is unavailable. The assignation of heat duties to every evaporator considered the influence of increasing the additional heat rate to every single evaporator, or the change of steam turbine power output, pinch point temperature increase, heat transfer from flue gases to water, and stack temperature. A satisfactory level of the heat duties of the solar back-up evaporators was found performing a detailed sensitivity analysis: respectively, 40, 20 and 70 MW of thermal energy should be provided by the solar fields at the low, intermediate and high-pressure evaporator levels. A general schematic of the power plant resulting from the solar hybridization is shown in Fig. 1.

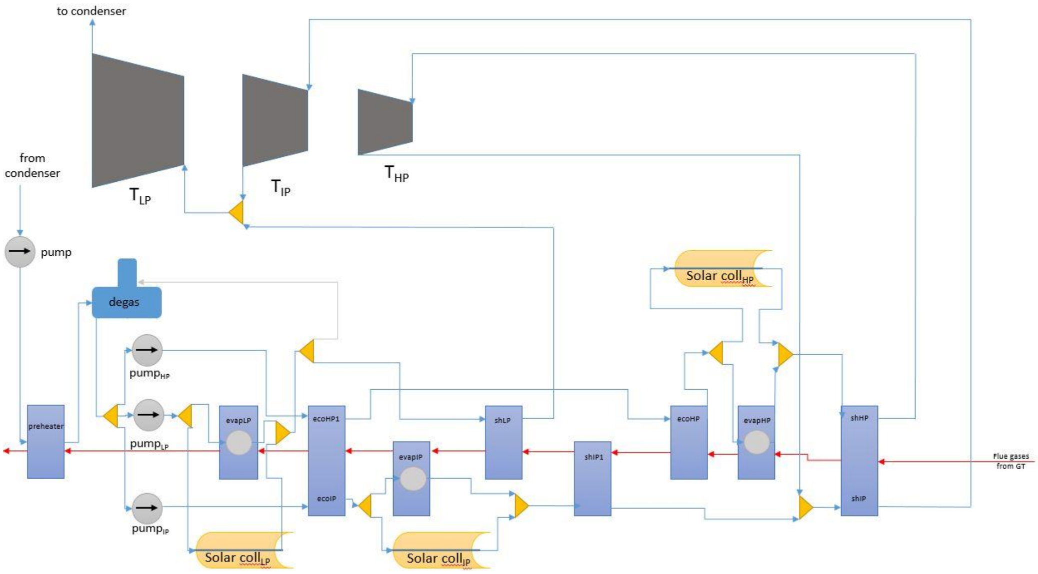

Fig. 1 General schematic of the hybrid ISCCS

5. Model of the solar collector fields

The model has been prepared using TRNSYS as a computational package enabling simulation of operation in quasi-stationary states. It is possible to perform a whole year simulation of the operation of the solar collector referring to built-in meteorological libraries. The mathematical formulation of the parabolic trough collector performance was written in EES and then coupled with the TRNSYS model.

Since the saturation temperature at the low pressure level is 412K, it is within the temperature limits for the mid-temperature solar collectors; the model *PolyTrough* 1800 manufactured by *NEP SOLAR AG* [15] was selected as a good economic alternative to high-temperature collectors,. The low-pressure solar evaporator is thus arranged as a completely separate component in terms of collectors type. For the high and intermediate pressure levels in the HRSG, the saturation conditions enforce the use of high-temperature solar collectors. *EuroTrough* collectors *ET-150* are considered [16], with heat transfer fluid *SYLTHERM 800* [17]. The collector loops for the high and intermediate pressure evaporators collectors are arranged as a flexible Solar Field: -in practice, the amount of loops dedicated to the intermediate or high-pressure evaporators can be adapted to the meteorological conditions (the intermediate pressure level is favoured for low radiation, allowing to operate the solar collectors with reasonable efficiency even under these conditions).

Knowing the heat demand under the design solar radiation condition for the backup of each evaporator, it is possible to determine the size of the solar fields. This requires the definition of the useful heat gain from one single loop of collectors, which can be calculated as follows:

$$\dot{Q}_u = \eta_{coll} \cdot \dot{Q}_{incoll}, \qquad (1)$$

$$\dot{Q}_{incoll} = A_{ap} \cdot x \cdot loops \cdot I_{ap}. \qquad (2)$$

The efficiency of the solar collector is defined as the ratio of the useful heat gain and the incident solar irradiation on the aperture area [16], and it is usually given as a function of $\Delta T/I_{ap}$:

$$\eta_{coll} = \eta_{0_n} \cdot IAM - (a_1 + a_2 \Delta T)\frac{\Delta T}{I_{ap}}. \qquad (3)$$

η_{0_n} is the optical efficiency without accounting for influence of the incidence angle, a_1 and a_2 are the linear and quadratic heat loss coefficients due to the convection, conduction and radiation. The IAM measures the losses connected with daily changes of angle of incidence influencing the directional sensitivity of the glass surface absorption and of the mirror reflectivity. According to technical data of the manufacturers, the data in table 2 were applied:

Table 2 Collectors efficiency parameters

Solar collector type	Optical efficiency	Heat loss coefficients	
	η_{0_n}	a_1 $\frac{W}{m^2 K}$	a_2 $\frac{W}{m^2 K^2}$
PolyTrough 1800	0,689	0,39	0,0011
ET-150	0,741	0,0431	0,000503

5.1. Temperature rise across the solar collector

The energy balance of the solar collector establishes the link between mass flow rate, useful heat and temperature rise of the heat transfer fluid:

$$\dot{Q}_u = \dot{m}\ c_f \cdot (T_{out} - T_{in}) \quad (5)$$

Referring to Fig. 2, once the evaporator temperature is assigned, the flow rate is determined specifying the value of ΔT_{app} (in the present case, 10K) and that of the heat transfer fluid temperature difference, eg. ΔT_{HTF}=10K.

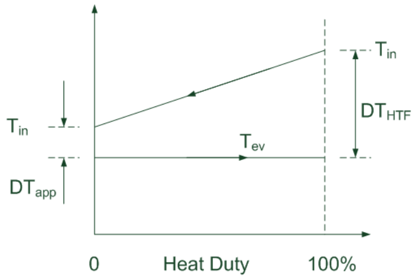

Fig. 2 Heat transfer diagram of solar evaporator

An enhancement of the control law in the solar collector model can be introduced, considering the increase of temperature of HTF flowing through absorber tubes. Rather than setting ΔT_{HTF} as a fixed value, its value can be dynamically adapted according to the radiation and environmental conditions. The fundamental idea of this control law is to maximize the collector exergy efficiency [19]. In the present model, it is assumed that the exergy of radiation equals its energy. In reality, the sun emits its radiation at very high temperature of nearly 6000K (or 5800K). A fraction of the exergy of solar radiation is lost due to transmission through the space, scattering by the atmosphere, to be finally utilized at respectively low temperatures [20], [21]. The approach here presented does not include the model of solar radiation exergy destruction itself, and it is justified by the fact, that the designer does not have the aspirations of improving the exergy balance by decreasing the irreversibility occurring during reactions in the sun at conditions equal to black-body temperature [22].

In practice, *Maximum Exergy Control* is implemented into an iterative numerical procedure searching the outlet temperature which maximizes collector's exergy efficiency for every hour of the whole year simulation. A standard chord interpolation method was applied in this case. Application of this control law typically produces an increase of 2 percentage points in the exergy efficiency of the IP and HP solar collectors.

5.2. Design conditions

A simulation of 8760 hours of operation for the three solar collector loops was performed using TRNSYS. The Solar Field was therefore designed considering the day and hour for which the useful heat gain from the high pressure level collectors is the highest (representative hour of 17th July). Table 3 presents the main results of the unitary simulation.

Table 3 Operational parameters for each evaporator loop

Parameter	Solar collector type		
	Low pressure back-up	Intermediate pressure back-up	High pressure back-up
HTF inlet temperature	421,71 K	516,41 K	611,88 K
HTF outlet temperature	503 K	673 K	650,78 K
Collector energy efficiency	55 %	64 %	62 %
Collector exergy efficiency	31 %	43%	43 %
HTF mass flow rate	0,03 kg/s	2,96 kg/s	11,17 kg/s
Useful heat gain from 1 loop, $\dot{\boldsymbol{Q}}_{\boldsymbol{u}}$	7,071 kW	1457,488 kW	1414,62 kW
Number of collectors in 1 loop	1	4	4

The difference in useful heat gain value for the low pressure back-up collector loops results mainly from the difference between aperture area of two collector types. Since the temperature difference depending on maximum-exergy routine in the high pressure solar collector is much lower than of intermediate-pressure collector, the mass heat transfer fluid flow rate is consequently higher. The different values of the temperature difference across the high and intermediate pressure collectors also influence the useful heat gain.

The proposed modelling approach allows to determine the nominal size of the three solar collector fields at the design point, which is a minimal part (one reference day and hour) of the whole year simulation. As usual in CSPP plants, an enlargement in the size of the solar field is applied by means of a solar multiplication factor, which is used to oversize the solar field relatively to the size needed to fulfil the design heat balance. In the present case, a SMF = 1,5 was applied. The resulting additional loops are not used in the design point energy balance, however they are available to be used in off-design mode and contribute to the overall installed field size and economical cost at every step of analysis [23].

5.3. Dynamic allocation of the solar field

The collectors used for the HP and IP evaporators are of the same type and model (Eurotrough ET150). Therefore, the two fields could represent, logistically, one joint Solar Field. The idea of dynamic allocation of the collector loops to the HP and IP circuits is coupled to the augmented possibilities for steam evaporation when a solar multiplication factor is introduced. The guideline concept is based on the statement that it is more valuable to generate more heat at lower temperature in the solar collectors, than to try to obtain a very high scheduled temperature, when the radiation conditions are poor. Consequently, a higher priority is given to the intermediate pressure solar collector loops than to the high pressure level. In order to apply the allocation plan to the solar field, a specific TRNSYS module named DARE (Dynamic Allocation Routine Ees) was programmed.

The DARE module implements a procedure working on logical conditions. As a first condition, the collector is turned to off-state, when no positive useful heat gain is available. It is possible that while for the HP collectors $\dot{Q}_u$= 0, for the IP collectors $\dot{Q}_u$ would have a positive value. Therefore, DARE assigns priority to the IP loops and first tries to cover completely the IP evaporator heat load, determining the necessary loops needed to fulfil the heat demand input at every calculation step. If excess collectors are available (also thanks to a SMF), the loops to be assigned to the HP level are defined. A simplified block diagram presented in Fig. 3 illustrates the logical procedure of DARE.

The implementation of the DARE idea into a real installation would require an advanced manifold arrangement and its automation. It would be advisable to equip the solar loops with three-way valves and a pressure-relief buffer system to allow switching of the field.

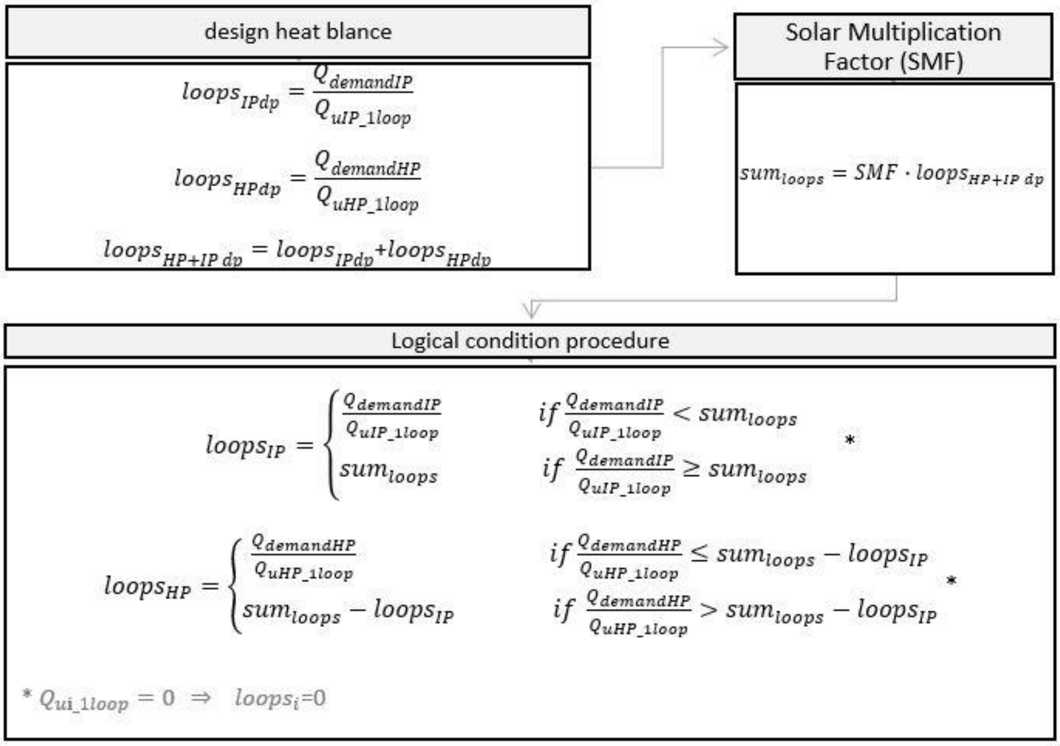

Fig. 3 Logical map of the Dynamic Allocation Routine EES (DARE)

6. Integrated model of solar combined cycle power plant

The task is to assemble a model with three solar collectors fields (LP; IP and HP with dynamic allocation) which support the flue gas/steam heat recovery evaporators operating inside the HRSG, covering the variable environmental conditions (radiation and ambient temperature) and pursuing the objective of fuel economy. The path followed for the implementation of the integrated model and the results are described in the following subchapters.

6.1. Design hour simulation

An initial one-hour simulation is performed to determine the potential of the proposed integration on the operation of the power plant under the best solar radiation conditions. As independent operation of the power plant in gas-only mode must be maintained, computations are done activating the off-design mode of ESMS, maintaining the original size of the heat exchangers in the HRSG. Table 4 presents the impact of the solar back-up on the main parameters of the HRSG evaporators for this design hour heat balance.

Table 4 Performance data of the evaporators with solar contribution (best radiation conditions) with the heat transfer rate from flue gases compared to conventional CCS

parameter	Evaporator level		
	Low pressure	Intermediate pressure	High pressure
Heat transfer from flue gases	12,81 MW	7,86 MW	93,01 MW
Heat transfer from flue gases no solar back-up	22,47 MW	20,25 MW	115,45 MW
% fraction of solar substitution (heat transfer)	43 %	61,2 %	19,4 %
Water mass flow rate to flue gases HE	5,68 kg/s	3,96 kg/s	70,79 kg/s
Heat transfer from solar steam generator	40 MW	20 MW	70 MW
Water mass flow rate to parallel solar HE	17,74 kg/s	10,08 kg/s	53,28 kg/s
Pinch point temperature difference	7,99 K	3 K	6,46 K

When the evaporators are supported by the maximum value of additional heat from the solar fields, the whole power plant profits from it: the stack temperature is decreased, the heat transfer inside HRSG is increased, the steam turbines power output and the overall fuel efficiency are improved. The results of the design-hour simulation are shown in table 5.

Table 5 Solar back-up effect on the Combined Cycle Power Plant operation

data	CCPP (no solar)	Integrated Solar CCPP	Difference
Stack temperature	363,45 K	356,16 K	7,29 K
Heat transfer in the HRSG	421,6 MW	424,2 MW	2,6 MW
Low pressure steam turbine power output	70,55 MW	92,92 MW	22,37 MW
Intermediate pressure steam turbine power output	55,02 MW	68,12 MW	13,1 MW
High pressure steam turbine power output	33,56 MW	40,13 MW	6,57 MW
Power plant fuel efficiency	58,61 %	64,16 %	5,55 %pts

The design-hour simulation allows to calculate the nominal size of the solar fields. As the amount of heat produced by one loop of collectors at every level under those solar condition is already known (table 3), this value has to be multiplied to meet the design heat balance; furtherly, the solar multiplication factor should be applied, while considering the intermediate-high-pressure Solar Field. The resulting number of loops assigned to every evaporator is given in table 6.

Table 6 Estimate of total solar field area

Evaporator	Number of loops (design heat balance)		Number of loops with SMF	Number of collectors in 1 loop	Surface area of 1 collector	Total area (collectors only)
LP	5656		SMF=1 → 5656	1	18,5 m^2	104636 m^2
IP	13	13+47=60	SMF=1,5 → 90	4	817,5 m^2	294300 m^2
HP	47					

6.2. Whole year simulation

Considering the variability in time of the solar radiation it is recommendable to perform a whole year simulation of the solar-integrated power plant, to obtain the overall profile of energy produced.

6.2.1. Application of the Dynamic Solar Field allocation

As a first step, a whole-year simulation of the solar heat production was performed in TRNSYS. The solar fields are nominally sized at this stage, and the dynamic allocation procedure is then applied with variable climatic data. Fig. 4 illustrates the effect determined by application of the Dynamic Solar Field allocation on the operation mode of the solar collectors.

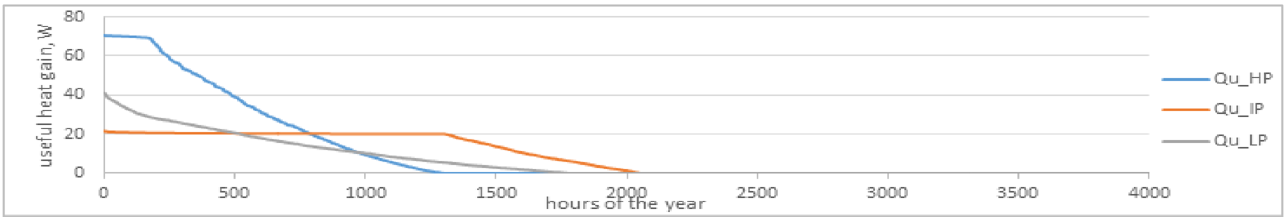

Fig. 4. Useful heat gain obtained throughout the year for the three solar-driven evaporators (HP-high pressure, IP-intermediate pressure, LP-low pressure)

Thanks to the application of the dynamic arrangement of loops, and to the enlargement of the solar field by application of the SMF, the number of operational hours at the design 20MW heat duty for the IP evaporator is about 1300 h/yr. This result is linked to the priority of assigning loops to the intermediate pressure level loops. The IP evaporator can be operated for 2051 hours (at reduced heat duty), while the HP evaporator profits from the sun only for 1296 hours, with a heat duty close to the design value of 70MW for about 170 h/yr. In consequence of selecting a lower-quality collectors in the LP field, and not applying the SMF, the solar support of the LP evaporator is limited to less than 2000 h/yr.

6.2.2. Results of the coupled simulation (solar field + power plant)

A real-time simulation was not conducted for the combined solar field and gas-fuelled power plant, because of the very large number of repetitive ESMS runs which would be necessary. However, after the TRNSYS simulation, a whole year set of results of heat rates from every collector field is known. This set was exported to ESMS as an “additional heat source” input in the evaporator-components. Consequently, it is possible to calculate the power generated in the solar-assisted off-design condition (covering both solar radiation and the effect of variable ambient temperature). Figure 5 shows the electric energy produced over one year by the ISCCS in comparison with that of the conventional CCS.

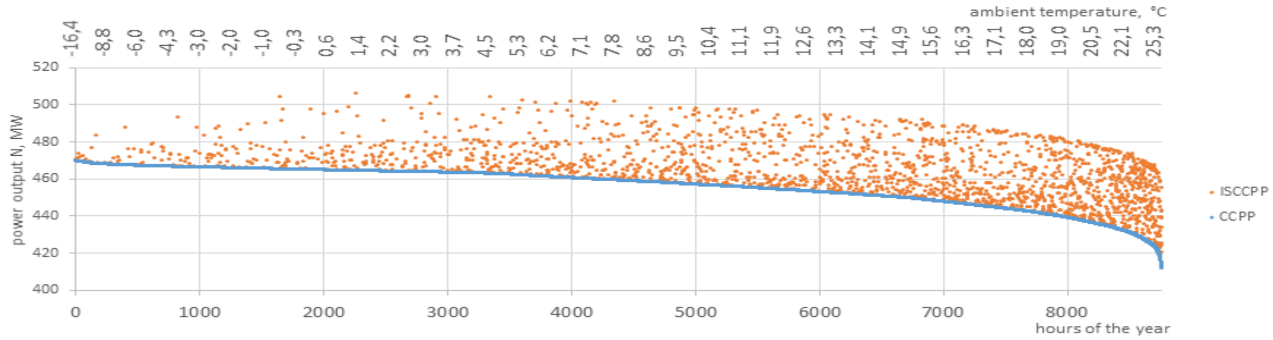

Fig. 5. Structured graph of net power output for conventional Combined Cycle System and Integrated Solar Combined Cycle System ordered by ambient temperature (top scale)

As the gas turbine produces the largest share of the total power plant electricity output, when the ambient temperature is increased the power generation of the ISCCS decreases, following the conventional CCS production trend. However, the increase in power output during sunny hours is remarkable, ranging between 0 and 40MW. One can also notice in the chart, that as the temperature becomes higher, the “cloud” of red dots becomes denser. This results from the fact that the warmest days of the year usually coincide with the sunniest days of the summer. It appears thus favourable that the solar back-up covers the decrease in the power output caused by the ambient dependence of thegas turbine performance.

Fig. 6 presents graphically the increase of the fuel-only power plant efficiency, obtained with the support of the solar field evaporators.

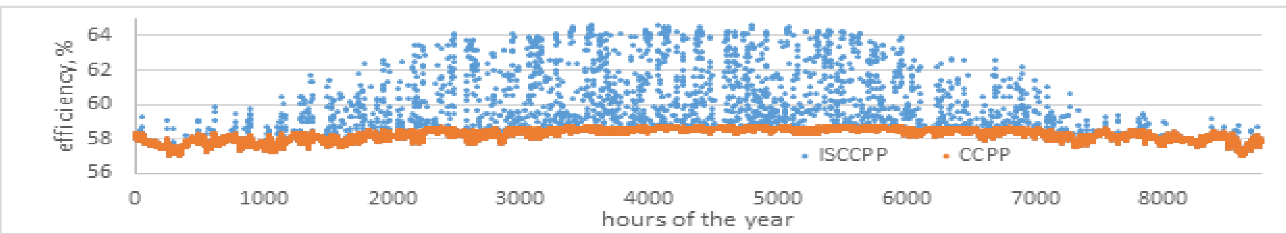

Fig. 6. Time history of net power plant fuel efficiency during the year for the case of ISCCS and conventional CCS

As mid-summer months are most often well irradiated as well, the efficiency increase is almost symmetrical. The same trend is visible while comparing the profile of flue gases temperature change for the power plant with and without solar collectors' contribution. The chart is presented in Fig. 7. It is clear that the objective of integrating the solar contribution in a manner which increases the heat recovery in the HRSG (rather than decreasing it as usual with conventional designs) has been reached. The lowest stack temperature scores 354,6K, and it occurs in the 4071st hour of the year, on June 19th; its value is nearly 10K less than the design value (but still reasonable in term of demand of corrosion-resistant materials).

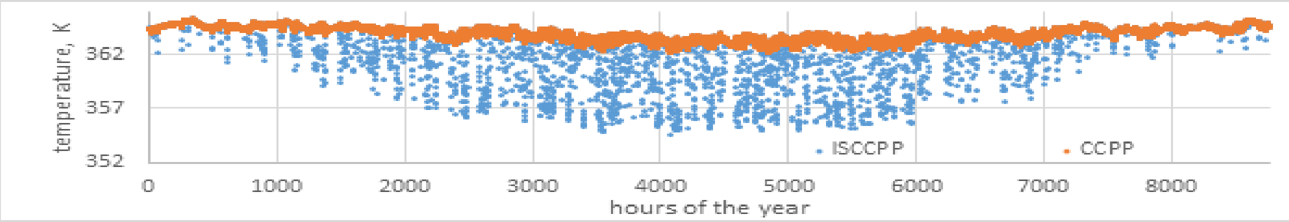

Fig. 7. Change of the stack temperature during the year for the case of ISCCS and conventional CCS

One year of ISCCS's operation means generation of more than 4040 GWh electric energy demonstrating an increase of 32,8 GWh, comparing to a pure CCS. The indicator of fuel energy consumption is decreased from 6161 to 6111 GJ/GWh. Consequently, 1636 GJ of natural gas resource are annually saved, that is about 33,4 tons of natural gas. This enables to avoid nearly 123 tons of CO_2 emissions per year. Compared to the overall emissions of the CCS, the reduction of CO_2 emissions achievable is about 7,5%.

7. Conclusions

The idea of implementing solar integration in a combined-cycle power plant, with a partial serial arrangement substituting part of the evaporators in a three-pressure-level HRSG has been demonstrated as interesting and advantageous. The proposed solution is not only effective in augmenting the power plant productivity and decreasing the consumption of the primary resource (natural gas); moreover the solar integration is able to achieve an improvement rather than a decrease in the heat recovery performance of the HRSG. For the design heat balance under the best (design) solar conditions, it was possible to reduce the heat transfer from the exhaust gases by 43% in the low pressure evaporator, by 61,2% in the intermediate pressure evaporator and by 19,4% in the high pressure evaporator.

The solar circuits are operated with a control law maximizing the exergy efficiency of the solar collectors, which are responsible of the largest irreversibilities in solar thermal energy conversion systems. The idea of a dynamic arrangement of the Solar Field for intermediate and high pressure level collectors was also introduced, giving priority of heat production to the loops working for the intermediate evaporator, which work with a more favourable thermal and exergy efficiency.

The analysis was not extended to the economic profitability, however it is possible to draw some final remarks about the attractiveness of the ISCCS solution for the reference case here examined, which is located in South Poland. The analysis showed that the heat from the solar collectors could be utilized only during 2000 hours per year, which is about 28% of the power plant operation time in one year. For the design conditions, the calculated net power plant's fuel efficiency equals 64,2%, which is distinctly higher than the CCS design efficiency of 58,6%. The design point efficiency can be compared to that obtained by Behar et al. [11] – the authors (referring to a smaller power plant with a single-pressure HRSG, and to a location in Algeria) claim to achieve a fuel efficiency of 67% on 21st June.

The increase in efficiency is directly related to the environmental conditions (radiation and ambient temperature); it appears that the solar integration is to some extent compensating the natural decrease in efficiency of the CCS with high ambient temperatures. In terms of whole-year average efficiency, the value of about 58,3% with no solar integration is increased to nearly 58,8% when concentrated solar power with the proposed arrangement is activated. Moreover, a slight increase in electricity production is present. Since the ISCCS generates this increment with no additional use of fossil fuels, the fuel consumption (natural gas) is decreased of about 33,4 t/yr; the decrease of CO_2 emissions with respect to the non-solar-assisted power plant is notable, both in absolute (-122,5 t/yr) and relative terms (- 7.5%).

Nomenclature

a_1	linear heat loss coefficient, W/(m^{2}K)
a_2	quadratic heat loss coefficient, W/(m^{2}K^2)
A_{ap}	collector area aperture, m^2
c_f	specific heat of the heat transfer fluid, J/(kg K)
I_{ap}	incident solar radiation, W/m^2
loops	number of loops in Solar Field, -
$\dot{m}$	mass flow rate, kg/s
$\dot{Q}_{incoll}$	heat transfer to collector, W
$\dot{Q}_u$	useful heat rate, W
SMF	Solar Multiplication Factor
T	temperature, K
x	number of solar collectors in one loops, -

Acronyms

CCS	Combined Cycle System
CRS	Central Receiver System
CSPP	Concentrated Solar Power Plant
ESMS	Energy System Modular Simulation
ISCCS	Integrated Solar Combined Cycle System
SMF	Solar Multiplication Factor

Greek symbols

ΔT	temperature increase, K
η	solar collector energy efficiency, -
η_o	solar collector optical efficiency, -

Subscripts

app	approach point
HP	high pressure level
HTF	Heat Transfer Fluid
IP	intermediate pressure level
LP	low pressure level

References

[1] Collective work of the National Renewable Energy Laboratory, 2011 Renewable Energy Data Book, U.S. Department of Energy, Report No.: DOE/GO-102012-3598, February 2013.

[2] Baharoon D.A., Rahman, H.A., Omar, W.Z.W., Historical development of concentrating solar power technologies to generate clean energy efficiently – A review, Renewable and Sustainable Energy Reviews 41, 2015, 996-1027

[3] Collective work, Thermosolar Borges, The first commercial CSP hybridized with biomass in the world, Abantia brochure, April 2012.

[4] National Renewable Energy Laboratory, Concentrating Solar Power Projects, Project overview- Available at: http://www.nrel.gov/csp/solarpaces/project_detail.cfm/projectID=75.[accessed 09.2.2014].

[5] Jamel M.S., Rahman A.A., Shamsuddin A.H., Advances in the integration of solar thermal energy with conventional and non-conventional power plants, Renewable and Sustainable Energy Reviews 20, 2013, 71-81

[6] Peterseim J.H., White S., Tadros A., Hellwig U., Concentrating solar power hybrid plants – Enabling cost effective synergies, Renewable Energy 67 (2014) 178-185.

[7] Bakos G.C., Parsa D., Technoeconomic assessment of an integrated solar combined cycle power plant in Greece using line-focus parabolic trough collectors, Renewable Energy 60 (2013) 598-603.

[8] Hosseini R., Solatani M., Valizadeh G., Technical and economic assessment of the integrated solar combined cycle power plants in Iran, Renewable Energy 30 (2005) 1541-1555.

[9] CSP World, Concentrated Solar Power and Renewable Energy news. A world map of CSP installations- Available at: http://www.csp-world.com/cspworldmap. [accessed 09.2.2014].

[10] Clean Energy Action Project, Hybrid Renewable Energy System Case Studies, Case study- Available at: http://www.cleanenergyactionproject.com/CleanEnergyActionProject/CS.FPL_Martin_Next_Generation_Solar_Energy_Center___Hybrid_Renewable_Energy_Systems_Case_Studies.html. [accessed 09.2.2014].

[11] Behar O., Kellaf A., Mohamedi K., Belhamel M., Instantaneous performance of the first Integrated Solar Combined Cycle System in Algeria, Energy Procedia 6 (2011) 185-193.

[12] Baghernejad A., Yaghoubi M., Exergoeconomic analysis and optimization of an Integrated Solar Combined Cycle System (ISCCS) using genetic algorithm, Energy Conversion and Management 52 (2011) 2193-2203.

[13] Carcasci C., Facchini B., A numerical method for power plant simulations, Journal of Energy Resources Technology 118 (1996) 37-43.

[14] Transient System Simulation Tool, Weather Meteonorm Data, PL-Karkow-Balice-125660

[15] Collective work of SPF Institut für Solartechnik, Technical data of *PolyTrough* 1800, ordered by NEP SOLAR AG, test report No.: *C1549LPEN*.

[16] Collective work, Extension, Test and Qualification of EUROTROUGH at Plataforma Solar de Almeria, Project funded by the European Community under the Non-Nuclear Energy Programme, 2003 report No.: ERK6-CT1999-00018.

[17] Collective work, Syltherm 800 technical data sheet, the Dow Chemical Company, 2001, report No.: CH-153-311-E-1101.

[18] Duffie J. A., Beckman W. A., Solar Engineering of Thermal Processes,John Wiley & Sons Inc., 1991

[19] Manfrida G., Gerard V., Maximum Exergy Control of a solar thermal plant equipped with direct steam collectors, Int. J. of Thermodynamics 11 (2008) 143-149.

[20] Petela R., Exergy of heat radiation, transactions of the ASME, Journal of Heat Trasnfer 86 (1964) 187-192.

[21] Petela R., Exergy of undiluted thermal radiation, Solar Energy 74 (2003) 469-488.

[22] Manfrida G., Kawambwa S.J.M., Exergy Control for a Flat-Plate Collector/Rankine Cycle Solar Power System, Journal of Solar Energy Engineering, 113 (1991) 89-93.

[23] St. Laurent S.J., Thermocline Thermal Storage Test for Large-Scale Solar Thermal Power Plants, Sandia National Laboratories student Intership Program 5th Annual Symposium, Albuquerque, 2000, report No.: SAND2000-2059C.