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An improved model to evaluate thermodynamic solar plants with cylindrical parabolic collectors and air turbine engines in open Joule–Brayton cycle

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A B S T R A C T

An improved model to analyze the performance of solar plants operating with cylindrical parabolic collectors and atmospheric air as heat transfer fluid in an open Joule–Brayton cycle is presented. In the new model, the effect of the incident angle modifier is included, to take into account the variation of the optical efficiency with the incidence angle of the irradiance, and the effect of the reheating of the fluid also has been studied.

The analysis was made for two operating modes of the plants: with variable air flow rate and constant inlet temperature to the turbine and with constant flow rate and variable inlet temperature to the turbine, with and without reheating of the fluid in the solar field. When reheating is used, the efficiency of the plant is increased.

The obtained results show a good performance of this type of solar plant, in spite of its simplicity; it is able to compete well with other more complex plants operating with different heat transfer fluids.

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1. Introduction

In almost all thermodynamic solar plants 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 work 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 an HTF and as a heat storage medium.

Recently, many 50 MW plants have been built in Spain: Andasol 1, Andasol 2, Andasol 3, Solnova 1, Solnova 3, Solnova 4, Ibersol Ciudad Real, Alvaredo 1, Extresol 1, La Florida; in all these plants synthetic oil is used as heat transfer fluid, molten salts as heat storage medium and water steam working in the Rankine cycle [2–5]. These plants combine the advantages of the tested use of the oil as an HTF and of the use of molten salts which present a heat capacity better than that of the oil.

Rubbia [6], 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 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 [5,7]. The main problem for such plants is the necessity to heat all the pipes containing the salts continuously to avoid solidification and this involves energy consumption.

In a previous paper [8], 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). The air is taken from the outside, compressed in an inter refrigerated compressor (in order to reduce the compression work), sent to the field of solar collectors for its heating, sent from the collectors to the turbine connected with the electrical generator and then discharged. Before discharging to the ambient, it is convenient to recover a part of its enthalpy, preheating the air outcoming from the compressor in the regenerator before its inlet to the solar field, see Fig. 1. The thermodynamic cycle followed from the air is an open-type Joule–Brayton cycle.

This plant is particularly simple and also attractive from an economic point of view. Theoretically, it is possible to operate the

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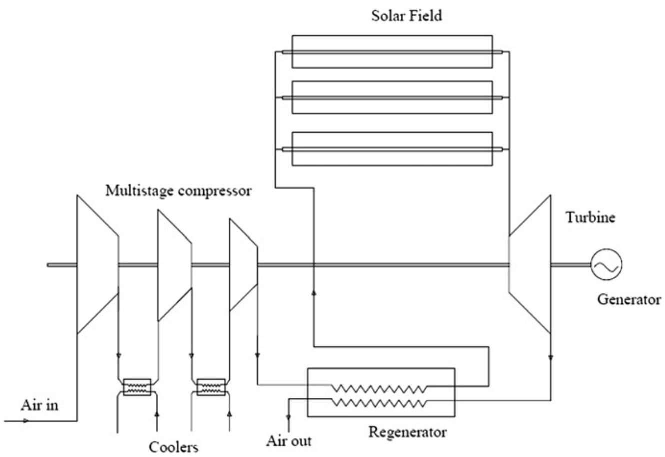

Fig. 1. Scheme of the air turbine solar plant without reheating

plant with variable flow rate and constant outlet temperature from the solar field, or with constant flow rate and outlet temperature variable with the solar irradiance. In the latter case, when the irradiance presents slow values, it is convenient to operate the compressor without intercooling, in order to heat the air by compression and also to by-pass the regeneration, see ref. [8].

In this paper, another plant configuration has been also studied, in which the solar field is split into two sections: the fluid heated in the first section is sent to the turbine where it undergoes a partial expansion, then it goes to the second section of the solar field to be reheated by the solar energy and then it completes its expansion in the low pressure part of the turbine, see Fig. 2. It will be demonstrated in the following that this type of plant has the best performance. It is first time, to knowledge of the authors, that reheating of the fluid inside the collectors is proposed.

No work on the analysis of parabolic trough power solar plants cooled by atmospheric air evolving in an open Joule–Brayton cycle was found in the literature, except ref. [8].

2. Analysis of parabolic collectors

In another previous work [9] two calculation codes were presented: the first, named STS code, for steady state conditions, the second, named STT code, for transient conditions, both able to carry out the thermal analysis of linear parabolic collectors. The receiver tube is subdivided into an arbitrary number of axial steps, each

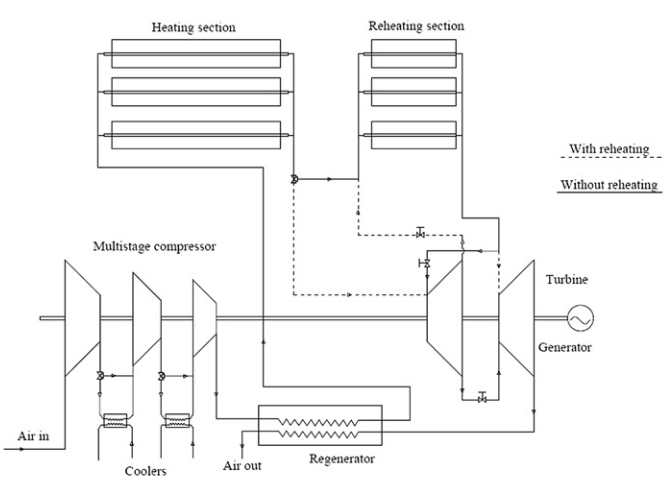

Fig. 2. Scheme of the air turbine solar plant with reheating

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 two codes can consider oil, molten salts, carbon dioxide and air as heat transfer fluid. The calculation model implemented in the codes and the solution technique of the heat balance equations are explained in detail in the reference [9].

Since the differences between the values of collector efficiencies obtained by the two codes were generally negligible, in the present work the stationary code STS was used.

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

$$P_{\mathrm{sol,abs}} = I_{\mathrm{bn}} \cdot \mathrm{IAM} \cdot \eta_{\mathrm{opt}} \cdot A_{\mathrm{col}} \tag{1}$$

where I_{bn} is the direct normal irradiance, IAM is the incident 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 the equation [4,10]:

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

The normal optical efficiency is defined as:

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

where ρ is the reflectivity of the parabolic mirrors, τ_{env} is the normal transmissivity of the glass cover, α_{abs} is the absorptivity of the absorber and γ is the interception factor of the radiation due to the error of the sun tracking apparatus.

In our previous paper the optical efficiency was kept constant, there is a value of IAM equal to cos i was assumed [8].

The useful thermal power P_{u} transmitted to the fluid in the collector is calculable by the equation:

$$P_{\mathrm{u}} = m \cdot (h_{\mathrm{u}} - h_{\mathrm{i}}) \tag{4}$$

were m is the flow rate, h_{u} 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 thermal efficiency η_{ter} of the collector is defined as:

$$\eta_{\mathrm{ter}} = \frac{P_{\mathrm{u}}}{P_{\mathrm{sol,abs}}} \tag{5}$$

The global efficiency of the collector η_{col}, ratio of the useful power and the direct normal solar power projected on the collector, is defined as:

$$\eta_{\mathrm{col}} = \frac{P_{\mathrm{u}}}{P_{\mathrm{sol,inc}}} \tag{6}$$

where

$$P_{\mathrm{sol,inc}} = I_{\mathrm{bn}} \, A_{\mathrm{col}} \cos i \tag{7}$$

The geometrical and optical data of the collectors utilized are reported in Table 1. 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. This is the commun orientation selected in the parabolic though plants because the annual integrated solar energy is greater than the one collected for the East-West orientation.

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