
A CALCULATION MODEL FOR A THERMODYNAMIC ANALYSIS OF SOLAR PLANTS WITH PARABOLIC COLLECTORS COOLED BY AIR EVOLVING IN AN OPEN JOULE-BRAYTON CYCLE

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ABSTRACT

The paper presents a model able to carry out a thermodynamic analysis and evaluation of the annual performance of solar plants provided with cylindrical parabolic collectors cooled by atmospheric air, evolving in an open-type Joule-Brayton cycle. In order to increase the cycle efficiency, the air compressor is inter-refrigerated and regeneration is used.

In the paper a variant of the plant configuration is also studied, with two passages of the air in the collectors and its reheating after a first expansion in the high pressure stage of the turbine. In some circumstances this type of plant performs better than the first.

The plant proposed is particularly simple, is able to compete well with other more complex plants operating with different heat transfer fluids and is also attractive from an economic point of view.

Calculation results are reported for plants located in some Italian and foreign places, in terms of annual electricity production, average efficiencies of collectors, turbine and whole plant.

1. INTRODUCTION

Up to now, the most used technology in the field of solar thermodynamic plants is that of Parabolic Trough Systems. Direct radiation is collected by parabolic mirror concentrators with a linear absorber tube [1], [2]. The first power plants of this type, still operational, were the nine Solar Electric Generating Systems (SEGS), constructed in California, USA, in the Mojave desert, starting from 1984, with electrical powers ranging from 13.8 to 80 MW, with a total power of 372 MW. Again, in the USA, other 150 MW of solar plants were installed. All these plants utilize synthetic oils a heat transfer fluid (HTF) and water steam as work fluid evolving in a Rankine cycle. The maximum temperature reached by synthetic oils cannot exceed the value of 400 °C, to avoid the degradation of the thermal properties of the fluid, and this limit reduces the thermodynamic efficiency of the power block.

At present, the world leader in the field of solar thermodynamics is Spain, where 37 parabolic through plants of 50 MW each are operational, among which are Andasol 1, Andasol 2, Andasol 3, Arcosol 50, Aste 1A, Aste 1B, Solnova 1, Solnova 3, Solnova 4, Extresol 1, Extresol 2, Extresol 3 and other plants [2], [5]. They all use synthetic oil as HTF and 17 of them are provided with thermal storage by molten salts filling two tanks: the hot tank at 390°C and the cold tank at 290°C. In the absence of sun radiation, the heat can be extracted from the hot tank, to heat the oil and produce water steam to feed the turbine. The autonomy of the storage system is between 7.5 and 9 hours.

The 5 MW Archimede plant [6] in Priolo Gargallo (SR) is operational in Italy. This plant utilizes a mixture of molten salts (60% of NaNO₃ and 40% of KNO₃) as HTF and as storage medium, with a maximum temperature of 550°C. The storage system, consisting of a hot tank at 550 °C and of a cold tank at 290°C, has an autonomy of 8 hours. The water steam obtained by the plant is added to that produced by a 130 MW fossil fuel combined cycle plant. The main problem existing in

this type of plant is the need to heat all the piping continuously to prevent the flowing salts from solidifying, which occurs for temperatures lower than 240 °C.

Other linear parabolic plants were built in Morocco, Algeria, Egypt and Thailand. The latter, a 5 MW plant, produces direct water steam at 340°C in the collectors.

Ferraro and Marinelli [7],[8] first proposed the use of atmospheric air as HTF in linear parabolic collectors and as work fluid in a turbogas engine.

The plant [7] can have the scheme presented in Fig. 1: the air is taken from the outside, compressed in an intercooled multistage compressor (in order to reduce the compression work), sent to the solar collectors field for its heating, sent to the turbine connected to the electrical generator and then discharged. Before discharging to the ambient, it is convenient to recover a part of its enthalpy, preheating the outgoing air from the compressor in the regenerator before being channelled to the solar field. The thermodynamic air cycle is an open-type Joule-Brayton cycle. In the work [8] another plant configuration was also considered, 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 solar energy and then it completes its expansion in the low pressure part of the turbine, see Fig. 2. It was found that in some circumstances, with high values of direct irradiance, this type of plant performs better than the former.

In both papers [7], [8] the fact that the selective cover of the receiver tube cannot withstand temperatures higher than 580 °C, owing to the deterioration of its radiative properties, was not considered, whereas, in reality, for high irradiance values, this temperature limit can be exceeded.

This paper presents more realistic calculation results, taking into account this temperature limit. Moreover, a wider parametric analysis than before was carried out, in which the performances of the plants were studied varying the airflow rate and location of the plant.

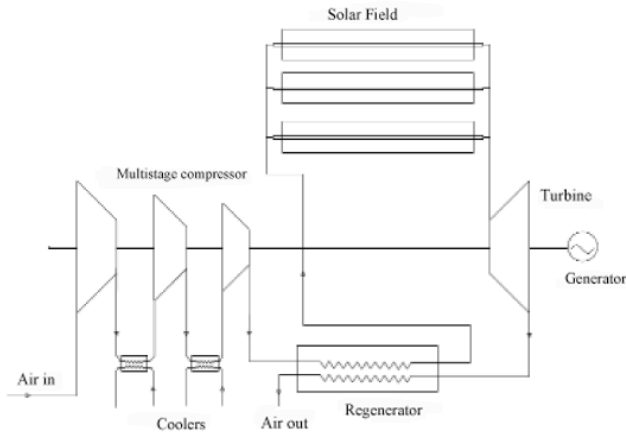


Fig. 1 - Scheme of the air turbine solar plant without reheating

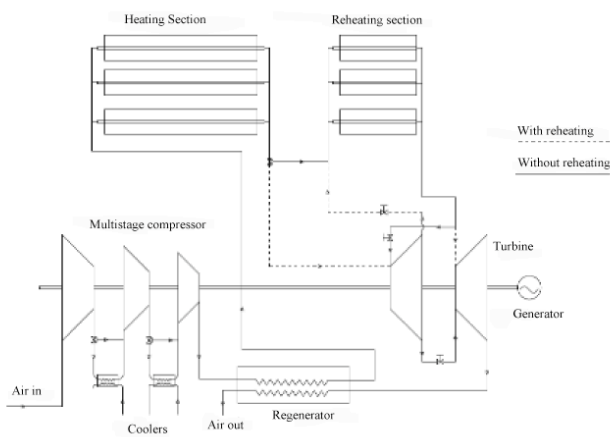


Fig. 2 - Scheme of the air turbine solar plant with reheating

2. MODEL OF PARABOLIC COLLECTORS

The authors [9] developed two calculation programs able to carry out a thermal analysis of linear parabolic collectors: the first for steady state conditions, named STS code, the second for transient conditions, named STT code. The two codes can consider oil, molten salts, carbon dioxide and air as heat transfer fluids. 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, the present work uses the stationary STS code.

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

$$P_{sol,abs} = I_{bn} \cdot IAM \cdot \eta_{opt} \cdot A_{col} \quad (1)$$

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 both the effect of the inclination of the direct irradiance on the plane of the collector (cosine effect) and the decreasing of the optical efficiency with the incidence angle. It is calculated by the equation [9]:

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

The normal optical efficiency is defined as:

$$\eta_{opt} = \rho \cdot \tau_{env} \cdot \alpha_{abs} \cdot \gamma \quad (3)$$

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.

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

$$P_u = m \cdot (h_u - h_i) \quad (4)$$

where 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:

$$\eta_{ter} = \frac{P_u}{P_{sol,abs}} \quad (5)$$

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

$$\eta_{col} = \frac{P_u}{I_{bn} \cdot A_{col} \cdot \cos i} \quad (6)$$

The geometrical and optical data of the collectors utilized are reported in ref.[9].

Generally, in the operation of thermodynamic solar plants, the fluid mass flow rate is continuously changed with the time variation of solar irradiance, in order to obtain a constant value of the fluid temperature at the outlet of the solar collectors during the day, in our air plants equal to 580 °C. This mode of functioning of the plant is called: “working mode at variable flow rate”.

The maximum temperature admitted by the absorber tubes considered in this work, produced by the Società Angelantoni [10] is in fact equal to 580 °C.

However, usually turbo compressors can usually work, maintaining acceptable values of the compressor and of the turbine isentropic efficiency, at flow rates not lower than 65-70% of nominal design flow rate, whereas the flow rate variations, varying the solar irradiance, can reach, for low values of irradiance, even 10% of the nominal flow rate. No reliable data on the performance of the turbo compressor in such severe out-of-design conditions were found in the literature, and in this paper, firstly, it was assumed in ideal behaviour mode with no deterioration of the efficiencies, to estimate the target performance of the engine in variable flow rate conditions and, secondly, a commercial program [11] was utilized in order to make a preliminary estimation of the behaviour of the engine in out of design conditions.

Moreover, a study was made of the performance of the air turbo compressor at constant flow rate conditions and variable temperatures at the outlet of the collectors. In this way the isentropic efficiencies remain at their maximum design values.

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.,

and utilizing the Data Fit software [12] developed by Oakdale Engineering, several correlations of global efficiency of the collector, of useful thermal power, of the outlet temperature of the fluid from the collectors and of other useful variables were obtained, as a function of the parameter x , defined as:

$$x = I_{bn} \cdot IAM \cdot \eta_{opt} \quad (7)$$

This parameter has the physical meaning of the solar energy absorbed at each instant from the receiver for unit area of collector aperture. It allows both consideration of the cosine effect and the variation of the optical efficiency with the incidence angle of radiation, while utilizing the same correlations.

All developed correlations have a correlation index of 99.99% and maximum errors lower than 0.5%.

For the working mode with variable flow rate

For a 100 m long collector, an ambient temperature of 15 °C, an inlet temperature T_i of 200 °C and an outlet temperature of 580 °C, the following correlations were obtained for the collector efficiency η_{col} , the useful thermal power P_u (kW) and the mass flow rate m (kg/s):

$$\eta_{col} = 0.653453 + 7.943053 \cdot 10^{-5} \cdot x - \frac{15987.6128}{x^2} \quad (8)$$

$$P_u = -228.683319 + 1.643157 \cdot 10^{-4} x^2 + 18.684934 \cdot x^{0.5} \quad (9)$$

$$m = -0.646798 + 3.339570 \cdot 10^{-10} \cdot x^3 + 5.248991 \cdot 10^{-2} \cdot x^{0.5} \quad (10)$$

The calculations showed that, in this working mode, the air temperature leaving the regenerator (with an efficiency of 0.85) and entering the collectors varies between 200 and 220 °C.

For the working mode with constant flow rate without reheating

For a 100 m long collector, an inlet temperature of 200 °C and a mass flow rate of 0.5 Kg/s, the following correlations were obtained:

$$\eta_{col} = -1.0266146 \cdot 10^{-12} \cdot x^4 + 2.474980 \cdot 10^{-9} \cdot x^3 - 2.161599 \cdot 10^{-6} x^2 + 4.932657 \cdot 10^{-4} \cdot x + 0.644348 \quad (11)$$

$$P_u = 6.805418 \cdot 10^{-8} \cdot x^3 - 3.3538825 \cdot 10^{-4} \cdot x^2 + 0.602784 \cdot x - 8.481684 \quad (12)$$

$$T_u = 1.550739 \cdot 10^{-7} \cdot x^3 - 6.958403 \cdot 10^{-4} \cdot x^2 + 1.154401 \cdot x + 185.4602 \quad (13)$$

T_u in eq. (13) is the outlet temperature of the fluid from the collectors.

Similar equations were obtained for a mass flow rate of 0.75 kg/s.

Fig. 3 shows the mass flow rate as a function of x , at constant outlet temperature of 580°C and variable flow rate for $T_i = 200^\circ\text{C}$ and 220°C . Obviously, in order to achieve the same outlet temperature, m has to be increased when the irradiance and the parameter x increase. A quasi-linear trend and a large variation of the flow rate with x can be noticed. In the interval 200-700 of x , the flow rate varies between about 0.1 kg/s and 0.9 kg/s per collector.

In Fig. 4 the global efficiency of the collector is plotted as a function of parameter x , at variable flow rate and the same

inlet temperatures. The efficiency increases with x because, increasing x , the flow rate increases and the heat transfer coefficient between wall and fluid increases, whereas the thermal heat losses, determined by the axial temperature profile along the absorber tube, remain almost constant.

Fig. 5 shows the trend of the outlet temperature of the fluid from the collector for the two cases of $m=0.5$ kg/s and $m=0.75$ kg/s, for an inlet temperature of 200°C. It can be observed that, for $m=0.5$ kg/s, the maximum allowable temperature of 580°C is reached for $x=455$, whereas, for $m=0.75$ kg/s, the same temperature is reached for $x=610$. Therefore, in the operation of the plant, it is necessary to control the absorber maximum temperature continuously and, when necessary, to act on the rotation angle of the parabola in order to reduce the value of the projected irradiance and to lower the temperature.

Fig. 6 shows the trend of the useful thermal power with x for the two values of mass flow rate.

In Fig. 7 the global efficiency of the collector is plotted as a function of parameter x , for the three cases of variable flow rate, of $m=0.5$ kg/s and of 0.75 kg/s.

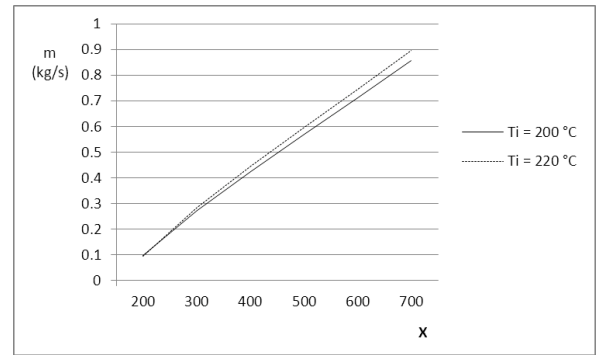


Fig. 3 - Mass flow rate as a function of x (variable flow rate)

This figure indicates that, in the case of constant mass flow rate, the efficiency decreases increasing x , because the outlet temperature of the fluid increases and the thermal losses along the receiver increase. In the same figure the efficiency of the collector at variable flow rate is also reported, for comparison purposes.

Other correlations similar to eqq. (11-13), valid for inlet fluid temperatures of 100, 300 and 400 °C, were developed. Once the actual value of the temperature at the inlet of the collector is known at a certain instant, the collector efficiency and the thermal power of the collector can be calculated by linear interpolation between the correlations.

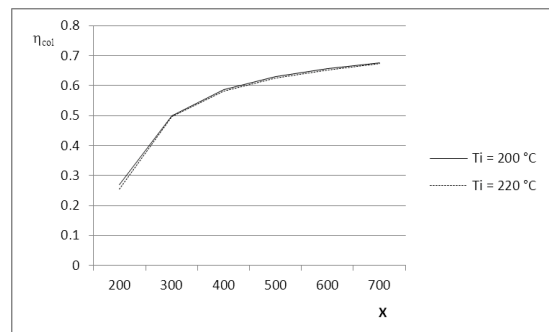


Fig. 4 - Collector efficiency as a function of x (variable flow rate)

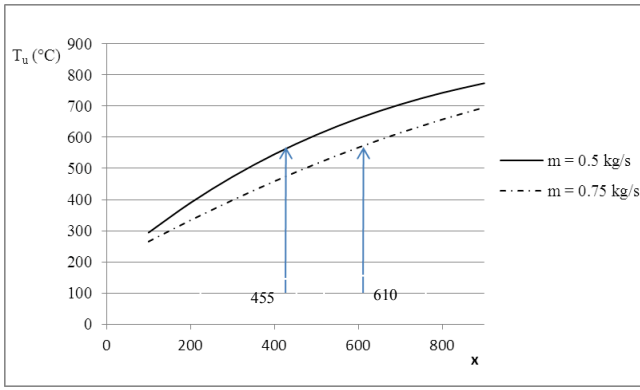


Fig. 5 - Collector outlet temperature as a function of x for $T_i=200\text{ }^\circ\text{C}$

This inlet temperature, however, is not known *a priori* because it is determined by the outlet temperature from the compressor, by the parameter x which influences the inlet temperature on the turbine, by the outlet temperature from the turbine and by the efficiency of the regenerator, see Fig. 1.

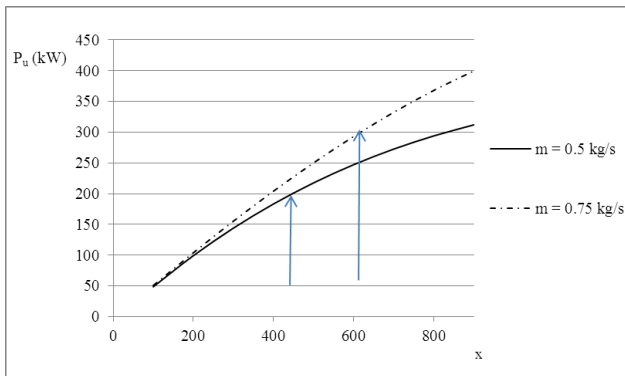


Fig. 6 - Collector useful thermal power as a function of x for $T_i=200\text{ }^\circ\text{C}$

For the working mode with constant flow rate with reheating
In this working mode, see Fig. 2, the fluid leaving the regenerator enters into the first section of the collector, then it expands in the first stage of the turbine, successively it leaves this high pressure stage and enters into the second section of the collector.

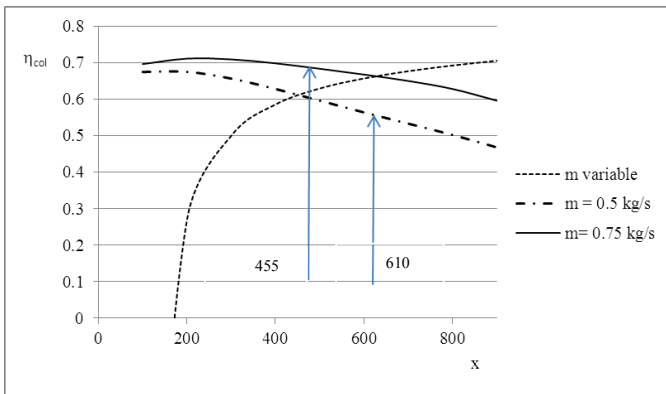


Fig. 7 - Collector efficiency as a function of x for $T_i=200\text{ }^\circ\text{C}$

Once reheated, the air expands in the second stage of the turbine and then enters into the regenerator. Opening and closing some valves in the plant in a suitable way, it can work without reheating or with reheating, see Fig. 2, according to the irradiance conditions.

The heating section was assumed to be 67 m long and the reheating section 33 m long.

For an inlet temperature to the collector $T_i=200\text{ }^\circ\text{C}$ and $m=0.5\text{ kg/s}$, the following correlations were developed for the whole collector efficiency and the thermal useful powers P_{u1} and P_{u2} (kW) obtained in the two sections of the collector:

$$\eta_{col} = 0.886830 - 3.091395 \cdot 10^{-4} \cdot x - \frac{1.611946}{x^{0.5}} \quad (14)$$

$$P_{u1} = -2.398194 + 0.365653 \cdot x - 2.692588 \cdot 10^{-6} \cdot x^{2.5} \quad (15)$$

$$P_{u2} = -1.369537 + 0.184642 \cdot x - 4.160483 \cdot 10^{-7} \cdot x^{2.5} \quad (16)$$

In Fig. 8 the collector efficiency with reheating is compared with the efficiency without reheating, for $m=0.5\text{ kg/s}$. The figure shows that the reheating has a beneficial effect on the collector efficiency.

Fig. 9 reports the trends of the air temperature at the outlet of the first (T_{u1}) and of the second section of the collector (T_{u2}), compared with the trend of the outlet air temperature in the case of functioning without reheating, for $m=0.5\text{ kg/s}$. In the latter case the outlet temperatures are higher.

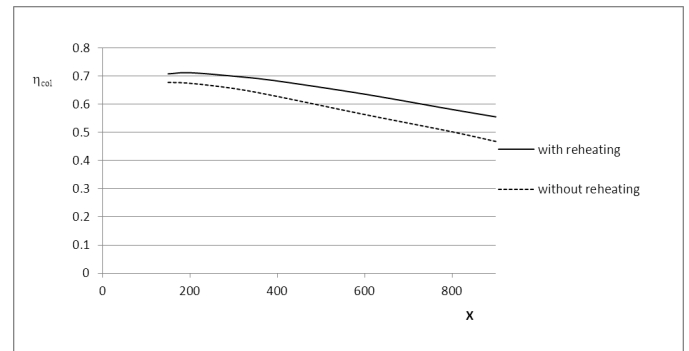


Fig. 8 - Collector efficiency as a function of x for $m = 0.5\text{ kg/s}$ and $T_i = 200\text{ }^\circ\text{C}$.

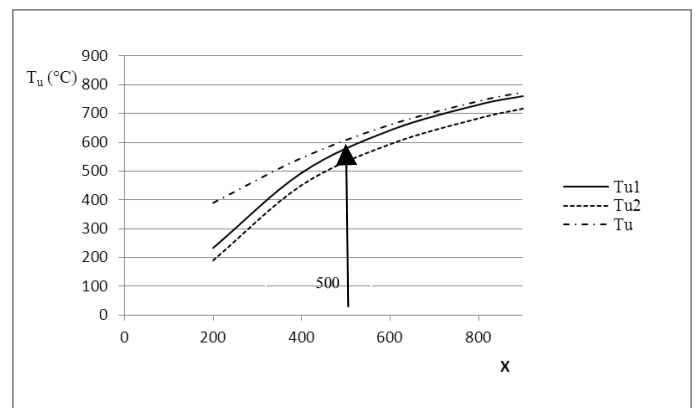


Fig. 9 - Collector outlet temperatures as a function of x with and without reheating for $m=0.5\text{ kg/s}$ and $T_i = 200\text{ }^\circ\text{C}$.

3. MODEL OF THE TURBOAIR ENGINE

The performance of the turboair plant working with variable flow rate, with constant flow rate without reheating and with constant flow rate and reheating were analysed, by means the electronic sheet TURBOARIA, developed by the authors [8]. The main input data to the sheet are: inlet air temperature to the compressor, inlet temperature to the turbine, pressure ratios of the compressor and of the turbine, polytropic efficiencies of the compressor and of the turbine (the isentropic efficiencies are functions of the polytropic efficiencies and vice versa), mass flow rate and others. In the implemented model the dependence on the temperature of the specific heats at constant pressure and constant volume of the air was considered. It was assumed that the thermodynamic transformations of the air take place along polytropic curves with exponent n variable with temperature and the compressor and the turbine were subdivided into 18 axial steps. This programme calculates, among other quantities, the thermodynamic efficiency of the gas turbine plant. In ref. [8] there is a lot of information on it.

One of the important advantages of the proposed plant is that no water is used in the process, and also the intercooling of the compressor may be accomplished by the ambient air: this is particularly useful if the plant is located in an arid location. A compressor pressure ratio $\beta_c=9$ was assumed, and to take into account the pressure drops of the fluid in the collectors in a conservative manner, a pressure ratio $\beta_t=8.5$ was assumed in the turbine. Two intercoolers were considered (see Figs. 1 and 2) and a regeneration Nusselt efficiency of 0.85 was assumed in the calculations. This is an input parameter to the TURBOARIA sheet. In reference [7] a discussion is conducted on the influence of various parameters on the plant efficiency.

For the working mode with variable flow rate

In this plant operating mode the turboair efficiency is only a function of the ambient air temperature and with two intercooling and regeneration it can be calculated by the correlation:

$$\eta_{\text{turb}} = 0.387021 - 1.834150 \cdot 10^{-3} \cdot T_a + 2.357514 \cdot 10^{-4} \cdot T_a^{0.5} \quad (17)$$

For the working mode with constant flow rate, without reheating

For $m=0.5$ kg/s, an ambient air temperature of 15 °C and $x < 375$ (in this case it is better not to use intercooling and regeneration), the following correlations were obtained:

$$\eta_{\text{turb}} = 0.300267 + \frac{3.518446}{x^{0.5}} - \frac{65.7991}{x} \quad (18)$$

If $x > 375$ (in this case it is convenient to have intercooling and regeneration):

$$\eta_{\text{turb}} = 0.453611 + \frac{1357.61}{x^{1.5}} - \frac{46655.7}{x^2} \quad (19)$$

The inlet temperature to the collector, equal to the outlet temperature from the regenerator T_R , is:

$$T_R = 300.258 + 2.778926 \cdot 10^{-8} \cdot x^3 - \frac{1799339}{x^2} \quad (20)$$

For the working mode with constant flow rate with reheating

For $m=0.5$ kg/s, an ambient air temperature of 15 °C and

$x < 300$ (no intercooling and no regeneration), the following correlations apply:

$$\eta_{\text{turb}} = 0.336968 + 9.147305 \cdot 10^{-8} \cdot x^2 - \frac{538.3246}{x^{1.5}} \quad (21)$$

$$T_1 = 263.1 + 1.906666 \cdot T_a \quad (22)$$

For $x > 300$ (intercooling and regeneration)

$$\eta_{\text{turb}} = 0.582162 - \frac{30.2771}{x} - \frac{1159.46}{x^{1.5}} \quad (23)$$

$$T_{i1} = 847.52 - \frac{12381.37}{x^{0.5}} + \frac{4631453.23}{x^2} \quad (24)$$

$$T_{i2} = 1050.07 - \frac{14963.84}{x^{0.5}} + \frac{5208164.73}{x^2} \quad (25)$$

In eq. (24) and eq. (25), T_{i1} and T_{i2} are the temperatures at the inlet of the two collector sections. Other similar correlations were developed for other ambient air temperatures. By linear interpolation it is possible to calculate the turboair efficiency for any air temperature. Other values of flow rate were also considered. In Fig. 10, the turboair engine thermodynamic efficiency trend as a function of the ambient air is shown, with 2 intercooling and regeneration, at variable

mass flow rate. Obviously, a reduction of the efficiency increasing the ambient temperature can be observed, owing to the increasing of the compression work. Fig. 11 shows, for an ambient air temperature of 15°C, the turboair engine efficiency trend as a function of the parameter x , at a constant flow rate of 0.5 kg/s, in the two cases of heating only (curves 1 and 2) and heating and reheating (curves 3 and 4). Curves 1 and 3 refer to working without intercooling and regeneration.

A constant efficiency applies for the working mode with variable flow rate. From the graph it is evident that the larger values of turbine efficiency are obtained operating the plant, for low values of x , along curve 1 (that is, with no intercooling, no regeneration and heating only) and then along curve 4 (use of intercooling, regeneration and reheating). The most important comparison of all the working modes can be made referring to the whole plant efficiency, equal to the product of the collector efficiency times the turboair efficiency, if the electrical generator efficiency is excluded.

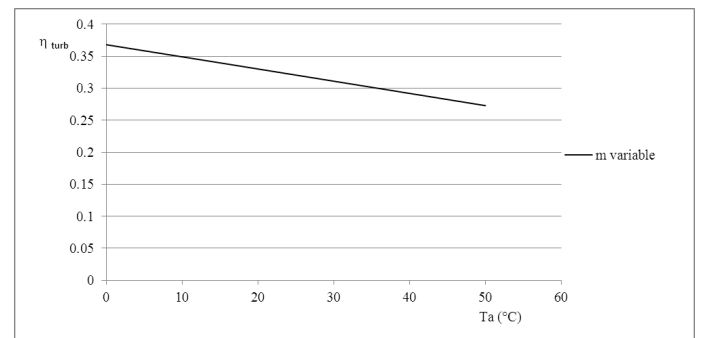


Fig. 10 - Turbine efficiency as a function of ambient air temperature

The efficiency of the solar collectors improves at variable flow rate, increasing the irradiance, whereas the turboair efficiency remains constant, so the plant efficiency increases (in this case, obviously, intercooling and regeneration have to be always activated). At constant flow rate, increasing the

irradiance, the efficiency of solar collector decreases, whereas the turboair efficiency increases: therefore, the plant efficiency presents a maximum value for a certain value of x .

Fig. 12 shows, for the same pressure ratios, and an ambient air temperature of 15°C , the plant efficiency trend as a function of the parameter x , at a constant flow rate of 0.5 kg/s , in the two cases of only heating (curves 1 and 2) and reheating (curves 3 and 4). The figure shows a strong improvement of the efficiency if reheating is used, and that, for low values of the parameter x , it is convenient to operate along curve 1 (no intercooling and regeneration and no reheating) instead of along curve 3 (no intercooling and regeneration, with reheating).

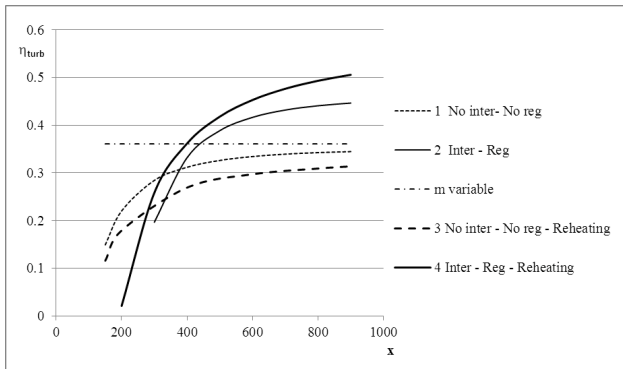


Fig. 11 - Turbine efficiency as a function of x with and without intercooling, regeneration and reheating for $m=0.5\text{ Kg/s}$ and $T_a = 15^{\circ}\text{C}$.

4. ESTIMATION OF THE YEARLY PERFORMANCE OF THE PLANT

All the above correlations were inserted in the electronic sheet VIVASOL, developed by the authors [8], in which the hourly values of normal direct irradiance and air temperature are furnished as input data for the localities considered. This sheet can be used to calculate the hourly values of the collector, engine and plant efficiency, the useful power, the electrical power and, for integration, the annual value of electrical energy produced by the plant and the average annual efficiencies.

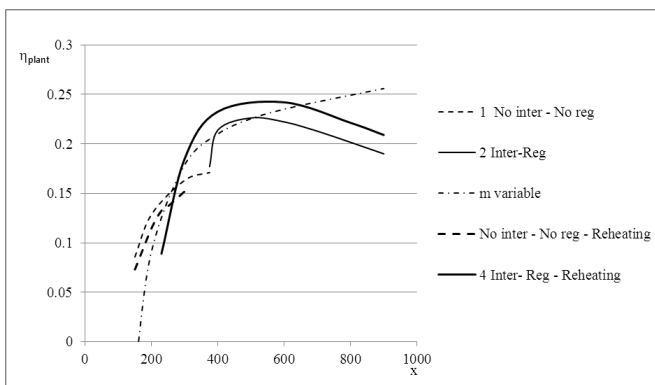


Fig. 12 - Plant efficiency as a function of x with and without intercooling, regeneration and reheating for $m=0.5\text{ Kg/s}$ and $T_a = 15^{\circ}\text{C}$.

In all calculations a constant value of 0.98 for the electromechanical efficiency of the alternator was

hypothesized.

Since plant performance is greatly influenced by the available direct irradiance [13],[14], some Italian localities with different climatic conditions, a Spanish locality and an American locality were considered. The chosen localities are: Milan, Rome, Crotona, Palermo, Priolo Gargallo (where the Archimede plant is located), Almeria (Spain) and Dagget (USA, California). In Table 1 the performance data (annual electrical energy, average collector, turbine and plant efficiencies and average electrical power) are reported for a plant operating at constant flow rate of 0.5 kg/s per collector, located in the different localities. The electrical energy produced is strongly influenced by the climatic data. In Italy, the largest value of electrical energy, among the chosen localities, is obtained in Crotona (159 MWh/year), whereas in Almeria, with respect to Crotona, an increase of electrical energy of 15% is obtained and in Daggett the increase is 32%. The average annual efficiency of the plant is everywhere close to 15%, with the exception of Milan, where it is 13%. The average efficiency of the collectors is close to 50% and the average efficiency of the turboair engine is variable from 27% for Milan to 32.5% for Almeria. Table 2 shows the influence of the mass flow rate for Crotona and Almeria. Varying the flow rate from 0.5 kg/s to 0.75 kg/s , for Crotona, the annual electrical energy varies between 159 and 176.7 MWh/year, with an increment of 11%; for Almeria, the electrical energy changes from 183.3 to 198.9, with an increase of 8.5%. The plant efficiency varies from 0.153 to 0.171 for Crotona and from 0.157 to 0.171 for Almeria. For the working mode at variable flow rate, the calculations performed by the commercial software [11] indicate an electrical energy of 207.9 MWh for Almeria (8 % lower than the maximum ideal value of 225.5 MWh), 4.5% higher than the value of 198.9 obtained by a constant flow rate of 0.75 kg/s . For Crotona, the electrical energy produced at variable flow rate is 180.5 MWh (9 % lower than the ideal value of 197.9); it is higher by 2.1% than the energy obtained with $m=0.75\text{ kg/s}$. Table 3 shows the influence of reheating on the performance of the plant, varying the flow rate between 0.4 kg/s and 0.75 kg/s , for Crotona and Almeria. Comparing the values reported in this Table with those reported in Table 4, valid for heating only, at the same flow rate, it can be observed that, for $m=0.5\text{ kg/s}$, reheating increases the electrical energy by 3% in Crotona and by 5.2% in Almeria. For $m=0.75\text{ kg/s}$, reheating produces an increase of 2.4% in Crotona and almost the same value of energy in Almeria. Reheating at $m=0.4\text{ kg/s}$, with respect to heating at $m=0.5\text{ kg/s}$, produces instead, in Crotona, a 17.2% increase of the electrical energy and 23% in Almeria. Comparing the electrical energy production with reheating and $m=0.4\text{ Kg/s}$ with production with heating and $m=0.75\text{ kg/s}$, an increase of 5.5% is achieved in Crotona, and of 13.4 in Almeria. In conclusion, in Crotona could be preferable to use heating only with $m=0.75\text{ kg/s}$, owing to the greater simplicity of the plant, whereas in Almeria it is better to use reheating with $m=0.4\text{ kg/s}$. A reference plant of gross nominal electrical power of 50 MW, located in Crotona, was then considered. To size the collector field, a design DNI equal to 800 W/m^2 was assumed. Supposing to make the calculation at 12 hours solar time of June 21th, one obtains an incidence angle $i=15.6^{\circ}$, $\cos i=0.964$, $\text{IAM}=0.949$ and the value of 598.3 for the design parameter x .

Table 1 - Plant yearly performance per collector

Locality	Direct norm. Irrad. DNI (kWh/m ² year)	Mode of working	Electrical Energy (MWh/year)	η_{col} (-)	η_{turb} (-)	η_p (-)	P_{el} (kW)
		L=100 m heating					
Milano	1372	m=0.5 kg/s	88.5	0.499	0.272	0.133	42,85
Roma	1757	m=0.5 kg/s	130.4	0.494	0.309	0.150	45.8
Crotone	2077	m=0.5 kg/s	159	0.491	0.319	0.153	49.88
Palermo	1897	m=0.5 kg/s	145.4	0.499	0.313	0.153	47.81
Priolo Gargallo	2038	m=0.5 kg/s	156.6	0.488	0.317	0.152	49.1
Almeria (Spain)	2308	m=0.5 kg/s	182.8	0.493	0.325	0.157	53.70
Dagget (USA)	2791	m=0.5 kg/s	209.6	0.467	0.323	0.148	54.38

Table 2 - Influence of mass flow rate on the plant yearly performance per collector

Locality	Mode of working	Electrical Energy (MWh/year)	η_{col} (-)	η_{turb} (-)	η_p (-)	P_{el} (kW)
Almeria	L=100 m heating					
	m=0.5 kg/s	183.3	0.493	0.326	0.157	53.70
	m=0.6 kg/s	182.7	0.531	0.301	0.157	53.73
	m=0.75 kg/s	198.9	0.585	0.297	0.171	59.75
	m variable (ideal turbine)	225.5	0.613	0.322	0.193	71.92
	m variable (com. software)	207.9	0.614	0.297	0.178	66.02
Crotone	L=100 m heating					
	m=0.5 kg/s	159	0.491	0.319	0.153	49.9
	m=0.6 kg/s	160.6	0.535	0.296	0.155	50.56
	m=0.75 kg/s	176.7	0.592	0.293	0.171	57.03
	m variable (ideal turbine)	197.9	0.597	0.326	0.191	62.74
	m variable (com. software)	180.5	0.597	0.297	0.174	63.13

Table 3 - Influence of reheating on the plant yearly performance per collector

Locality	Mode of working	Electrical Energy (MWh/year)	η_{col} (-)	η_{turb} (-)	η_p (-)	P_{el} (kW)
Almeria	L=67/33 m reheating					
	m=0.4 kg/s	225.6	0.514	0.385	0.193	65.93
	m=0.5 kg/s	192.84	0.526	0.321	0.165	57.78
	m=0.75 kg/s	199.2	0.548	0.331	0.171	60.32
Crotone	L=67/33 m reheating					
	m=0.4 kg/s	186.37	0.486	0.377	0.18	62.56
	m=0.5 kg/s	163.77	0.512	0.315	0.158	52.98
	m=0.75 kg/s	175.2	0.550	0.313	0.169	57.22

Table 4 - Yearly electrical production and average electrical power

Locality	Mode of working	Mass flow rate (kg/s)	Electrical Energy (MWh/year)	P_{el} (MW)
Almeria	L=100 m	variable (software)	114345	36.31
	L=100 m	m=0.5	100815	29.53
	L=100 m	m=0.75	109422	32.86
	L=67/33 m	m=0.4	124080	36.26
	L=67/33 m	m=0.75	109560	33.17
Crotone	L=100 m	variable (software)	99275	34.72
	L=100 m	m=0.5	87450	27.44
	L=100 m	m=0.75	97168	31.36
	L=67/33 m	m=0.4	102503	34.40
	L=67/33 m	m=0.75	96360	31.47

The total area of the collectors $A_{t,col}$ is calculable by the relation:

$$A_{t,col} = \frac{P_e}{DNI \cdot \cos i \cdot \eta_{col} \cdot \eta_{turb} \cdot \eta_a} \quad (26)$$

Inserting in eq. (26) the values $P_e=50.10^6$ W, $DNI=800$, $\cos i=0.964$, and the efficiencies obtained by VIVASOL, equal, in the case of variable flow rate, to $\eta_{col}=0.641$, $\eta_{turb}=0.326$, $\eta_a=0.98$, an area of collectors $A_{t,col}=316593$ m² is obtained.

Each collector having an area of 576 m² (100 m x 5.76 m), 550 parabolic collectors are necessary. The design mass flow rate results to be $m=0.741$ kg/s. Each collector having an area of 576 m² (100 m x 5.76 m), 550 parabolic collectors are necessary. The design mass flow rate results to be $m=0.741$ kg/s. In the case of constant flow rate, with $m=0.75$ kg/s, one obtains instead $\eta_{col}=0.708$, $\eta_{turb}=0.338$, $\eta_a=0.98$, $A_{t,col}=270927$ m² and 470 collectors.

The size of the collector field is therefore determined by the plant operating mode. Table 5 reports the values of annual electrical energy production (MWh/year) and of annual average power (MW) of a plant with 550 collectors located in Crotona and in Almeria, operating the plant at variable flow rate and at constant flow rate with and without reheating. Looking at the table, one observes that the maximum electrical production of 124080 MWh/year is obtained in Almeria with reheating and a mass flow rate $m=0.4$ kg/s; in Crotona, this maximum value is of 102500 MWh/year. Reheating is therefore more convenient for Almeria, where there is a direct irradiance 12% higher than that of Crotona.

CONCLUSIONS

This work presents a model able to evaluate the hourly and yearly performances of thermodynamic solar plants provided with parabolic linear collectors, utilizing atmospheric air as heat transfer fluid as well as working fluid in an open-type Brayton-Joule cycle. The plants were studied in two operating modes: at variable flow rate and constant temperature at the outlet of the collectors, and at constant flow rate and variable outlet temperature. The influence of some parameters such as direct irradiance, mass flow rate and collector length on the performance of the plant was studied. Moreover, a variant of the plant was also considered, in which the fluid is reheated after a partial expansion in the turbine. In all calculations the limit temperature of 580°C for the air flowing in the collectors was respected, in order to prevent the deterioration of the radiative properties of the absorbing tubes. Table 5 reports the values of annual electrical energy. The results obtained demonstrate a very good performance by this type of plant, which utilizes the ambient air in place of the expensive and more problematic fluids such as synthetic oils and molten salts used in already constructed plants; it is very simple from the constructional point of view and does not need any water because the working fluid in the engine is the air and the intercooling of the compressor can also be done by atmospheric air. The absence of water makes this plant very attractive for its installation also in arid regions. Moreover, if a larger and more constant production of electrical energy is needed, this plant can be very easily hybridized by adding a fuel combustion chamber.

The performances of these plants are comparable and sometimes even superior to those of the Spanish oil and water steam plants with Rankine cycle, which present average yearly global efficiencies near 16% [15], against the 18-19% achieved by our plants. Use of the reheating can be convenient in some localities, since it increases the average yearly efficiency of the plant.

A prototype turboair plant should be constructed in order to test its real performance in the field.

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