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Thermal energy losses during night, warm-up and full-operation periods of a CSP solar field using thermal oil

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Abstract

In Concentrating Solar Power (CSP) systems, solar radiation allows to keep the Heat Transfer Fluid (HTF) at the design temperature (250-400 °C using thermal oil) during daylight. During night, the thermal losses of the receiver tubes lead to a fast reduction of this temperature. The very first hours of daily solar irradiance are used to warm-up the Solar Field to the nominal temperature. This work focuses on a detailed analysis of the thermal losses of a 8,400 m^2 Solar Field based on Linear Fresnel Collectors (LFC) using thermal oil as Heat Transfer Fluid. The proposed simulation model evaluates the performance of the Solar Field as a function of solar radiation, solar position, ambient temperature and wind speed for given values of the main geometrical and technical characteristics of the SF components (insulated piping and solar receivers), as well as for assigned thermodynamic properties of the Heat Transfer Fluid. The time-step considered (1 second) and the dense spatial discretization chosen allow the energy-balance-equation-based model to be suited to simulate night, warm-up and full-operation phases.

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Keywords: Concentrating solar power; Thermal losses; Linear Fresnel Collectors; Night; Warm up; Transients.

1. Introduction

Today 4.4 GW of CSP plants are operational worldwide, 1.4 GW are under construction and 4.3 GW are under development [1].Installing one megawatt of CSP avoids the emission of hundreds ofCO_2tonsper year compared to a combined cycle system.In many countries (and in Italy, in particular) the wide extension areas required by large-size CSP plants are very hard to find. In this case, medium and small-size CSP plants may be a suitable option due to the lower land requirement. The plant configuration of

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medium-size CSP plants differ from that of large-size plants. In particular, for power outputs in the range of 1 MWe, steam Rankine cycles are often replaced by Organic Rankine Cycles (ORC), where steam is substituted by organic fluids with high molar weight [2]. Moreover, since ORC systems require thermal energy inputs with temperature levels in the range of 250-350 °C, LFC may be a viable option. Finally, for medium-size CSP units the most suitable option for the Thermal Energy Storage(TES) section is based on a two-tank direct system using thermal oil as HTF and storage medium [3-4].The analysis developed in this work was carried out with reference to the CSP section of the pilot solar power plant currently under construction in Ottana (Sardinia-Italy). The facility is based on a 600 kWe CSP plant with a two-tank direct TES system.

1.1. Concentrating Solar Power Plant layout

Fig.1(a) shows a simplified diagram of the CSP plant considered in this paper. It includes the Solar Field (SF), the Power Block (PB) and the TES. The thermal energy produced by the SF is used in the ORC unit to heat and vaporize the organic fluid. After the condenser, the organic fluid is compressed and then preheated in the regenerator. The condensing heat is removed by dry coolers due to the lack of cooling water. The TES section is based on a two-tank direct system, which includes a low-temperature (150 °C) and a high-temperature (260 °C) storage tank. The HTF from the cold tank flows through the SF, where it is heated and sent to the hot tank. When required, the hot thermal oil is pumped to the Power Block, based on a ORC unit, where it is cooled and sent back to the cold storage tank.

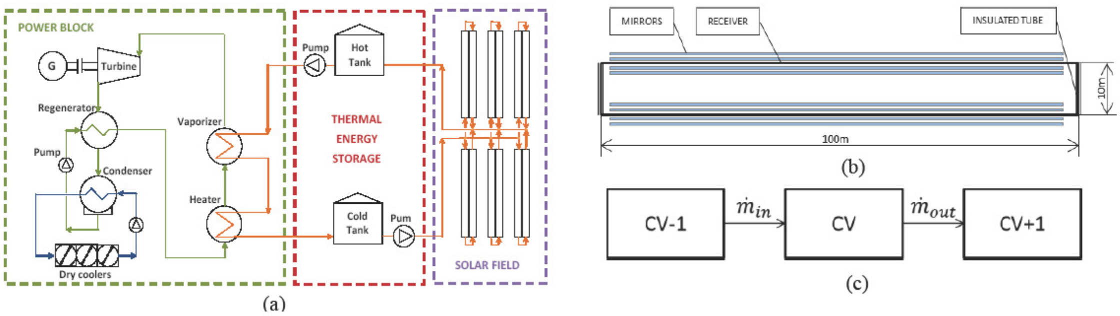


Fig. 1. (a) CSP plant layout; (b) SF loop; (c) SF loop control volumes

1.2. Solar field configuration

The SF is based on six lines of 200 m each of Linear Fresnel Collectors connected in parallel. The SF layout chosen is the center feed configuration, in order to minimize piping. Each collector line includes several collector modules, which in turn are connected in series, and each collector module is composed of several rows of flat mirrors. The mirrors are placed at about 0.5m above the ground and concentrate radiation onto a fixed receiver installed 4.9m above the mirror plane. The receiver includes a secondary reflector (reference optical efficiency is 62%). The solar collector lines are aligned along the North-South direction and are equipped with a single-axis tracking system. The pipeline (main: DN150)is composed of steel tubes with 8 cm of mineral wool insulation and an external protective aluminum foil. For this work the analysis was carried out considering one out of the six loops of the SF (Fig.1(b)).

2. System modeling and assumption

2.1. Mathematical model

The following energy balance equation, expression of the first law of thermodynamics for an open system, is the base of the simulation model proposed herein:

$$\frac{d\mathrm{E_{CV}}}{dt} + \dot{m}_{in}\left(h_{in} + \frac{c_{in}^2}{2} + gz_{in}\right) - \dot{m}_{out}\left(h_{out} + \frac{c_{out}^2}{2} + gz_{out}\right) = \dot{Q} + \dot{W} \quad (1)$$

Where the first term represents the time variation of the control volume total energy. The second and third terms of (1) are referred to the energy related to the mass transport. In this case, $\dot{m}_{in} = \dot{m}_{out}$, $z_{in} = z_{out}$ and $c_{in} = c_{out}$. The enthalpy values h_{in} and h_{out} are calculated considering a specific heat that is evaluated as a function of the mean temperature between the considered and the reference temperature of 25 °C. Pump work is neglected and thus $\dot{W} = 0$ in (1). The term $\dot{Q}$ takes into account the energy collected by the receiver $\dot{Q}_{RCV}$ and coming from the sun, the energy losses due to thermal exchanges with the environment ($\dot{Q}_{L,receiver}$, $\dot{Q}_{L,tube}$). While internal convection has been neglected in this work, internal conduction between adjacent control volumes has been considered. The equation (1) has been solved for each control volume associated to a length of one meter represented in Fig.1(c) and with a discretization time step of one second. The thermal power collected is evaluated by means of the following equation:

$$\dot{Q}_{RCV} = A_c DNI \eta_{opt,r} IAM_L \cdot IAM_T \eta_{CLN} \quad (2)$$

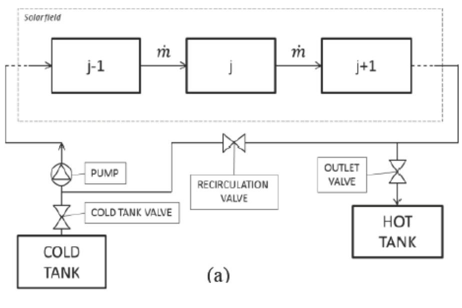
where A_C is the collecting area, $\eta_{opt,r}$ is the reference optical efficiency, IAM_L and IAM_T are the longitudinal and transversal components of the Incidence Angle Modifier (IAM) and η_{CLN} is the surfaces cleanliness factor. The two IAM components considered in this study were assumed starting from manufacturer information [5-6].The thermal losses $\dot{Q}_{L,receiver}$ are evaluated in function of the temperature difference ΔT_R between oil and air.

$$\dot{Q}_{L,receiver} = (a_1 \cdot \Delta T_R + a_2 \cdot \Delta T_R^4) \cdot A_C \quad (3)$$

where a_1 and a_2 are specific coefficients[7] (a_1=10.44 kW/(m^2K); a_2= 4.8 10-7 kW/(m^2K^4)). The thermal losses $\dot{Q}_{L,tube}$ are also evaluated in function of the temperature difference ΔT_t between oil and air considering an overall heat transfer coefficient. The heat transfer coefficient was evaluated taking into account the heat convection between oil and internal tube wall, heat conduction through the tube wall and the insulation layer and the heat convection from the outer surface to the ambient air [7].

2.2. Management strategy

Basically, three different operating conditions have been analyzed: night, warm-up and full-operation.



(a)

Operating mode	Pump	Cold tank valve	Recirculation valve	Outlet valve
Night	OFF	CLOSED	CLOSED	CLOSED
Warm-up	ON	CLOSED	OPEN	CLOSED
Full operation	ON	OPEN	CLOSED	OPEN

(b)

Fig. 2. (a) SF layout; (b) Table of components status depending on operating mode

ring the **night**, solar radiation is missing and the oil contained in the insulated tubes and receivers is still at high temperature. The initial temperature profile of the SF loop during night, equal to the last temperature profile achieved via sunlight warming, cools down to the environment temperature due to radiative losses. The oil temperature of the SF is influenced also by convective and conductive thermal exchanges within the oil. The **warm-up mode** is used in any case the SF has to be warmed up to the nominal conditions of temperature. This mode not only includes the sunrise warm-up, but also during-the-day warm-ups needed after clouds of long duration. In this operating mode, themain pumpis turned on and the HTF mass flow recirculates through the SF until it reaches the nominal/desired temperature (cold tank valve and outlet valve are closed). Once this condition is fulfilled, the cold tank valve and the outlet valve can be opened. Once the SF design temperature is reached, the **full operation mode** can be started. The thermal oil is extracted from the cold tank, it passes through the SF and then it is sent to the hot tank.

3. Results

During long absences of solar radiation, the oil mass contained in the SF is stationary. Considering an initial condition as the last temperature profile reached in nominal conditions, for an ambient temperature of 20°C the entire **cooling** process to reach the environment temperature lasts less than 4 hours.Fig.3(a) shows the average heat loss of the receiver and of the tube across time. The total energy loss due to night cooling is 505.6 MJ related to receiver losses and 13.1 MJ related to tube losses.

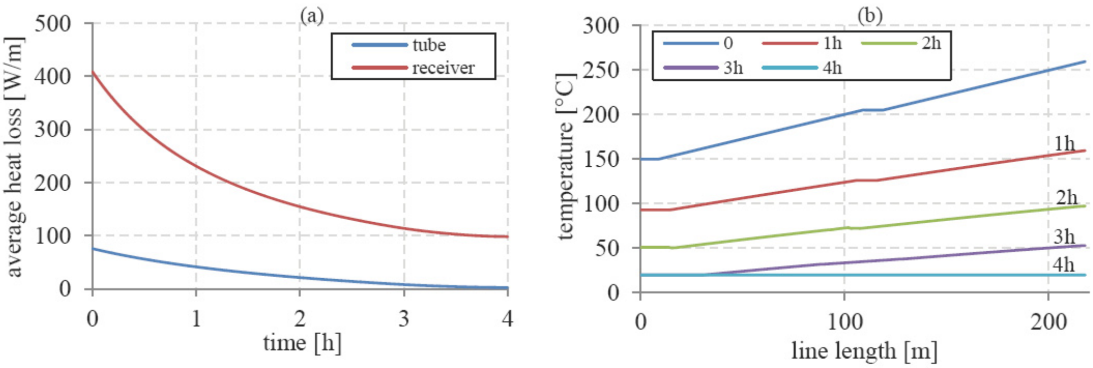


Fig. 3. (a) Average heat loss during cooling; (b) Temperature distribution along the SF line during cooling

Fig.3(b)shows the temperature distribution along the SF loop during the cooling process. The initial condition (t=0) is the last temperature profile reached in nominal conditions, and then the temperature gradually drops along the typical loop. During **warm-up mode**, the HTF mass flow recirculates through the SF until it reaches the nominal temperature. Fig.4(a) shows the time necessary to reach the nominal temperature of 150°C starting from an oil temperature assumed to be equal to an average ambient temperature of 20°C for two different days of the year (representative of a spring and a winter day) characterized by the different DNI curves displayed in Fig.4(b).

For both cases, the recirculation mass flow is assumed to be 2 kg/s. During a typical spring day, a value of DNI of 700 W/m^{2}is reached after 2 hours from sunrise, and after only one hour, the DNI value is already above 400 W/m^2. In this case, the outlet temperature rapidly rises and after 45 minutes it reaches the nominal temperature. During the typical winter day analyzed even after 2 hours the DNI value is under 200 W/m^2 and so the outlet design temperature is reached after approximately 2 hours and a half. Once the design temperature is reached, the recirculating pumps are turned off and the full operation mode can begin.

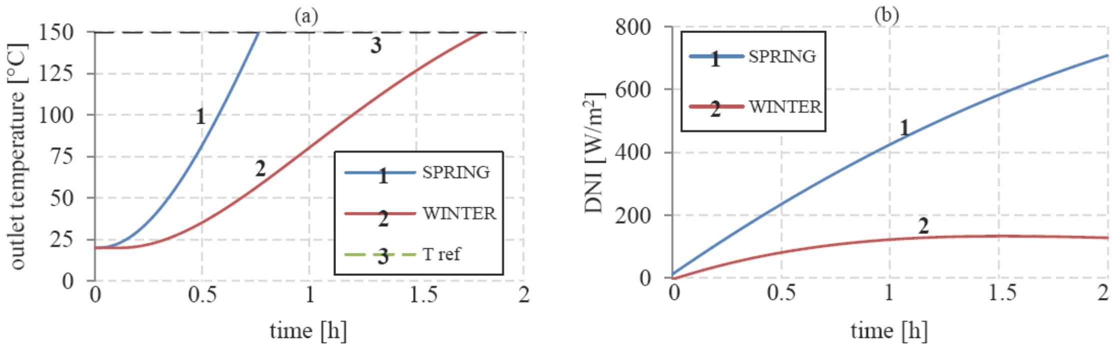


Fig. 4. (a) Time necessary to reach the nominal temperature; (b) DNI curves

During **full operation mode**, the control system purpose is to achieve always the nominal output temperature. In order to do so, the SF inlet valve regulates the HTF mass flow that enters and circulates through the lines depending on the output temperature. The outlet temperature rises depending on the available DNI and on the HTF speed (mass flow). In Fig.5(a) is shown the output temperature behavior depending on the mass flow for different values of DNI. The output temperature displayed is the one reached after the travel time related to the mass flow considered.

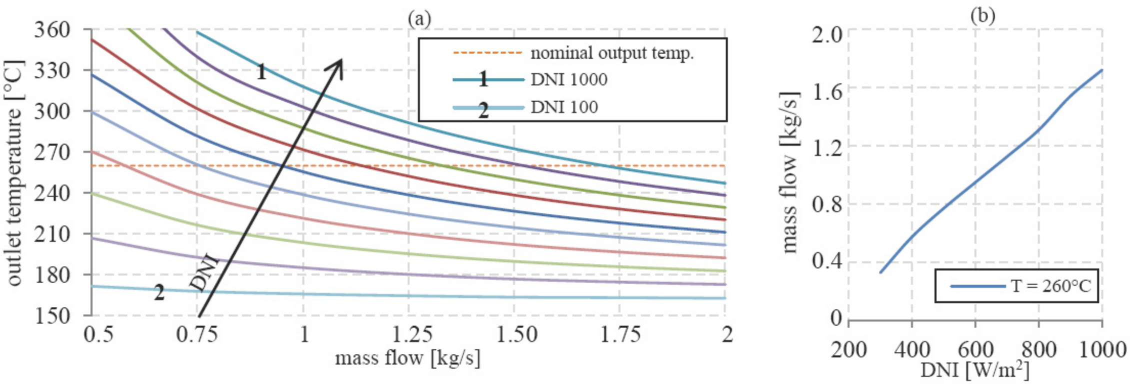


Fig. 5. (a) Outlet temperature depending on the mass flow and DNI; (b) Mass flow required

For the lower values of DNI, the curves slope is flat. This means that when the solar radiation is below 150-200 W/m^2, the SF outlet temperature dependence on the mass flow rate is negligible. For example, considering a DNI of 100 W/m^2, the SF outlet temperature is little above 170 °C for a mass flow of 0.5 kg/s and still above 160 °C for a mass flow of 2 kg/s. On the contrary, for the higher values of solar radiation, the mass flow strongly influences the SF outlet temperature. For instance, considering a DNI of 800 W/m^2, the SF outlet temperature is above 320 °C for a mass flow of 0.75 kg/s and 230 °C for a mass flow of 2 kg/s. Still referring to this graph, the operating field to prevent thermal oil to deteriorate can be seen: in fact, commercial thermal oils have a temperature upper limit in the range of 260-390°C. Considering that the design outlet temperature of the SF for this paper is 260 °C, the Fig.5(b)shows the mass flow required as a function of the available solar radiation in order to obtain the design outlet temperature. For DNI values below 300 W/m^2, the design temperature can never be reached for any value of mass flow due to thermal losses. Since solar thermal power plants source of energy is the sun, and the direct normal irradiance strongly varies depending on clouds, it is very important to analyze what happens to the SF temperature during cloud transients.

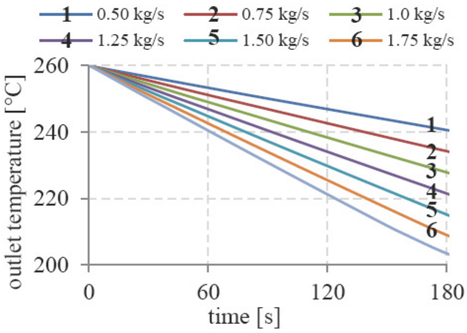


Fig. 6. SF outlet temperature depending on cloud duration

Fig.6 shows the SF outlet temperature depending on cloud duration for different values of mass flow. It is interesting that higher mass flow rates cause lower outlet temperatures. The temperature trend can be considered as linear and so it can be defined a rate of temperature decrease in function of cloud duration and mass flow. For a mass flow of 2.0 kg/s, the temperature decreases of about 60 °C during a three-minute-long cloud so the temperature decrease rate is about 20 °C/min. For a slower mass flow of 0.5 kg/s, even the temperature decrease rate is slower, and it is about 7 °C/min.

4. Conclusions

The results demonstrate that thermal energy losses during an average night are in the range of 140 kWh_t, 97.5% of which are due to receivers and 2.5% are due to insulated tubes. After 3 hours the maximum oil temperature is below 50 °C and after 4 hours its temperature is uniform and equal to the ambient temperature. The warm up phase of the solar field can last from 45 minutes up to 2 hours, depending on DNI. During full operation, the cloud influence on the oil temperature drop is strongly determined by its mass flow, and usual values range from 7 to 20°C/min. Experimental validation of the model is foreseen to be carried out once the construction will be completed and it will be reported in a future work.

5. Copyright

Acknowledgements

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Biography

Luca Migliari received the MA degree (summa cum laude – best graduate of the year) in mechanical engineering with an emphasis on solar systems, from University of Cagliari, Italy. He is a Ph.D. student in Industrial Engineering. His research interests include renewables and energy storage systems, simulation and management strategies.