Organic Rankine Cycle coupling with a Parabolic Trough Solar Power Plant for cogeneration and industrial processes

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ABSTRACT

Over the last 25 years solar power plants based on parabolic trough concentrators have been developed for the commercial power industry. On the other hand, in recent years, a way to harness the solar energy is to cogenerate through Concentrated Solar Power (CSP) technology coupled to an Organic Rankine Cycle (ORC) with potential applications to industrial processes. In this work we present a study of a small CSP plant coupled to an ORC with a novel configuration since useful energy is directly used to feed the power block and to charge the thermal storage. In order to analyze this novel configuration we consider a case study with cogeneration applied to textile industrial process at medium temperature. It turns out that this configuration reduces the size of the thermal storage disposal. The performance of the solar power plant was simulated with TRNSYS to emulate real operating conditions. We show the design, study and simulation results, including the production and efficiency curves for our load profile. Our results show that our system is a promising option for applications to medium temperature processes where electrical and heat generation is required.

1. Introduction

Currently the technological development of thermal generation plants and electric power recorded a preview and amazing flexibility. Multiple generation, i.e. the combined generation of mechanical and thermal power (low, medium and high temperature) using fossil fuels and/or renewable energy multiples rapidly, not only with high energy and exergy efficiencies but with very attractive economic returns and low environmental impacts.

In a few years, the large and medium users, public and private, have the possibility of having a diverse portfolio of energy with final prices depending on the season, from treasury weather and conditions of supply and demand exist, converting these energy systems in large energy and financial markets, as a necessary stage of the energy transition we experience. The incorporation of solar energy as a primary energy source in multiple generation will be key in these emerging markets, regardless of whether a country has abundant solar resources or not, because these markets will be carried over geographical boundaries, even for long distances between producers and consumers.

A parabolic trough concentrator (PTC) is a promising solar concentration technology to integrate solar energy as primary energy source. This technology converts the solar beam radiation into thermal energy in their linear focus receiver. PTC applications can be divided into two main groups. The first and most developed is Concentrated Solar Power (CSP) Plants. This technology is one of the main renewable energy alternatives for the production of electricity by solar power plants where the Rankine cycle is a common technology employed for commercial projects in the capacity range from 10 MWe to 90 MWe, and the operating temperature reached is in the range from 300 to 400 °C. CSP projects have recently become more economically appealing due to the improvements in concentrated solar power technology and cost. In particular, power plants with CSP and Organic Rankine Cycle, ORC, technologies have become a profitable choice due to the high performance, reliable and easy to use ORC power block units supplied by many worldwide manufacturers offering simple operation and low cost maintenance. Nonetheless, the question of the optimum Rankine cycle capacity remains an open issue. The second group is meant to provide thermal energy to applications that require temperatures between 85 and 250 °C. These applications use primarily industrial process heat, such as cleaning, drying,
evaporation, distillation, pasteurization, sterilization, cooking, among others, as well as applications with low-temperature heat demand and high consumption rates (domestic hot water, space heating and swimming pool heating), and heat-driven refrigeration and cooling. However, one of the aims of solar-thermal engineering is to develop collectors that are suitable for applications in the temperature range between 85 and 250 °C. Up to now only very limited experience exists for this temperature interval [1–3].

In the literature some researches have addressed the study of small-scale solar thermal power combining heat and power (CHP) systems by using solar thermal collectors coupled to an Organic Rankine Cycle (ORC) heat engine. In the following, we present a brief review of literature related to the generation of thermal energy with solar ORC.

In 2010, Delgado-Torres and García-Rodríguez [4] studied twelve substances as working fluids of the ORC and four different models of stationary solar collectors (flat plate collectors, compound parabolic collectors and evacuated tube collectors) were considered. They determined the operating conditions of the solar ORC that minimizes the aperture area needed per unit of mechanical power output of the solar cycle for each working fluid and each solar collector. Their results can be useful in techno-economic analysis, selection of working fluids of the Rankine cycle, sizing of systems and assessment of solar power cycle configuration. In the same year, Jing et al. [5] reported a low temperature solar thermal electric generation system that consists of a compound parabolic concentrators (CPC) and a ORC working with HCFC-123. They established the optimization of the system by considering the connection between the heat exchangers and CPC collectors, the tilt angle adjustment and the ORC evaporation temperature.

In 2011, Quoilin et al. [18] described the design of a solar ORC for rural electrification purposes. The system consisted of parabolic trough collectors, a storage tank, and a small-scale ORC engine using scroll expanders. They developed a model of each component based on experimental data for the main key components. They established a model for sizing the different components of the cycle and they evaluated the performance of the system. In their study, different working fluids were compared, and two different expansion machine configurations were simulated (single and double stage).

In 2012, Ya-Ling He et al. [6] developed a model for a typical parabolic trough solar thermal power generation system with ORC. They studied the system by using the transient energy simulation package TRNSYS [16]. They considered in their modeling the integration of several submodels for the trough collector system, a single-tank thermal storage system, an auxiliary power system and the heat-electricity conversion system. On basis of their modeling procedure they examined the influences of several designing and operating parameters on the performance of the collector field as well as the whole system. In the same year, Fahad A. Al-Sulaiman et al. reported the performance assessment of a novel system based on parabolic trough solar collectors and an Organic Rankine Cycle for combined cooling, heating and power (CCHP) [17]. They considered in their study that a portion of waste heat is used for heating through a heat exchanger and the other portion is used for cooling through a single-effect absorption chiller. They reported three modes of operation: a solar mode, which is characterized by a
low-solar radiation; a solar and storage mode, which is characterized by a high-solar radiation; and a storage mode, which is the operation of the system at night time through a thermal storage tank subsystem. Their modeling procedure is carried out by varying the ORC evaporator pinch point temperature, ORC pump inlet temperature, and turbine inlet pressure.

In 2014, Fahad A. Al-Sulaiman, described a detailed exergy analysis of selected thermal power systems driven by parabolic trough solar collectors (PTSCs). In Ref. [19] Al-Sulaiman reported some exergetic parameters: exergetic efficiency, exergy destruction rate, fuel depletion ratio, irreversibility ratio, and improvement potential. He concluded that parabolic trough solar collectors are the main source of the exergy destruction in which more than 50% of the solar inlet exergy is destructed. This value accounts for around 70% of the total exergy destructed. The evaporator is another source of exergy destruction in which around 13% of the solar inlet exergy is destructed. This value accounts for around 19% of the total exergy destructed in the system. The overall exergetic improvement potential of the systems was estimated to be around 75% In the same year, Ziviani et al. [20] reviewed the most recent advances and challenges for the exploitation of low grade thermal energy resources, with particular emphasis on ORC systems, based on information gathered from the technical literature. They presented an outline of the issues related to ORC system modeling and some guidelines drawn to develop an effective and powerful simulation tool. They reported a summary conclusion of the revised models, and a simulation tool of an ORC system suitable for the exploitation of low grade thermal energy.

One aspect that is very relevant in solar-powered Organic Rankine Cycle engines is the thermal storage. In 2011 Gang et al. [21] studied the electric generation by means of low-temperature based on the compound parabolic concentrator (CPC) of small concentration ratio and Organic Rankine Cycle (ORC). In their study they considered a two-stage system preheated by flat plate collectors (FPCs) prior to entering a higher temperature heat exchanger connected with the CPC. The two-stage heat storage units were composed of two types of phase change material (PCM) with diverse melting temperatures. They showed the benefits of the preheating concept and cascaded heat storages in comparison with the single-stage system. Their results indicate that the increase in collector efficiency of the two-stage system was appreciable. In 2013, Casati et al. [22] developed a study about extending direct flow-storage methods applicable to steam power plants to ORC power systems. They concluded that the so-called direct thermal storage systems are feasible, whereby the fluid circulated in the heat source, also serves as thermal storage medium, and is also the working fluid of the ORC turbogenerator.

We are far yet to see the end of developing solar-powered Organic Rankine Cycle engines, since the current and future challenges of these plants go through a flexibility in operating modes and business strategies and complex competitiveness, because they not only must be able to use in combination various primary energies (natural gas, biodiesel, solar energy, waste, geothermal, etc) according to the oscillating prices in the markets, but also they must be able to produce and deliver a combined electricity and thermal energy generation variable temperatures (80–400 °C) and soon fuels such as hydrogen, all with great flexibility and temporal powers, to maximize the profitability of these plants and adapt to the users needs.

The aim of this work is to conduct applied research to encourage the use of solar concentrated thermal technology at medium temperature for cogeneration and its application to industrial processes. Market studies show that in several industrial process the share of heat demand at temperature below 100 °C is about 30%, the share of heat demand at medium temperature [100 °C, 400 °C] is about 27% and the share of heat demand at high temperature >400 °C is about 47% [1].

This work presents a detailed study of a small Concentrated Solar Power (CSP) system composed by parabolic trough concentrators coupled to an ORC. By means of this system it is possible to generate electricity and useful heat to use them in a textile industrial process. It is important to point out that the configuration of the system is a novel design in order to increase the flexibility of the system and reduce the size of the storage system.

We consider, as a case study, a CSP plant with an ORC as shown in Fig. 2. The system is considered to be located in Almeria, Spain and the meteorological data file was obtained from the METEONORM software database [23] taking into account direct normal irradiance, ambient temperature, relative humidity and wind. The plant cogenerates thermal and electric power and the charge profile corresponds to the textile industry. The textile industry provides facilities for the treatment (washing, bleaching, mercerization) of fibers or textiles where the treatment capacity exceeds 10 tonnes per day. The temperature range is 90 °C for washing, discoloring and staining and between 140 °C to 200 °C for reheating with a charge profile of 5328 working hours per year, corresponding to 2 daily shifts of 8 h each, from 08:00 h till 24:00 h, throughout the year except one month, in this case December. The CSP plant is mainly composed of:

1. Solar field;
2. Thermal storage system;
3. Regulating reservoir;
4. Auxiliary heaters;
5. Combined heat and power system;
6. Balance of plant (B.O.P.) system, which takes care of the control system.

The paper is structured as follows: Section 2 describes the fundamentals of the CSP technology, the ORC technology and both technologies coupled to each other. Section 3 describes the case study and the methodology followed in the study of the solar power plant. Section 4 presents the simulation results including an analysis comparing the configuration used in the present paper with the typical configuration used in solar thermal systems. Conclusions are presented in Section 5.

2. Fundamentals

In this section we present a brief description about Organic Rankine Cycle and Parabolic Trough Solar Power Plant. We present the mathematical model formulation for the CSP plant composed of a parabolic trough collectors field and we describe an ideal ORC cycle that consists of four processes: isobaric evaporation, isentropic expansion, isobaric condensation and isentropic pump.

2.1. Parabolic Trough Solar Power Plant

In this subsection the mathematical model formulation for the CSP plant composed of a parabolic trough collectors field is given. A

![Fig. 1. Organic Rankine Cycle.](image-url)
2.2. Organic Rankine Cycle

The ORC is a thermodynamic cycle which converts heat into work. Its working principle is similar to the Rankine Cycle but the working fluid is a high molecular mass organic one and it is described as follows: the working fluid is pumped to a boiler where it is evaporated, passes through a turbine and is finally condensed as shown in Fig. 1. The fluid allows Rankine cycle heat recovery from lower temperature sources such as biomass combustion, industrial waste heat, geothermal heat, solar ponds, etc.

The ideal cycle consists of four processes: isobaric evaporation, isentropic expansion, isobaric condensation and isentropic pump. The cycle efficiency is defined by,

\[ \eta = \frac{\dot{W}_{\text{exp}} - \dot{W}_{\text{pump}}}{Q_{\text{boil}}} \]  

where \( \dot{W}_{\text{exp}} \), \( \dot{W}_{\text{pump}} \) and \( Q_{\text{boil}} \) are the power produced by turbine, the pump and the boiler respectively [25]. ORC applications include waste heat recovery, solar thermal power, biomass cogeneration, geothermal power, solar desalination and solar cogeneration [26] and [27].

2.3. CSP coupled to ORC

The concentrated solar technology coupled to an ORC is very useful due to its potential industrial applications to medium-temperature processes. In particular, the ORC can be used in the solar parabolic trough technology instead of the usual steam Rankine cycle. The ORC allows a lower collector temperature, with a better collecting efficiency (reduced ambient losses), and hence the possibility of reducing the size of the solar fields. In order to increase the amount of harnessed solar energy, thermal energy storage (TES) is used, such as water or oil tanks, hot rocks, concrete, pebbles, molten salts technology, phase change materials and cryogenic energy storage [28].

There are already studies and plants with the parabolic trough ORC technology in the world [29–31]. The aim of this study is to encourage the use of this technology in Mexico and the present study is our first step to contribute in the field.

In this work we study the following configuration: a solar field connected simultaneously to a thermal storage tank and a ORC power generator supported by auxiliary heaters. We call this novel configuration the direct-feed storage configuration. It operates such that at good radiation, the solar field simultaneously feeds the ORC and charge the storage, whereas at medium radiation, it only feeds the ORC engine, and in the absence of radiation the storage tank will feed the ORC power block. The solar field consists of a set of concentrated mirrors, parabolic trough collectors, which convert the solar energy into thermal energy through the concentration of direct normal irradiance (DNI) into the heat transfer fluid (HTF) to increase its temperature to a medium temperature. The hot HTF is used to mainly feed a combined heat and power (CHP) block which by means of an ORC will generate electric and thermal power, and at maximum radiation, it the hot HTF heats an oil tank used as thermal energy storage to augment the solar fraction. The power plant is shown in the black diagram shown in Fig. 2. In contrast, the typical configuration is such that the solar field is only connected to the storage tank, and the storage tank feeds the ORC engine as shown in the red diagram in Fig. 2.

The plant efficiency is the quotient of the energy delivered by the ORC cycle divided by the energy input, defined as,

\[ \eta_{\text{plant}} = \frac{\dot{W}_{\text{turbine}} - \dot{W}_{\text{pump}} + \dot{Q}_{\text{cool}}}{\dot{\text{IncRad}} + \dot{\text{AH1}} + \dot{\text{AH2}}} \]  

where \( \dot{W}_{\text{turbine}} \) and \( \dot{W}_{\text{pump}} \) are the works done by the turbine and the pump respectively, \( \dot{Q}_{\text{cool}} \) is the energy from the condenser, \( \dot{\text{IncRad}} \) is the incident radiation on the collectors and \( \dot{\text{AH1}} \) and \( \dot{\text{AH2}} \) are the energies from the auxiliary heaters 1 and 2. On the other hand, the solar field efficiency is defined

\[ \eta_{\text{col}} = \frac{\dot{Q}_{\text{col}}}{\dot{\text{IncRad}}} \]  

where \( \dot{Q}_{\text{col}} \) is the useful energy gain from the collector field. Whereas the thermal and electric efficiencies of the plant are defined as

\[ \eta_{\text{t}} = \frac{\dot{Q}_{\text{cool}}}{\dot{\text{IncRad}} + \dot{\text{AH1}} + \dot{\text{AH2}}} \]  

\[ \eta_{\text{e}} = \frac{\dot{W}_{\text{turbine}} - \dot{W}_{\text{pump}}}{\dot{\text{IncRad}} + \dot{\text{AH1}} + \dot{\text{AH2}}} \]  

On the other hand, the solar fraction is calculated as
In order to carry out the modeling of the system composed by the ORC and the CSP plant, the energy simulation package TRNSYS [16] is adopted. TRNSYS is a transient systems simulation program package with modular structures, in which the user can specify the components that constitute the unevaled system and the manner in which they are connected.

3. Exergetic analysis of the solar ORC system

In this section an exergetic analysis is carried out in order to establish a thermodynamic assessment of the solar ORC system. Since a conventional energy analysis based on the First Law does not give the available work of the system, an exergy analysis, based on the Second Law, is required to describe quantitatively and qualitatively the solar ORC system.

The exergy analysis of the solar ORC system is based on two subsystems (see 2), one of them is the solar field defined by a group of parabolic trough collectors arranged in modules coupled to the storage tank, and the heaters 1 and 2, AH1 and AH2; the other subsystem is the power block, that consists of a heat engine operating as an ORC, and it includes a heat exchanger (evaporator/boiler), a turbine, a condenser, and a pump as is depicted in Fig. 1.

In the following subsections, energy and exergy analysis are performed for both subsystems, the solar field and the power block. This analysis is based on the Refs. [7–9]. Nevertheless, it is important to point out that the analysis given in the subsections could not be further used for other typical systems since it is based on a novel direct feed-storage configuration used in this work.

3.1. Solar field subsystem

The available exergy of the solar field system consists of the exergies delivered from the energy sources, namely, the solar irradiation and the auxiliary heaters 1 and 2, AH1 and AH2, and it is described by,

\[ E_s = IncRad \left( 1 - \frac{T_a}{T_s} \right) + AH1 \left( 1 - \frac{T_a}{T_f} \right) + AH2 \left( 1 - \frac{T_a}{T_f} \right). \]  

(8)

where \( T_s \) is the apparent temperature of the Sun as an exergy source which is of the order of 4500 K [10]. It is important to point out that the TES is not a source of energy, it only provides an interface between the solar field and the power block subsystem. The exergy \( E_{s} \) delivered by the parabolic arrangement is given by,

\[ E_{s} = N \dot{m} \left( h_{\text{out}} - h_{\text{out}} - s_{\text{out}} \right) - \left( h_{\text{in}} - h_{\text{in}} \right)_{\text{col}} = \dot{Q}_{U} \left( 1 - \frac{T_a}{T_{\text{out}}} \right). \]  

(9)

where the heat gain of the collector is given by

\[ \dot{Q}_{U} = m (h_{\text{out}} - h_{\text{in}})_{\text{col}}. \]  

(10)

and \( N \) is the number of collectors in the arrangement, \( m \) is the mass flow rate in the solar field subsystem and \( h_{\text{in}}, h_{\text{out}}, s_{\text{in}} \), and \( s_{\text{out}} \) are the specific enthalpies and entropies at the inlet and, outlet of the arrangement respectively, and \( T_{\text{out}} \) is the outlet temperature of the fluid.

From Eqs. (8) and (9), the instantaneous exergy efficiency of the solar collector field can be defined as the ratio of the exergies of the increased thermal fluid to the one of the solar radiation and the auxiliary heaters [11].

\[ \eta_{\text{t,CS}} = \frac{\dot{Q}_{U} \left( 1 - \frac{T_a}{T_{\text{in}}} \right)}{IncRad \left( 1 - \frac{T_a}{T_s} \right) + AH1 \left( 1 - \frac{T_a}{T_f} \right) + AH2 \left( 1 - \frac{T_a}{T_f} \right)}, \]  

(11)

or as

\[ \eta_{\text{t,CS}} = \frac{\dot{Q}_{U} \left( 1 - \frac{T_a}{T_{\text{in}}} \right)}{IncRad \left( 1 - \frac{T_a}{T_s} \right)}, \]  

(12)

in the absence of auxiliary heaters.

On one hand, the irreversibility of the collector subsystem is calculated by considering the exergy destruction.

\[ IR_{\text{CS}} = E_s - \dot{E}_{U} = IncRad \left( 1 - \frac{T_a}{T_s} \right) - \dot{Q}_{U} \left( 1 - \frac{T_a}{T_{\text{out}}} \right), \]  

(13)

and the entropy rate \( S_{\text{gen,CS}} \) [W K\(^{-1}\)] in the collector system is defined by

\[ S_{\text{gen,CS}} = \frac{Q_{\text{lost}}}{T_a} + \frac{\dot{Q}_{U}}{T_s} - \frac{IncRad}{T_s}, \]  

where

\[ Q_{\text{lost}} = IncRad - \dot{Q}_{U}. \]  

(14)

is the collector field energy loss in the surroundings, and \( T_s \) is the outlet temperature of the fluid. Therefore, the entropy rate, \( S_{\text{gen,CS}} \), is

\[ S_{\text{gen,CS}} = \frac{1}{T_a} \left( IncRad \left( 1 - \frac{T_a}{T_s} \right) - \dot{Q}_{U} \left( 1 - \frac{T_a}{T_{\text{out}}} \right) \right). \]  

(15)

Note that the irreversibility of the solar field subsystem is established by the Gouy-Stodola theorem

\[ IR_{\text{CS}} = T_a S_{\text{gen,CS}}. \]

3.1.1. Exergy analysis for the storage tank

Unlike conventional heating systems, thermal energy generation of a solar energy system does not always match energy demands when considering time and rate of use. In this case, a Thermal Storage Subsystem (TES) is required to meet energy needs. As we mentioned in the previous subsection, the storage tank exergy does not contribute to the exergy of the solar field subsystem however, as a matter of completeness, we calculate it to describe its exergetic behavior. To perform the exergy analysis for the TES, we only need to consider the energy source delivered from the parabolic trough collectors arrangement.

In this subsection we present a simplified model of the TES during its three operating stages: charging, storing, and discharging, and we described the overall exergy efficiency of the TES during an overall cycle.
3.1.1.1. TES charging stage. In the charging stage, the extra energy collected by the solar field is transferred by the flow of the HTF to the TES. The HTF flow mass is then determined as follows by considering Eq. (10):

\[ m_{\text{col}} = \frac{Q_u}{(h_{\text{out}} - h_{\text{in}})_{\text{col}}}, \]  

(16)

where \( m_{\text{col}} \) is the mass of the HTF to the TES during charging. An exergy balance for the TES system during the charging stage can be written as,

\[ E_{UJ} - E_{\text{dest}} - E_{\text{loss}} = E_{\Delta \text{Ch}}, \]  

(17)

where \( E_{UJ} \) is the exergy delivered by the parabolic arrangement (Eq. (9)), \( E_{\text{dest}} \) is the exergy destruction, \( E_{\text{loss}} \) is the exergy loss, and \( E_{\Delta \text{Ch}} \) is the exergy accumulation during the charging stage period. The exergy loss \( E_{\text{loss}} \) is due to the energy flux to the surroundings \( Q_{\text{loss}} \) and it can be estimated considering,

\[ E_{\text{loss}} = Q \left( 1 - \frac{T_a}{T_e} \right), \]  

(18)

where \( T_e \) is the equivalent temperature of a mixed TES, and it can be expressed as

\[ T_e = \exp \left( \frac{T_a(\ln T_a - 1) - T_b(\ln T_b - 1)}{(T_a - T_b)} \right). \]  

(19)

where \( T_a \) is the temperature at the top of the TES, and \( T_b \) is the temperature at the bottom. The exergy accumulation \( E_{\Delta \text{Ch}} \) during the charging stage is described by,

\[ E_{\Delta \text{Ch}} = E_{\text{out}} - E_{\text{in}} = m[h_{\text{out}} - u_i - T_a(s_{\text{out}} - s_i)], \]  

(20)

where \( h_{\text{out}} \) and \( u_i \) are the specific internal energy at the outlet and inlet states of the TES respectively, and \( s_{\text{out}} \) and \( s_i \) are the outlet and inlet specific entropy of the TES. The exergy destruction \( E_{\text{dest}} \) is quantified by

\[ E_{\text{dest}} = N m_{\text{col}}((h_{\text{out}} - T_a s_{\text{out}}) - (h_{\text{in}} - T_a s_{\text{in}})) - Q_{\text{loss}} \left( 1 - \frac{T_a}{T_e} \right) - m_{\text{col}}[u_{\text{out}} - u_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}})]. \]  

(21)

The exergy efficiency \( \eta_{\text{ch}} \) of the TES during the charging stage is described by:

\[ \eta_{\text{ch}} = \frac{E_{\Delta \text{Ch}}}{E_{UJ}}. \]  

(22)

3.1.1.2. TES storing stage. The storing stage is the interim stage for a TES to store energy without charging or discharging. An exergy balance for the storing stage can be expressed as:

\[ -E_{\text{loss}} - E_{\text{dest}} = E_{\Delta \text{St}}, \]  

(23)

where \( E_{\Delta \text{St}} \) is the exergy of the storage and the exergy efficiency of storing period \( \eta_{\text{st}} \) is quantified by:

\[ \eta_{\text{st}} = \frac{E_{\Delta \text{St}}}{E_{\Delta \text{Ch}}}. \]  

(24)

3.1.1.3. TES discharging stage. During the discharge stage, the stored energy in the TES is recovered to use it in the power block. The outlet oil flow mass \( m_{\text{arc}} \) can be evaluated as:

\[ m_{\text{arc}} = \frac{Q_{\text{rec}}(h_{\text{out}} - h_{\text{in}})_{\text{rec}}}{(h_{\text{out}} - h_{\text{in}})_{\text{rec}}}, \]  

(25)

where \( Q_{\text{rec}} \) denotes the recovered energy from the TES and \( h_{\text{in}}, h_{\text{out}} \) are the specific enthalpies, at inlet and outlet, respectively.

An exergy balance for the TES can be expressed as follows:

\[ -(E_{\text{rec}} + E_{\text{loss}}) - E_{\text{dest}} = E_{\Delta \text{Dis}}. \]  

(26)

where \( E_{\text{dest}} \) is the exergy destruction, \( E_{\text{loss}} \) is the exergy loss, \( E_{\Delta \text{Dis}} \) is the exergy accumulation during discharging stage period, and \( E_{\text{rec}} \) is quantified by

\[ E_{\text{rec}} = m_{\text{arc}}(h_{\text{out}} - h_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}})). \]  

(27)

Here, \( E_{\text{rec}} \) denotes the recovered exergy, \( h_{\text{out}} \) and \( h_{\text{in}} \) are the specific enthalpy of the TES outlet and inlet oil respectively, and \( s_{\text{out}} \) and \( s_{\text{in}} \) represent the specific entropy of outlet and inlet oil from/to the TES respectively. Note that,

\[ E_{\Delta \text{Dis}} = E_{\text{out}} - E_{\text{in}} = m[h_{\text{out}} - u_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}})]. \]  

(28)

and

\[ E_{\text{dest}} = m_{\text{arc}}((h_{\text{out}} - T_a s_{\text{out}}) - (h_{\text{in}} - T_a s_{\text{in}})) - Q_{\text{loss}} \left( 1 - \frac{T_a}{T_e} \right) - m[u_{\text{out}} - u_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}})]. \]  

(29)

where \( u_{\text{out}} \) and \( u_{\text{in}} \) are the outlet and inlet states internal energy of the TES respectively.

The exergy efficiency of the TES during the discharging period, \( \eta_{\text{Dis}} \), can be expressed as:

\[ \eta_{\text{Dis}} = \frac{E_{\text{rec}}}{E_{\Delta \text{St}}}. \]  

(30)

3.1.1.4. TES overall cycle. An overall exergy balance for the TES during a period of time is defined by:

\[ E_{UJ} - (E_{\text{rec}} - E_{\text{loss}}) - E_{\text{dest}} = E_{\Delta}, \]  

(31)

where \( E_{\Delta} \) is change in the exergy accumulation. Then, the overall Second Law efficiency of the TES can written as

\[ \eta_{\text{TES}} = (\eta_{\text{ch}})(\eta_{\text{st}})(\eta_{\text{Dis}}) = \frac{E_{\text{rec}}}{E_{UJ}} \]  

(32)

or in explicit form as

\[ \eta_{\text{TES}} = \frac{m_{\text{arc}}(h_{\text{out}} - h_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}}))}{N m_{\text{col}}(h_{\text{out}} - h_{\text{in}} - T_a(s_{\text{out}} - s_{\text{in}}))_{\text{col}}} = \frac{Q_{\text{rec}} \left( 1 - \frac{T_a}{T_{\text{rec}}} \right)}{Q_{\text{UJ}} \left( 1 - \frac{T_a}{T_{\text{rec}}} \right)}. \]  

(33)

It is important to point out that the overall efficiency established by the First Law \( \eta_{\text{TES}} \) is:

\[ \eta_{\text{TES}} = \frac{Q_{\text{rec}}}{Q_{\text{UJ}}}. \]  

(34)
Considering the available exergy by the solar field system \( \eta_{II,CSO} \) established by Eq. (12), the instantaneous Second Law efficiency taking into account the TES, can be described by

\[
\eta_{II,TES,CSO} = \left( \frac{\eta_{II,CSO}}{\eta_{II,TES}} \right) = \frac{Q_{rec} \left( 1 - \frac{T_{rrec}}{T_{e}} \right)}{IncRad \left( 1 - \frac{T_{i}}{T_{e}} \right)},
\]

where \( IncRad \) is the incident radiation on the collectors calculated by,

\[ IncRad = \frac{A}{I_{beam}}. \]

where \( A \) is the total area and \( I_{beam} \) is the amount of beam solar radiation incident on the plane of the collector surface.

### 3.2. Power block subsystem

In this case, the subsystem is operated by the energy obtained from the parabolic trough collectors, and the heaters 1 and 2, \( AH1 \) and \( AH2 \). Therefore the exergy balance is described by considering the exergy delivered from those heat sources.

The useful exergy \( Ex_U \) delivered by the solar field system is transferred to the ORC organic fluid through the heat exchanger (evaporator/boiler). The Second Law efficiency of the heat exchanger \( \eta_{II,Ex} \) is then established by,

\[
\eta_{II,Ex} = \frac{Ex_{ex}}{Ex_U} = \frac{Q_R \left( 1 - \frac{T_{i}}{T_{e}} \right)}{Q_U \left( 1 - \frac{T_{i}}{T_{e}} \right)},
\]

where \( T_R \) is the inlet temperature of the ORC, and the exergy delivered by the heat exchanger, \( Ex_{ex} \), is calculated by,

\[
Ex_{ex} = \dot{m}_{orc} ((h_{out} - T_aS_{out}) - (h_{in} - T_aS_{in}))_R = Q_R \left( 1 - \frac{T_a}{T_R} \right).
\]

where \( \dot{m}_{orc} \) is the mass flow rate of the organic fluid in the power block subsystem, \( h_{in}, h_{out}, S_{in} \) and \( S_{out} \) are the specific enthalpies and entropies at the inlet and outlet of the arrangement respectively, and \( Q_R \) is estimated heat considering the enthalpy difference between the inlet and outlet of the heat exchanger,

\[
Q_R = \dot{m}_{orc} (h_{out} - h_{in})_R.
\]

The exergy destruction in the heat exchanger is described by

\[
\dot{R}_{Ex} = Ex_U - Ex_{ex}.
\]

In the heat engine cycle the available exergy of the working fluid is the exergy delivered by the heat exchanger \( Ex_{ex} \). On the other hand, the turbine exergy output is the net work done by the heat engine:

\[
\dot{W}_{net} = \dot{W}_{turbine} - \dot{W}_{pump}
\]

and the exergy destruction is defined by the irreversibility

\[
\dot{R}_{HE} = Ex_{ex} - \dot{W}_{net}.
\]

The transferred exergy to the cooling medium in the condenser is calculated by

\[
Ex_{cool} = \dot{m}_{orc} ((h_{out} - T_aS_{out}) - (h_{in} - T_aS_{in}))_R = \dot{Q}_{cool} \left( 1 - \frac{T_a}{T_{cool}} \right),
\]

where the condenser useful heat, \( \dot{Q}_{cool} \), can be established by considering the enthalpy balance,

\[
\dot{Q}_{cool} = \dot{m}_{orc} (h_{out} - h_{in})_{cool}.
\]

Thus, the overall Second Law efficiency of the solar ORC system is given by

\[
\eta_{II,plant} = \frac{\dot{W}_{net} - \dot{Ex}_{cool}}{Ex_{s}}.
\]

Therefore the Second Law efficiency is

\[
\eta_{II,plant} = \frac{\dot{W}_{turbine} - \dot{W}_{pump} + \dot{Q}_{cool} \left( 1 - \frac{T_a}{T_{cool}} \right)}{IncRad \left( 1 - \frac{T_a}{T_{e}} \right) + AH1 \left( 1 - \frac{T_a}{T_{e}} \right) + AH2 \left( 1 - \frac{T_a}{T_{e}} \right)}
\]

Whereas the thermal and electric Second Law efficiencies of the plant are described by

\[
\eta_{II,t} = \frac{\dot{Q}_{cool} \left( 1 - \frac{T_a}{T_{cool}} \right)}{IncRad \left( 1 - \frac{T_a}{T_{e}} \right) + AH1 \left( 1 - \frac{T_a}{T_{e}} \right) + AH2 \left( 1 - \frac{T_a}{T_{e}} \right)}
\]

and

\[
\eta_{II,e} = \frac{\dot{W}_{turbine} - \dot{W}_{pump}}{IncRad \left( 1 - \frac{T_a}{T_{e}} \right) + AH1 \left( 1 - \frac{T_a}{T_{e}} \right) + AH2 \left( 1 - \frac{T_a}{T_{e}} \right)}
\]

In the next section we present a case study in order to show the novel design of the solar ORC system.

### 4. Case study and methodology

The aim of this work is to show the operation of the novel configuration, the direct feed-storage configuration, we are introducing. In order to achieve this we illustrate the operation with a suitable operation of the solar field and the storage. Likewise the organic fluid and operating conditions taking into account the commercially available components in the market with a good quality/price performance available. As a result, we considered the commercial NEPSolar parabolic trough collectors and the ORC power block from ORMAT as described below. The dimensioning of the plant was calculated based on the technical specifications of these components, and the selection of the HTF was made according to a suitable operation of the solar field and the storage. We consider the operation of a concentrated thermal solar power plant with storage coupled to an Organic Rankine Power Block to produce thermal and electric energy for industrial applications. In particular we consider one case in the textile industry which
The parabolic trough collectors include a solar tracking system and the characterized efficiency curve is given by.

We consider the thermal oil Therminol 55 as the heat transfer fluid (HTF) with an optimal flow of [1 m³/h, 1.5 m³/h] through each collector, as specified by the manufacturer, and we will consider a tank for the thermal storage.

On the other hand, Therminol 55 is the thermal fluid that is used in the parabolic trough collectors. This a synthetic heat transfer fluid designed to provide reliable, consistent heat transfer performance over a long life. This heat transfer fluid delivers superior cost-performance compared to common mineral oil-based heat transfer fluids. Main features of Therminol 55 are reported in Table 2.

For moderate enthalpy heat sources, ORC cycles offer many advantages over the conventional steam cycle, primarily due to the simplicity of the turbine, the control system, and the balance of plant. The distinguishing features of an ORC cycle have been treated in numerous papers [12–15]. For an ORC plant the turbine and piping sizes are smaller and thus less costly due to the fluid density differences. The condensing pressure in an organic cycle is generally above atmospheric thus eliminating the need for complex vacuum and gas purging equipment that is utilized in a steam condensing cycle. The ORC is considered as a thermodynamic process where heat is transferred to a fluid at a constant pressure.

\[
\dot{Q}_{\text{to ORC}} = \dot{m}_{\text{to ORC}} \cdot c_p \Delta T_{\text{to ORC}}
\]

where \(\dot{Q}_{\text{to ORC}} = 11075 \text{ kW} \) at the required temperature and \(\Delta T_{\text{to ORC}}\) is the difference between the HTF temperature at the input, \(T_{\text{in ORC}}\), and output, \(T_{\text{out ORC}}\), of the ORC power block. As indicated in the ORC power block specifications, an inlet temperature of 170 °C is good for the cycle operation, therefore we set \(T_{\text{in ORC}} = 170\) °C as our desired working temperature. By considering \(\Delta T_{\text{to ORC}} \approx 90\) °C it turns out that \(\dot{m}_{\text{to ORC}} = 233172 \text{ kg/h}\).

The dimensioning of the thermal storage is analogous. The

### Table 1

The specifications of a PolyTrough 1200 Base Module.

<table>
<thead>
<tr>
<th>Feature</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aperture area, (A)</td>
<td>28.8 m²</td>
</tr>
<tr>
<td>Aperture width, (W_a)</td>
<td>1.2 m</td>
</tr>
<tr>
<td>Focal length, (f)</td>
<td>0.65 m</td>
</tr>
<tr>
<td>Receiver temperature, (T_{\text{R}})</td>
<td>215 °C</td>
</tr>
<tr>
<td>Glass tube temperature, (T_{\text{TR}})</td>
<td>40 °C</td>
</tr>
<tr>
<td>Ambient temperature, (T_{\text{a}})</td>
<td>20 °C</td>
</tr>
</tbody>
</table>

Nominal efficiency is close to 55% at operating range of 120–220 °C outlet temperature and 1 kW m⁻² direct normal radiation.

### Table 2

Typical properties of Therminol 55.

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity, (k)</td>
<td>0.1191 \text{ W/m K}</td>
</tr>
<tr>
<td>Kinematic viscosity, (\nu)</td>
<td>4.03 \text{ mPa s}</td>
</tr>
<tr>
<td>Expansion coefficient, (\beta)</td>
<td>9.6 × 10⁻⁴ K⁻¹</td>
</tr>
<tr>
<td>Density, (\rho)</td>
<td>822 \text{ kg/m³}</td>
</tr>
<tr>
<td>Heat capacity, (c_p)</td>
<td>2.17 \text{ kJ/kg K}</td>
</tr>
<tr>
<td>Thermal diffusivity, (\alpha)</td>
<td>6.68 × 10⁻⁴ \text{ m²/s}</td>
</tr>
</tbody>
</table>

Average properties at operating range of 120–220 °C outlet temperature.
volume of the tank, $V_{tank}$, is given by

$$V_{tank} = \frac{E_{stored}}{\rho C_p \Delta T_{tank}}$$

(50)

where

$$E_{stored} = 11075 \text{kW} \text{torage}$$

(51)

is the desired energy stored in some hours, $t_{storage}$, $\rho$ is the HTF density and we neglect the thermal losses. Therefore the estimated volume for 4 and 6 storage hours is approximately 1000 m$^3$ and 1500 m$^3$ respectively.

The solar plant works such that at maximum incident radiation it simultaneously feeds the ORC power block and charges the thermal storage (see Fig. 2).

The usual solar power plant configuration is such that the collected energy from the solar field is directly stored in the thermal storage and then the thermal storage feeds the process, as shown in the red diagram in Fig. 2. However we are interested in directly harnessing the solar energy to simultaneously feed the process and charge the storage field as we propose in the configuration shown in the black diagram in Fig. 2. The plant consists of a solar field directly connected to the storage tank and the ORC power block. There are 2 auxiliary heaters, AH1 and AH2, to heat the HTF in case there is no enough energy coming from the solar field or the storage tank. There is also a regulation tank in case the flow temperature coming from the solar field exceeds the required 170 °C. It counts with a sensor system located in ST1, ST2 and ST3 and a control system on the pumps, P1, P2 and P3, and the valves, V1, V3, V5, to regulate flows and temperatures.

Therefore the dimensioning of the solar field needs to take into account a) the required power delivered to the ORC power block and b) the required power to charge the storage tank. The number of collectors in series is given by the fact that the HTF should be as close as possible to the desired temperature ($T_{desired}$) at the output of the solar field. This is achieved by considering 3 or 4, depending on the incident radiation, NEPSolar parabolic trough collectors in series. The number of collectors in series is defined by the required flow

$$\dot{m} = \dot{m}_{ORC} + \dot{m}_{Tank}$$

(52)

where $\dot{m}_{ORC} = 242 \text{m}^3/\text{h}$ and $\dot{m}_{Tank}$ depends on the desired time to charge the storage tank. We set the charging time, $t_{charging}$, and estimated the required flow by

$$\dot{m}_{tank} = \frac{E_{stored}}{C_p \Delta t_{tank} t_{charging}}$$

(53)

In this case we consider an ideal charging time of 8 h for both tanks resulting on an approximate required flow of $\dot{m}_{tank} = 121 \text{m}^3/\text{h}$ for the 1000 m$^3$ tank and $\dot{m}_{tank} = 181 \text{m}^3/\text{h}$ for the 1500 m$^3$ tank. Therefore the possible solar collectors configurations to satisfy the previous requirements are reported in Table 4.

The operation of the solar power plant is shown in Fig. 4. The cycle starts when the load is required. If there is a minimum incident radiation $I_{min}$ the solar field is activated through the variable pump P1. If there is no incident radiation and there is still demand from the load profile the energy from the storage tank is used. In case the HTF temperature out of the tank is not 170 °C the auxiliary heater AH2 heats the fluid to reach the desired temperature and then to feed the ORC power block. When the solar field is active the flow through the solar field is regulated by the variable pump P1 such that for high incident radiation, $I > I_{crit}$, the flow is $\dot{m} = \dot{m}_{ORC} + \dot{m}_{Tank}$ and for low radiation, $I < I_{crit}$, the flow is $\dot{m} = \dot{m}_{ORC}$. In the case of low radiation, the flow, $\dot{m} = \dot{m}_{ORC}$, directly feeds the ORC power block using the auxiliary heater, AH1, in case its temperature is not 170 °C. On the other hand, in the presence of high radiation the flow, $\dot{m} = \dot{m}_{ORC} + \dot{m}_{Tank}$, is split by the valve, V1, such that $\dot{m}_{ORC}$ is directed to feed the ORC power block, and $\dot{m}_{Tank}$ is directed to charge the storage tank.

The plant counts with a control system shown in Fig. 5. Pumps, valves and auxiliary heaters are controlled with 5 controllers which are described as follows:

- The first controller, C1, has as set variables the incident radiation, the load profile and the desired temperature ($T = 170$ °C). It sends a signal to the variable pump P1 and the valve V1 to regulate and direct the flow depending on the measured temperature, ST1, at the end of the solar field. It regulates the flow through the solar field such that the temperature at the end of it is as closer as possible to the desired temperature ($T = 170$ °C). If the there is no load demand it stops P1. If the radiation is low it sends a signal, S1, such that the flow is the minimum one to feed the ORC power block. If the radiation is high S1 is such that the flow through the solar field is the maximum to feed the ORC and the storage and for intermediate radiation values it regulates P1 to the flow such that its temperature is close to 170 °C. The signal, S1, sent to the variable pump is summarized as follows:

$$S1 = \begin{cases} 0 & P1 \text{ OFF} \rightarrow \dot{m} = 0 \\ 2/3 & P1 \text{ ON} \rightarrow \dot{m}_{min} = \dot{m}_{ORC} \\ (2/3, 1) & P1 \text{ ON} \rightarrow \dot{m} \\ 1 & P1 \text{ ON} \rightarrow \dot{m}_{max} = \dot{m}_{ORC} + \dot{m}_{Tank} \end{cases}$$

C1 also sends a signal to the valve, Val 1, to divert the flow such that in the presence of flow there is always $\dot{m}_{ORC}$ directed to the power block and the remaining flow is directed to the storage tank.

- The second controller, C2, has as set variable the desired temperature and as input variable the temperature measured by ST1 at the end of the solar field. It sends a signal, S2, to the auxiliary heater, AH1, to heat the HTF to 170 °C in case it is necessary. S2 is either 0 or 1.

- The third controller, C3, has as set variable the desired temperature and as input variable the HTF temperature out of the valve Val 2. In case the HTF temperature exceeds 170 °C,
measured with the sensor ST2, it sends a signal, S3, to the pump P2 to activate to regulate the temperature by mixing the flow coming from the solar field with the one from the regulation tank. It also sends a signal to the valve V3 to divert back the exceed flow, such that at the output of Val 3 the flow will be \( m_{\text{ORC}} \), necessary flow for the ORC power block. S3 takes vales between [0,1].

The fourth controller, C4, has as set variables the radiation, the load profile and the desired temperature. In case there is a lack of radiation and load demand it activates the pump P3 in order that the storage tank feeds the ORC power block. It also sends a signal, S4, to the valve Val5 to divert back to the storage tank its flow. S4 is either 0 or 1.

The fifth controller, C5, has as set variable the desired temperature. Depending on the measured temperature in the storage tank, ST3, it sends a signal to the auxiliary heater, AH2, to heat the flow from the storage tank to reach 170 °C. S5 is either 0 or 1.

5. Simulation results

We simulate the operation of the plant with TRNSYS 17 and the “High Temperature” library provided by Thermal Energy Systems Specialists (TESS) together with the Engineering Equation Solver EES to simulate the ORC power block. We use meteorological data from Almeria for a typical year and study the configurations shown in Table 4 corresponding to 6 and 4 storage hours and 4 or 3 collectors in series. In Table 5 the plant configurations are reported.

Fig. 6 shows the plant operation during the first week of January. The cyan (in Web version) curve shows the load profile corresponding to a constant 11075 kW delivered to the ORC power block from 8:00 to 24:00 h. The red (in Web version) line shows the power from the Sun corresponding to approximately 8 h of radiation. The plant dimensioning is such that the power from the Sun exceeds the load profile in order to cover the power demand and the heat the storage in such a way that the remaining power (the area above the cyan line) corresponds to the power from the
storage delivered to the power block (blue (in Web version) line) when the radiation is over. Finally the power from the auxiliary heater AH1, shown in the green (in Web version) line, is used when the HTF temperature is not high enough during the radiation hours, and the power from the auxiliary heater AH2, purple line, adds to the power from the storage to provide the required power to the ORC power block.

The four types of plants present a similar behavior. To illustrate it we consider the particular case of type a). Fig. 7 shows the monthly load curves for plant a. The purple (in Web version) lines show the monthly energy delivered to the ORC power block. The blue (in Web version) line shows the power from the Sun which is bigger during summer and lower during winter. On the other hand, the delivered energy from the auxiliary heater AH1, red (in Web version) line, is lower during summer and bigger during winter. The energy from the storage, cyan (in Web version) line, is also higher during summer and lower during winter, however the energy needed from the auxiliary heater AH2 is more than the one from AH1 and it is higher during winter and lower during summer.

Likewise, Fig. 8 shows the monthly efficiencies for the configuration type a. The plant efficiency, red lines, is around 55%, the collector efficiency, blue lines, is around 50%, and the thermal, purple lines, is around 48% whereas the electric, green lines, is around 7%.

The thermal and electric efficiencies of the plant are defined by Eqs. (5) and (6), and the solar fraction is calculated by using Eq. (7).

Every plant produces 1.32 MW_e and 9.38 MW_t and the average annual efficiencies by First and Second Laws, and solar fraction are summarized in Table 6.

It is important to point out that the bigger the solar fraction the lower the efficiencies as showed in Table 6.

5.1. Comparing with a typical configuration

If, on the contrary, the plant is designed with the conventional configuration -storage configuration-, as shown in the red diagram in Fig. 2, its operation differs. We first estimate the storage tank volume by considering, for example, an average of 8 h radiation plus the 4 h or 6 h of storage as before, so that the stored energy required, \( E_{\text{stored}} \), in the tank is

\[
E_{\text{stored}} = 11075\text{ kW} \times (8h + t_{\text{stored}}) = 11075\text{ kW} \times 8h + E_{\text{stored}}. \tag{54}
\]

where \( E_{\text{stored}} \) is as defined in 51 and corresponds to the required energy for 4 or 6 storage hours.

### Table 5
Plant configurations.

<table>
<thead>
<tr>
<th>Type</th>
<th>( V_{\text{tank}} ) [m³]</th>
<th>Hours of storage</th>
<th>Collector in series</th>
<th>Collector in parallel</th>
<th>Total area (m²)</th>
<th>TES efficiency ( \eta_{\text{TES}} ),</th>
<th>TES efficiency ( \eta_{\text{TES}} ),</th>
</tr>
</thead>
<tbody>
<tr>
<td>a)</td>
<td>1500</td>
<td>6</td>
<td>4</td>
<td>282</td>
<td>32,486.4</td>
<td>52.25</td>
<td>19.28</td>
</tr>
<tr>
<td>b)</td>
<td>1500</td>
<td>6</td>
<td>3</td>
<td>282</td>
<td>24,364.8</td>
<td>55.39</td>
<td>20.05</td>
</tr>
<tr>
<td>c)</td>
<td>1000</td>
<td>4</td>
<td>4</td>
<td>242</td>
<td>27,878.8</td>
<td>28.35</td>
<td>10.26</td>
</tr>
<tr>
<td>d)</td>
<td>1000</td>
<td>4</td>
<td>3</td>
<td>242</td>
<td>20,908.8</td>
<td>29.00</td>
<td>10.50</td>
</tr>
</tbody>
</table>
The tank volume for this configuration, $V_{tank}$, is now estimated as

$$V'_{tank} = \frac{E_{stored}}{\rho C_p \Delta T_{tank}}$$  \hspace{1cm} (55)

$$= 2000 \text{m}^3 + V_{tank};$$  \hspace{1cm} (56)

where $V_{tank}$ is defined as in Eq. (50) and is either 1000 m$^3$ for 4 storage hours or 1500 m$^3$ for 6 storage hours. The first thing we see is that if all the solar energy is transferred to the thermal storage disposal we need 2000 m$^3$ of storage capacity and of Therminol 55. If we estimate the recharging time for this additional volume considering $m_{ORC}$ as the recharging flow it turns out that it needs approximate 7.3 h of good radiation to charge.

Moreover, we can dimensionate the solar field for this configuration demanding a 4 h charging time which implies.

**Fig. 7.** The monthly energy delivered for the particular case of type a).

**Fig. 8.** Monthly efficiencies where the red lines represent the plant efficiency, the blue ones the solar field efficiency and the purple and green ones the thermal and electric plant efficiency respectively. (For interpretation of the references to color in this figure legend, the reader is referred to the web version of this article.)

<table>
<thead>
<tr>
<th>Type</th>
<th>Solar fraction ($f$) [%]</th>
<th>Thermal efficiency</th>
<th>Electric efficiency</th>
<th>Plant efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>($\eta_t$) [%]</td>
<td>($\eta_{th}$) [%]</td>
<td>($\eta_{pl}$) [%], ($\eta_{pl\text{total}}$) [%]</td>
</tr>
<tr>
<td>a)</td>
<td>46.29</td>
<td>48.64, 17.61</td>
<td>6.79, 7.26</td>
<td>55.43, 24.87</td>
</tr>
<tr>
<td>b)</td>
<td>36.11</td>
<td>53.59, 19.40</td>
<td>7.53, 8.05</td>
<td>61.12, 27.45</td>
</tr>
<tr>
<td>c)</td>
<td>31.73</td>
<td>55.58, 20.13</td>
<td>7.76, 8.30</td>
<td>63.34, 28.43</td>
</tr>
<tr>
<td>d)</td>
<td>24.34</td>
<td>59.80, 21.65</td>
<td>8.35, 8.93</td>
<td>68.15, 30.58</td>
</tr>
</tbody>
</table>
For 4 h storage: a charging flow of $m = 637500\text{kg/h}$ and then a
minimum solar field configuration with 750 collectors in par-
allel with 3 or 4 collectors in series;
For 6 h storage: a charging flow of $m = 743750\text{kg/h}$ and then a
minimum solar field configuration with 875 collectors in par-
allel with 3 or 4 collectors in series

requiring a much bigger, and therefore more expensive, solar field
than the one for the direct feed configuration, defined in Table 4.

This implies that this configuration is not feasible for this
required charge profile since 1) the storage volume increases a lot
increasing at the same time the cost of the plant; b) the storage
charging time is not fast enough to provide the required instantan-
eous energy to the power block; c) much bigger solar field.
Therefore the direct feed-storage configuration, black diagram in
Fig. 4, is better for this application.

6. Conclusions

The analysis shows the detailed simulated production of the
main components of the CSP plant taking into account meteor-
ological data from the site, control operation constraints and a
realistic load profile which can easily be applied to more industrial
processes such as pasteurization. We conclude that the direct feed-
storage configuration presented in this work is more suitable
compared to the typical configuration since the required volume of
the storage tank is less compared to the one needed in the typical
configuration, for the same storage hours. This configuration is a
promising option for applications to medium temperature indus-
trial processes with electrical and heat generation. As it is shown in
Table 5, the energy and exergy efficiencies of the thermal storage
systems decrease as the solar fraction increases. Likewise, as it is
shown in Table 6, the energy and exergy efficiencies of the solar-
ORC power plant decreases as the incident solar radiation in-
creases. On the other hand, as shown in Table 6, the overall system
efficiency is increased by using waste heat as a heat source. The
energy recovery is close to 55% and the global efficiency is
enhanced.

In order to increase the solar fraction and to improve the system it
is worth to extend the study to more storage technologies. Molten
salts could be a very good possibility to highly increase the solar
fraction but it is much more expensive. On the other hand concrete
storage is much cheaper but with less heat capacity. However phase
change materials may be a good and interesting candidate with a
balanced compromise between heat capacity and price.

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