Novel analysis of second law and irreversibility for a solar power plant using heliostat field and molten salt

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Abstract
Nowadays, the low efficiency of solar energy power plant systems brings about uneconomical performance and high-cost uncompetitive industries compared with the traditional fossil fuel ones. In order to overcome these kinds of issues, in this study, the performance of a solar tower coupled with a Rankine cycle is scrutinized using the first and second laws of thermodynamic. Moreover, a numerical code has been developed in the GNU Octave software environment to calculate the precise value for both energy and exergy losses of each component. Furthermore, the sensitivity analysis approach corresponds to the variation of several inner parameters such as direct normal irradiation (DNI), molten salt outlet temperature (MSOT), and the molten salt velocity (MSV), environmental parameters such as wind speed and ambient temperature have been performed. In addition, the overall losses are calculated by considering all possible forms of losses such as convection, conduction, reflection, and emission, and also, the portion of each source of these losses are determined. The obtained results indicate that the maximum exergy loss occurs in the central receiver system (CRS), while the major energy loss occurs in the turbine located in the power block. The sensitivity analysis shows that the rise of DNI significantly increases the exergy%energy efficiency of the cycle and also, the portion of loss related to emission heat transfer would be enhanced. Moreover, the variation of the MSV and MSOT illustrate influencing the performance of the cycle; consequently, the MSOT changes demonstrate a reverse relation with energy and direct relation with exergy efficiency. Accordingly, an optimum value of 650K is calculated for the MSOT as well as an optimum value of 2 m%s for the MSV. The environmental sensitivity analysis indicates that the enhancement of wind speed and ambient temperature has a negative impact on the net output of the cycle, which is not desirable.

KEYWORDS
energy loss, exergy loss, heliostat field, molten salt, solar power tower plant
All over the world, electrical energy is still being generated via environmental destructive techniques, consisting mainly from direct burning of conventional fossil fuel resources that are nonrenewable carbon-based materials resulted from anaerobic decomposition of buried ancient organisms.\(^1\) While burning them provides a significant amount of energy, it is known to be the main source of releasing carbon dioxide into the atmosphere, a greenhouse gas blamed to trigger global warming.\(^2,3\)

Despite burning fossil fuels, there are other approaches to generate electricity in an environmentally benign way: hydro energy, wind turbines, geothermal, and solar energy.\(^4,6\)

Among the noted methods, solar energy has been highlighted by providing an inexhaustible, nonpolluting, and safe source of energy.\(^7,8\) Repetitively recommended by environmental scientists, since the irradiation of the sun is efficient enough to establish solar-based power plants in most parts of the world (such as Asia, Africa, and both northern and southern parts of America and even in some regions of Europe), it can be a promising alternative energy source.\(^9,10\)

In the power plant industry, concentrated solar power plants (CSPP) appeared to be a suitable solar-based technology capable to produce electricity on a large scale.\(^11\) Unfortunately, CSPP's low efficiency results in high downstream costs, making them uncompetitive to fossil fuel power plants, yet.\(^12\)

Among all studies that have been carried out to overcome this deficiency, energy, and (especially) exergy analyses proved to be powerful tools to optimize a cyclic-based system.\(^13\) The exergy analysis is the result of the thermodynamics second law, which reflects the system loss, including causes, locations, and finally shows the imperfection points of the design while the first law only gives limited information for designing (and calculating) components of the cycle.\(^14\) In combination, they were utilized as a practical tool to optimize many thermodynamic-based systems throughout solar researches.\(^15\) Repeatedly stated, it was proven that the main source of energy/exergy loss occurs inside the receiver components.\(^16\)

This subject encouraged researchers to do more investigations on the receiver's performance.\(^17\) As an instance, Yao et al\(^18\) simulated a 1 MW solar thermal central receiver system in China. Even though they were pioneers in solar energy, unfortunately, it only comprised the energy (first law) analysis; therefore, only the overall performance of the receiver was investigated. Later, Parkash et al\(^19\) carried out a more detail-oriented investigation on the calculation of heat loss in a cylindrical cavity receiver. Since the loss of receiver had a crucial role in exergy analysis, the output data as well as conclusions were utilized in further studies in this field. Their survey led to several conclusions on the relationship between convective heat loss and fluid inlet temperature. For instance, they found that increasing the mean fluid temperature results in a higher convective loss.

Thereafter, Li et al\(^20\) presented an accurate thermal model of a molten salt cavity receiver. Several effective parameters such as receiver area, number of tubes in the receiver panel, tube diameter, and receiver surface temperature were tested, and mathematical formulas were generated based on the thermal performance of the receiver. This mathematical modeling was appreciated widely in various investigations.\(^21-23\) They were either directly utilized, or adapt to an improved version of it for their applications. For instance, using Li et al model, Tiryaki et al\(^23\) investigated to optimize cavity receiver performance in a solar power tower (SPT) system. They tested the effect of various factors such as heat loss and receiver surface temperature on view factor and concentration ratio and concluded that the design of the system with the smallest possible view factor and concentration ratio of 1000 resulted in the highest efficiency and the lowest heat loss for the system.

Coinciding with the articles that evaluated/optimized the performance of receivers, others focused more on the cyclic effect of the receivers on the energy/exergy loss of the solar subsystems. In this regard, the study carried out by Xu et al\(^21\) was above all, including both energy and exergy analysis over the SPT plants and closely correlated with the data obtained from Sandia national laboratory. They examined effective parameters such as the effect of DNI, molten salt temperature, and concentration ratio on energy and exergy efficiency. Finally, they enhanced the overall energy and exergy efficiencies of the solar tower system to some extent by integrating advanced power cycles such as reheating Rankine.

Although the scientific procedure was valuable, yet several deficiencies could be mentioned. First, the article solely focused on the analysis of the solar subsystem rather than the coupled cycle (whole power plant). Second, like previous articles, the survey was deeply concerned with the influence of receiver parameters; therefore, the ill effects of other parameters, especially environmental, have not met properly. Finally, the cycle was evaluated using overall energy/exergy integration; however, not much attention was given to the loss and how they were generated.

Despite the noted incompetence, their study was one of the leading ones at the time and inspired subsequent investigations. Afterward, various combinations of conventional-solar cycles were the subject of studies aiming to minimize the energy/exergy loss. For instance, Mehrpooya et al\(^24\) investigated a power plant integrated with a parabolic high-temperature energy solar trough system from thermodynamic and exergy aspects. They investigated the effect of different parameters such as inlet temperature and outlet pressure of gas turbines on the exergy and electrical efficiencies of the entire cycle. The most critical result was the total electrical efficiency and exergy efficiency of the power system were 47% and 38%, respectively.
Bonyadi et al.\textsuperscript{25} evaluated the exergy and the thermodynamic features of a combined solar and geothermal electric steam-Rankine power plant. Their findings revealed that the proposed model had a higher temperature at the inlet of the turbine, which led to higher efficiencies. Also, they showed that proper design could increase solar efficiency by almost 12%.

In another research, Mata-Torres\textsuperscript{26} investigated the thermodynamics and exergy performance of a Rankine cycle integrated into a multieffect distillation plant with molten salt fluid. Also, in their study, the effect of parameters such as part-load operation and ambient temperature has been analyzed. Their results showed that while the effect of ambient temperature on the water costs is negligible, the part-load operation has a significant impact on the thermo-economic and exergy costs.

Novales et al.\textsuperscript{27} proposed a model for the supercritical CO\textsubscript{2} cycle integrating with concentrating solar power plants using molten salt. The most objective part of their work focused on the sensitivity, optimization, energy, and exergy analysis of each component of the system and the entire cycle. They reported that the highest possible percentage of exergy efficiency is 79%, and the most exergy destruction appears in the heat recovery unit. Yang et al.\textsuperscript{28} developed a new model of solar power combined with direct steam generation system and molten salt fluid. In their study, the thermodynamic and exergy were compared with without the reheat section of the cycle, and the influence of system arrangement and the steam turbine was evaluated. Additionally, they showed that the thermal and exergy efficiencies of the direct steam generation molten salt system with a reheat section is lower than the system without the reheat section.

As the literature review indicated, even though various investigations have been conducted on the solar-based power cycles over the past decades, several important features are assumed to be discarded. First, almost all the noted investigations were carried out based on commercial simulators such as HYSIS, Thermoflow, and SeisEarth. Due to the complexity of the cycles, the users were unable to develop self-designed numerical codes, which would be able to calculate the target parameters. Although using commercial software is acceptable for industrial applications (time costs money), it would generate various types of limitations in academic usage. The least damage of this approach would either lead to miss-evaluation of some parameters, or unintentional discard of others if the user would not be extremely careful through the analysis. The scenario is deteriorated as the cycle expands regarding the various untraceable default parameters. Furthermore, some restrictions (lack of user control) in commercial simulators prevented the implementation of adding novel solution techniques for curious researchers. Second, in the aforementioned studied reviews, as the cycle was expanded, less analysis was conducted on the loss and the arisen causes. The optimizations were either by adding some components or utilizing different component arrangements rather than identification of loss causes, their impact on the overall loss, and control its effects whether it comes from several components or environmental factors. Third, due to the complexity of the cycles used, the exact (raw) influence of effective parameters on components and the influence of components on each other was not investigated properly. Fourth, even though the influence of cycle-based parameters (such as solar irradiation, concentration ratio, and molten salt temperature) was evaluated repeatedly, the impact of environmental parameters such as wind speed and ambient temperature was not properly investigated. Finally, in many of the previous literature, the power cycle was considered as a black box and only the solar subsystem calculations were conducted. Even though utilizing this approach simplified the inner calculations of the power cycle (to remove a thermodynamic calculation of working flow condition), it would add up severe restriction for an overall consideration of both solar and power subsystems.

Rectifying mentioned limitations is the motivation of the current study. For this, a numerical code was developed in GNU Octave to calculate the cycle parameters. GNU Octave is an open-source software featuring a high-level programming language, primarily intended for numerical computations. The Octave helps in solving linear and nonlinear problems numerically, and for performing other numerical experiments using a syntax that is almost compatible with MathWorks MATLAB.

The developed code (for the current investigation) consists of several sub-components. The first part calculates almost every detail of the solar subsystem components. For this reason, each component keeps its influence and the numerical techniques could be easily expanded for further usage. The other component of the code calculates thermodynamic properties (temperature, pressure, density, enthalpy, and entropy) of water or steam in the power subsystem (Rankine cycle).

In the current study, the behavior of the solar power tower system, consisted of the heliostat field, solar receiver with cavity-type and power block, and molten salt as the primary heat transfer fluid was evaluated using the exergy energy methodology. Due to its simplicity as well as industrial applicability, the previous work of Xu et al.\textsuperscript{21} was considered as the main reference throughout the investigations; however, the current study represents novelty at least from four perspectives. First, instead of an overall exergy energy calculation of components, the impact of each parameter on exergy and energy loss was investigated. This was achieved by applying a wide range of variations to the effective parameters (such as DNI, outlet temperature, the velocity of molten salt, and ambient temperature, solar irradiation, concentration ratio, and molten salt temperature) and evaluate their local and overall loss contributing to the system. Although energy exergy
calculations were repeatedly investigated before, their impact on the energy/exergy loss has never been investigated properly up to our knowledge. Moreover, the impact of several novel parameters (either environmental or cyclic) on the energy/exergy loss was investigated. Second, different types of energy loss (convection, conduction, and radiation), their corresponding sources, and their impact on overall energy/exergy efficiency were investigated widely throughout the current study. Third, the power cycle components were fully simulated, and the Rankine cycle was not considered as a black box. Therefore, an equal degree of importance was considered for energy/exergy calculations during the investigations. Finally, despite the results presented in ref.21 Here, the investigation is focused on the raw solar-Rankine cycle rather than other variations of the Rankine cycle.

Before going any further, an important discussion must be made regarding the different concepts between exergy loss and exergy destruction. Exergy loss is more related to the losses due to the environmental influence while the exergy destruction is the loss that is occurred in the process of energy conversion or transmission. However, from the industrial perspective, the exergy loss could be considered equivalent to the exergy destruction since there would be a little chance that the process could be considered reversible; therefore, the reversibility of all the energy/exergy losses would be impossible and the loss and destruction concept could be considered in the same manner. In this regard the same as,29,30 here, the loss was considered irreversible, and the loss statement was used instead of destruction to avoid repetitive explanation throughout the simulation. For a more enthusiastic reader, further important notes between these two concepts could be found in ref31 for further advanced details.

The remainder of this research will be presented in three sections. In Section 2, it is aimed to briefly discuss the modeling techniques of corresponding components. Since the utilized formulations were provided from various articles, they are sufficient for further modeling simulations of an enthusiastic reader. Section 3 contains validation of the current simulation, as well as various numerical investigations on the cycle. Moreover, detailed discussions (on the topic of particle energy and exergy loss of each component and so on), will be carried out in this part. Finally, in Section 4, the discussion is summarized as a conclusion.

2 | CYCLE ENERGY AND EXERGY ANALYSIS

2.1 | Problem definition

A schematic diagram of a molten salt SPT and the corresponding thermodynamic cycle are shown in Figures 1 and 2, respectively. The depicted SPT consists of the heliostat field, solar receiver, and power block. According to Figure 1, as solar-based steam, the generator was used instead of a conventional boiler. Moreover, since the cycle operates in the steady-state condition, thermal storage was not considered in the cycle.

The cycle starts with the irradiation of sun on rotating mirrors called heliostats. The purpose of heliostat is to reflect the rays in the central receiver with the lowest possible loss, which was located at the top of a high tower. The increased concentration of rays resulted in the higher temperature of receiver heat transfer fluid (HTF), which is usually a family of molten salts with different combinations of KNO3 and NaNO3 and other compounds that are analyzed in the engineering database of liquid salt.32 The hot molten salt passes through the steam generator and subsequently transferred its heat with subcooled water entered from the other side of the steam generator. After the transfer of heat from HTF, the salt is directed to the receiver to be heated for the next process.

FIGURE 1 Schematic diagram of a power plant.
The T-S diagram of the corresponding.

FIGURE 2 The T-S diagram of the corresponding.

On the Rankine side of the power plant cycle, the subcooled water turns into superheated steam and passes through the steam turbine. Almost all of this steam is used for power generation, but after the first stage of the turbine, a part of it enters into the deaerator for warming the subcooled water that is pumped from the condenser. Subsequently, the subcooled water is pumped into the steam generator, and the cycling procedure repeats.

The energy and exergy can be analyzed based on the first and second laws of thermodynamics for both solar and Rankine subsystem. Solar irradiation is assumed constant while heat loss and pressure drop are discarded in all pipes. Besides, no chemical energy and exergy have been considered in both cycles because of the high computational complexity for this order of analysis. Considering the discussed assumptions, the first and second laws of thermodynamics for a control volume can be expressed as below.\(^{(33)}\)

\[
\sum_i \dot{H}_i + \sum_{\text{inlet}} \dot{m}_{\text{inlet}} h_{\text{inlet}} = \dot{W}_{\text{net}} + \sum_{\text{outlet}} \dot{m}_{\text{outlet}} h_{\text{outlet}} \quad (1a)
\]

\[
\dot{E}_{x,\text{loss}} = T_{\text{amb}} \dot{S}_{\text{gen}} = \sum_{\text{inlet}} \dot{m}_{\text{inlet}} \dot{E}_{x,\text{inlet}} - \sum_{\text{outlet}} \dot{m}_{\text{outlet}} \dot{E}_{x,\text{outlet}} + \sum_i \left( 1 - \frac{T_{\text{amb}}}{T_{\text{sun}}} \right) \dot{h}_{\text{inlet}} - \dot{W}_{\text{net}} \quad (1b)
\]

2.2 Helioestat field analysis

The helioestat field includes arrays of rotating mirrors reflecting sunrays toward a receiver. Various parameters such as daytime, cleanliness, reflect angle, the helioestat surface temperature, weather condition, and the moisture percentage of the air affect the energy/exergy loss. These parameters are assumed constant during the current investigations for three reasons. First, the aforementioned parameters are mostly associated with the geographical conditions, varying based on the location. Second, the simulation was considered to be in a steady-state (not transient), so the movements of the sun, as well as helioestats, were not considered in the formulations. Third, the purpose of study was to evaluate the energy loss after the absorption instead of the way it was captured. Therefore, the same as,\(^{(21)}\) they were all set constant during the investigations. Moreover, for an enthusiastic reader, it is recommended to go through.\(^{(34)}\) Besides, other factors such as cleanliness, shading, blocking, or material of helioestat and roughness of the surface are also effective on the helioestat performance. Helioestat efficiency \(\eta_{\text{hel}}\) is an indication of this performance. Based on,\(^{(21)}\) the energy and exergy loss quantities are calculated as follows:

\[
\dot{H}_{\text{hel.loss}} = \dot{H}_{\text{sun}} - \dot{H}_{\text{hel.use}} = (1 - \eta_{\text{hel}}) SIR_{\text{sun}} N_{\text{helioestat}} \quad (2a)
\]

\[
\dot{E}_{x,\text{hel.loss}} = \dot{E}_{x,\text{sun}} - \dot{E}_{x,\text{hel.use}} = \left( 1 - \frac{T_{\text{amb}}}{T_{\text{sun}}} \right) \dot{H}_{\text{hel.loss}} \quad (2b)
\]

where \(\dot{H}_{\text{loss}}\) and \(\dot{E}_{x,\text{loss}}\) are the heat and exergy loss, respectively. \(H_{\text{sun}}\) is the incident solar radiation. \(E_{x,\text{rec}}\) stands for the quantity of exergy delivered to the receiver. \(H_{\text{sun}}\) is calculated based on the amount of solar radiation received per unit surface of helioestat \(SIR_{\text{sun}}\) multiplied by the overall surface of helioestats. Energy and exergy efficiencies are given by equations (3a, 3b).

\[
\eta_{\text{en,hel}} = \eta_{\text{hel}} = \frac{\dot{H}_{\text{hel.use}}}{\dot{H}_{\text{sun}}} \quad (3a)
\]

\[
\eta_{\text{ex,hel}} = \frac{\dot{E}_{x,\text{hel.use}}}{\dot{E}_{x,\text{sun}}} \quad (3b)
\]

2.3 Central cavity receiver analysis

The central receiver is responsible for transfers helioestats absorbed energy to the molten salt flow. The analysis comprises the quantity of heat rate delivered to the receiver divided into two fractions. The useful fraction was the quantity that the receiver absorbed, whereas the second fraction was the total loss due to the heat transfer with the environment. The total loss occurred due to convection, emission, reflection, and conduction of heat transfer.\(^{(20)}\) The energy and exergy analysis can be expressed as follows:

\[
\dot{H}_{\text{rec.loss}} = \dot{H}_{\text{hel.use}} - \dot{H}_{\text{rec.use}} = \dot{H}_{\text{hel.use}} - \dot{m}_m (h_b - h_a) \quad (4a)
\]

\[
\dot{E}_{x,\text{rec.loss,tot}} = \dot{E}_{x,\text{hel.use}} - \dot{E}_{x,\text{rec.use}}
\]

\[
\dot{E}_{x,\text{rec.loss,1}} + \dot{E}_{x,\text{rec.loss,2}}
\]
\( \dot{E}_{\text{rec,loss1}} \) represents the exergy loss associated with the total heat loss of the receiver, whereas \( \dot{E}_{\text{rec,loss2}} \) associated with the loss imposed by the cycle and irreversibility of the receiver. The first part of exergy loss could be calculated using Equation (5). Contrarily, the second term of exergy loss could not be evaluated until all other terms were calculated.

\[
\dot{E}_{\text{rec,loss1}} = \dot{H}_{\text{rec,loss}} \left( 1 - \frac{T_{\text{amb}}}{T_{\text{rec,surf}}} \right) \quad (5)
\]

Finally, the energy and exergy efficiencies of the receiver are given as follows:

\[
\eta_{\text{en,rec}} = \frac{\dot{H}_{\text{rec}}}{\dot{H}_{\text{sun}}} \quad (6a)
\]

\[
\eta_{\text{ex,rec}} = \frac{\dot{E}_{\text{rec,use}}}{\dot{E}_{\text{rec}}} \quad (6b)
\]

Seven unknown terms existed in equations (6). Four of them, including \( \dot{H}_{\text{rec}} \), \( \dot{H}_{\text{rec,use}} \), \( \dot{E}_{\text{rec}} \), and \( \dot{E}_{\text{rec,use}} \) can be calculated straightforwardly; however, two problems existed within the calculation of the other three terms. First, the term \( \dot{H}_{\text{rec,loss}} \) is unknown, and second, the term \( \dot{E}_{\text{rec,loss1}} \) is strictly related to \( \dot{H}_{\text{rec,loss}} \). There exists no equation to calculate \( \dot{E}_{\text{rec,loss1}} \) directly. For this purpose, the mathematical modeling and formulation introduced by Li et al.\(^{20}\) were employed to balance the equations and variables. For calculating the \( \dot{H}_{\text{rec,loss}} \), all four sources of heat loss for a molten salt cavity receiver should be defined. Based on,\(^{20}\) the mathematical formulations of parameters associated with \( \dot{H}_{\text{rec,loss}} \) is as follows:

\[
\dot{H}_{\text{rec,loss}} = \dot{H}_{\text{rec,conv}} + \dot{H}_{\text{rec,em}} + \dot{H}_{\text{rec,ref}} + \dot{H}_{\text{rec,cond}} \quad (7)
\]

Afterward, based on,\(^{35}\) the Equations (8-11) were employed to calculate all four sources of heat loss as follows:

\[
\dot{H}_{\text{rec,em}} = \frac{e_w}{e_w + (1 - e_w)F_r} \sigma (T_{\text{rec,surf}}^4 - T_{\text{amb}}^4) \quad (8)
\]

\[
\dot{H}_{\text{rec,ref}} = \dot{H}_{\text{rec}} F_r \rho \quad (9)
\]

\[
\dot{H}_{\text{rec,conv}} = h_{\text{tot,conv}} (T_{\text{rec,surf}} - T_{\text{amb}}) \frac{A_{\text{hel,tot}}}{C} \quad (10)
\]

\[
\dot{H}_{\text{rec,cond}} = \frac{(T_{\text{rec,surf}} - T_{\text{amb}})}{(\delta_{\text{insu}} \% \lambda_{\text{insu}} + 1 \% h_{\text{air,do}}) F_r C} \quad (11)
\]

Whereas, the surface temperature of the receiver \( T_{\text{rec,surf}} \) is obtained based on\(^{20}\) as follows:

\[
T_{\text{rec,surf}} = \frac{T_a + T_b}{2} + \dot{H}_{\text{rec}} \left( \frac{d_d}{d_h} \ln \frac{d_b}{d_i} \right) \frac{F_r C}{A_{\text{hel,lot}}} \quad (12)
\]

Furthermore, the convective coefficient of the air is calculated based on\(^{36}\) as below:

\[
C_{\text{f,air}} = 0.334 \left( \frac{\mu_{\text{air}} C_{\text{p,air}}}{k_{\text{air}}} \right)^{0.8} + 0.81 (T_{\text{rec,surf}} - T_{\text{amb}})^{1.426} \quad (13)
\]

### 2.4 Steam generator analysis

The steam generator is a heat exchanger with the primary purpose of performing the boilers’ responsibility (generate steam), but instead of the conventional fossil fuels, a clean energy source (solar) is utilized. In this component, the heated molten salt transfers its energy to the subcooled water to generate superheated steam for the steam turbine. Therefore, the only assumption regarding this component is the degree of energy loss among the molten salt and the subcooled water. By the use of simple energy%exergy conservation law for this component, the energy and exergy loss were calculated as follows:

\[
\dot{H}_{\text{sg,loss}} = \dot{H}_{\text{rec,use}} - \dot{H}_{\text{sg,ase}} = m_{\text{m}} (h_b - h_a) - \eta_{\text{sg,steam}} (h_5 - h_4) \quad (14a)
\]

\[
\dot{E}_{\text{sg,loss}} = \dot{E}_{\text{rec,use}} - \dot{E}_{\text{sg,ase}} = \dot{E}_{\text{rec,use}} - \dot{m}_{\text{m}} (h_5 - h_4 - T_{\text{amb}} (s_5 - s_4)) \quad (14b)
\]

Moreover, the exergy efficiency is calculated as below:

\[
\eta_{\text{en,sg}} = \frac{\dot{H}_{\text{sg,use}}}{\dot{H}_{\text{rec,use}}} \quad (15a)
\]

\[
\eta_{\text{ex,sg}} = \frac{\dot{E}_{\text{sg,use}}}{\dot{E}_{\text{rec,use}}} \quad (15b)
\]

### 2.5 Steam Turbine analysis

For the Rankine subsystem, the analysis was started by applying the first and second law of thermodynamic for the steam turbine to calculate energy%exergy losses as follows\(^{37}\):

\[
\dot{H}_{\text{turb,loss}} = \dot{H}_{\text{sg,ase}} - \dot{H}_{\text{turb}} = \dot{H}_{\text{sg,ase}} - \dot{m}_{\text{steam}} (h_b - h_a) + (1 - y) \left( h_7 - h_6 \right) \quad (16a)
\]

\[
\dot{E}_{\text{turb,loss}} = \dot{E}_{\text{turb}} = \dot{E}_{\text{gen,cond}} = (1 - y) \dot{m}_{\text{steam}} T_{\text{amb}} (s_7 - s_1) \quad (16b)
\]
Finally, the energy and exergy efficiencies of the steam turbine are given by equations (17-a, 17-b) as follows.

\[ \eta_{en,turb} = \frac{\dot{W}_{turb}}{\dot{H}_{g,use}} \]  

(17a)

\[ \eta_{ex,turb} = \frac{\dot{E}_{x,turb,use}}{\dot{E}_{x,g,use}} \]  

(17b)

### 2.6 Condenser analysis

For the simulation of the condenser, several critical assumptions were considered. First, all the heat (from the last stage of the) the steam turbine was dampened out through the condenser. Second, the pressure drop inside the condenser was discarded. Third, at the end of the condensing tubes, only the saturated liquid remained. Finally (as stated in the introduction), no exergy loss over the phase-change transformation (chemical entropy generation) of steam-water was considered. The assumption indicates that condenser energy efficiency is ideal (since all the heat was successfully delivered to the environment) while the exergy efficiency would be equal to zero (since all the turbine heat was dampened to the environment without further use). So, the energy and exergy loss in the condenser is calculated as follows:

\[ \dot{H}_{condenser} = \dot{H}_{condenser,use} = (1-y) \dot{m}_{steam} (h_s - h_1) \]  

(18a)

\[ \dot{E}_{x,condenser,loss} = T_{amb} s_{gen,condenser} = (1-y) \dot{m}_{steam} T_{amb} (s_s - s_1) \]  

(18b)

### 2.7 Low-pressure pump analysis

The low-pressure pump was located after the condenser, aiming to deliver the saturated subcooled water to the steam generator. The exergy efficiency was calculated from the results of the exergy and energy balance, which are given below:

\[ \dot{W}_{LPP,loss} = (1 - \eta_{LPP}) \dot{m}_{steam} (h_2 - h_1) \]  

(19a)

\[ \dot{E}_{x,LPP,loss} = \dot{E}_{x,LPP} - \dot{W}_{LPP,loss} = T_{amb} \dot{m}_{steam} (1-y) (s_2 - s_1) \]  

(19b)

The energy and exergy efficiencies are given by equations (20a, 20b), respectively.

\[ \eta_{1,LPP} = \frac{\dot{W}_{LPP,use}}{\dot{W}_{LPP}} \]  

(20a)

\[ \eta_{2,LPP} = \frac{\dot{W}_{LPP,use}}{\dot{E}_{x,LPP}} \]  

(20b)

### 2.8 Open feedwater heater (deaerator) analysis

A deaerator is responsible to remove oxygen and other dissolved gases from liquids before it entered the steam generator. Moreover, it provides an additional reheat over the subcooled water to increase the cycle’s performance. The vital assumption is that no heat exchanges occurred with the environment due to the careful insulation and, therefore, the energy efficiency is ideal. On the other hand, since all the exergy is transformed to heat with no further use, the exergy efficiency is considered zero. Thus, similar to the condenser analysis, the energy and exergy efficiencies are given in equations (21a, 21b), respectively.

\[ \dot{H}_{FWH,use} = (1-y) \dot{m}_{steam} (h_1 - h_2) = y \dot{m}_{steam} (h_6 - h_1) ; \dot{H}_{FWH,loss} = 0 \]  

(21a)

\[ \dot{E}_{x,FWH,loss} = \dot{m}_{steam} T_{amb} (s_3 - y s_6 - (1-y) s_2) \]  

(21b)

### 2.9 High-pressure pump analysis

The high-pressure pump was located after the deaerator to deliver the saturated subcooled water to the steam generator. By employing the same approach of the low-pressure pump, energy/exergy loss, and efficiency for the high-pressure pump is calculated as below:

\[ \dot{W}_{HPP,loss} = (1 - \eta_{HPP}) \dot{m}_{steam} (1-y) (h_4 - h_3) \]  

(22a)

\[ \dot{E}_{x,HPP,loss} = \dot{E}_{x,HPP} - \dot{W}_{HPP,loss} = T_{amb} \dot{m}_{steam} (1-y) (s_3 - (Q_{2b}) \]  

(22b)

\[ \eta_{en,HPP} = \frac{\dot{W}_{HPP,use}}{\dot{W}_{HPP}} \]  

(23a)

\[ \eta_{ex,HPP} = \frac{\dot{W}_{HPP,use}}{\dot{E}_{x,HPP}} \]  

(23b)

### 2.10 Power cycle analysis

Up to this section, comprehensive energy/exergy analysis was carried out separately for the components involved in the Rankine part of the cycle; however, an evaluation of Rankine components was studied for at least three reasons; first to verify the assumptions considered for the Rankine components, second to evaluate the dependency of the Rankine cycle to the investigated parameters, and third, to validate the current model with the state-of-the-art results presented in ref. In this regards, by applying the first and second
laws of thermodynamic for the whole Rankine cycle, the energy/exergy loss would be as follows:

\[
H_{ps,\text{loss}} = H_{sg,\text{use}} - \left( W_{\text{turb}} - W_{\text{HP}} - W_{\text{LPP}} \right) \frac{w_{\text{net}}}{\dot{W}_{\text{net}}} \quad (24a)
\]

\[
\dot{E}_{ps,\text{loss}} = \dot{E}_{sg,\text{use}} - \dot{W}_{\text{net}} \quad (24b)
\]

The energy and exergy efficiencies of the power cycle are given in Equation (25a, 25b), respectively.

\[
\eta_{en,ps} = \frac{\dot{W}_{\text{net}}}{H_{sg}} \quad (25a)
\]

\[
\eta_{ex,ps} = \frac{\dot{W}_{\text{net}}}{\dot{E}_{sg}} \quad (25b)
\]

### 2.11 | Total energy and exergy analysis

The total net energy/exergy efficiencies of the (power plant) cycle are calculated as follows:

\[
\eta_{en,total} = \eta_{para} \frac{\dot{W}_{\text{net}}}{H_{sun}} \quad (26a)
\]

\[
\eta_{ex,total} = \eta_{para} \frac{\dot{W}_{\text{net}}}{\dot{E}_{sun}} \quad (26b)
\]

where \( \eta_{para} \) is called parasitic efficiency of the whole power plant cycle and was assumed to be 89%, which is typical for these kinds of solar cycles coupled with Rankine power plants.\(^2\)

### 3 | RESULTS AND DISCUSSIONS

#### 3.1 | Validation of the model

As discussed in previous sections, the present model is based on the first and second laws of thermodynamics, or the energy and exergy balance of each component. The analysis of components such as the turbine, pumps, feedwater heater, and condenser in the Rankine cycle depended on the thermodynamic properties of the water/steam in the cycle. Additionally, the steam generator condition depended on both steam/water and molten salt properties.

The presented model was validated with the results of Xu et al.\(^2\) under similar conditions based on the properties provided by the mentioned authors presented in Table 1.

| TABLE 1 | Initial condition and parameters of the model |
|----------|-----------------------------------------------|
| Control volume | Properties                  | Values | Unit |
| Ambient condition | Ambient temperature     | 20     | °C   |
|                 | Sun temperature           | 4500   | K    |
|                 | Wind speed                | 5      | m/s  |
|                 | Sun irradiation           | 800    | W/m² |
|                 | Ambient pressure          | 101 325 | Pa |
| Heliostat field | Total Heliostat Field    | 10 000 | m²   |
|                 | Aperture area             | 12.5   | m²   |
|                 | Heliostat efficiency      | 0.75   | –    |
| Central receiver | Receiver height         | 6      | m    |
|                  | Concentration ratio       | 800    | –    |
|                  | Outer Tube Diameter       | 0.019  | m    |
|                  | Tube thickness            | 0.00165| m    |
|                  | Tube conduction           | 19.7   | W/m.K|
|                  | View Factor               | 0.8    | –    |
|                  | Surface Reflectivity      | 0.04   | –    |
|                  | Surface Emissivity        | 0.8    | –    |
|                  | Molten Salt inlet         | 290    | °C   |
|                  | Temperature               | 565    | °C   |
| Steam generator  | Water inlet              | 239    | °C   |
|                  | Temperature               | 552    | °C   |
|                  | Molten Salt Outlet        |        |      |
|                  | Temperature               | 5      | m/s  |
| General          | Stefan Boltzmann          | 5.67e-08 | W/m².K⁴ |
|                  | Insulation Thickness      | 0.07   | m    |

The results indicated that the calculated energy and exergy efficiencies of the receiver were 90.02% and 55.47%, respectively. These efficiencies were aligned with the results of Xu et al.\(^2\) Moreover, the overall energy and exergy efficiencies of the power plant were 22.83% and 24.42%, respectively. These efficiencies had 0.26% error for energy and 0.25% error for exergy analysis compared with 22.89% and 24.48% calculated in,\(^2\) which is appropriate for a user-generated code for these kinds of calculations. Therefore, the results of the current study were reasonable, and the assumptions made for every component were confirmed. It must be noted that the operation condition in each point of the state is also given in Table 2. Moreover, both the energy and the exergy analysis output in each component are given in Tables 3 and 4, respectively.

Finally, some assumptions were made regarding the energy/exergy efficiency of some components (such as pumps, condensers and steam generator) to validate the outputs with
the previous work of Xu et al.\textsuperscript{21} within the base analysis. In their simulation as discussed before, they considered the power cycle as a black box; therefore, the inner efficiencies of the power cycle components (as well as components directly related to the power cycle such as the steam generator) were considered having ideal performance. Fortunately, in the current study, no such restriction exists so that each component maintains its behavior as the operating condition varies.

### 3.2 General discussion

The base analysis of the cycle components (such as turbine, steam generator, deaerator, pumps, heliostat field, and receiver) points toward some interesting facts that should be discussed before the sensitivity analysis of the parameters. As depicted in Figure 3 (as well as Tables 3 and 3), exergy analysis was able to capture the magnitude of the loss occurred in each component of the cycle, as well as pointing out where the loss occurred. This was the advantage of second law analysis over the first law investigations. In this regards several significant points were discussed as below. First, the heliostat energy-exergy efficiency was equal to 75%. This was obtained since the field efficiency was considered 75% in ref.\textsuperscript{21} however, this does not mean that the amount of energy and exergy losses were equal. On the contrary, the calculations for the energy and exergy were separated and its fundamentals were discussed in section 2 thoroughly. As a final point, several improvements could have been made for the calculations of solar ray absorption in the heliostat component such as \textsuperscript{41,42} however, since the aim of this study was to evaluate the loss after the energy was absorbed rather than how it was absorbed, so these kinds of improvements were not considered in this research.

### Table 2 Properties of state points in base calculations

| State | Temperature (°C) | Pressure (bar) | Enthalpy (kJ/kg) | Entropy (kJ/kg.K) |
|-------|------------------|----------------|------------------|------------------|
| 1     | 45.80754821      | 0.1            | 191.8122952      | 0.649218083      |
| 2s    | 45.9157863       | 31.5           | 194.9670197      | 0.649218083      |
| 2     | 46.00810771      | 31.5           | 195.3523751      | 0.650483478      |
| 3     | 236.5795396      | 31.5           | 1021.249382      | 2.670592746      |
| 4s    | 238.7066688      | 126            | 1032.724391      | 2.670592746      |
| 4     | 239              | 126            | 1034.042         | 2.673106641      |
| 5     | 552              | 126            | 3480.728704      | 6.633345908      |
| 6s    | 328.3674476      | 31.5           | 3060.696313      | 6.633345908      |
| 6     | 354.3558686      | 31.5           | 3123.044871      | 6.734830453      |
| 7s    | 45.80754821      | 0.1            | 2100.492344      | 6.633345908      |
| 7     | 45.80754821      | 0.1            | 2305.371179      | 7.275685965      |

### Table 3 Energy analysis output in base calculations

| Power plant system | Total energy (kW) | Used energy (kJ) | Loss energy (kJ) | Energy efficiency |
|--------------------|-------------------|------------------|------------------|-------------------|
| Heliostat Field    | 800 000           | 600 000          | 200 000          | 0.75              |
| Solar Receiver     | 600 000           | 540 159.6202     | 59 840.37985     | 0.900266034       |
| Steam Generator    | 540 159.6202      | 540 159.6202     | 0                | 1                 |
| Steam Turbine      | 540 159.6202      | 208 561.6895     | 334 983.2571     | 0.386111219       |
| Condenser          | 334 983.2571      | 334 983.2571     | 0                | 1                 |
| LP Pump            | 561.0761548       | 561.0761548      | 0                | 1                 |
| Deaerator          | 130 898.4915      | 130 898.4915     | 0                | 1                 |
| HP Pump            | 2824.250267       | 2824.250267      | 0                | 1                 |
| Rankine Cycle      | 540 159.6202      | 205 176.363      | 334 983.2571     | 0.379843949       |
| Power Plant (overall) | 800 000           | 182 606.9631     | 617 393.0369     | 0.228258704       |
Second, the energy and exergy efficiency of the receiver were measured 90.02% and 55.47%, respectively. As the results indicated, even though the energy efficiency was above 90%, the receiver’s exergy performance was not efficient since nearly half of the exergy was lost during the cycle operation. A separate part is embedded in the next section, which deals with the values of loss and root of each. However here in a brief discussion, the most loss contributed to the emission loss with 34.28%, and then to the reflection loss with a value of 32.08%.

Third, even though the steam generator performance was considered ideal (the heat loss would become zero), the exergy efficiency was obtained 91.21%. This proves that for the best-case scenario nearly 10% of the exergy was lost; therefore, the steam generator component is a potential source of exergy loss.

Fourth, the turbine energy and exergy efficiencies were 38.61% and 62.30%, respectively. Unlike other components, exergy efficiency is higher than energy efficiency. Generally, in Rankine-based power plants, the steam turbine energy efficiency varies from 15% for low tech to near 55% for extremely high tech steam turbines. The performance of the turbine cannot exceed a certain amount for several reasons. For instance, the quality of the steam should not be less than 88% because it could harm the ending floor blade of the turbine, or the inlet steam could not be heated over a certain amount of temperature due to heat tension.43 Due to the reasonable energy and exergy efficiencies obtained for the steam turbine, not much ambiguity observed for further investigations. Moreover, the exergy efficiency was acceptable and the energy and the energy efficiency were typical for a steam turbine with this kind of cycle.44

Fifth, the overall calculations demonstrated that from 8MW%7.478 MW input energy%exergy from the sun, only 1.82MW output electricity was generated, meaning that 6.17 MW%5.6527 MW amount of energy%exergy was lost. The energy and exergy efficiency of the Rankine subsystem was 37.98% and 72.81%, respectively, and the energy and exergy efficiency of the solar subsystem was 67.25% and 41.60%, respectively. The overall energy and exergy efficiencies were 22.82% and 24.42%, respectively, in which these calculated amounts were less than the separated solar subsystems.

### TABLE 4 Exergy analysis outputs in base calculations

| Power Plant System       | Total Exergy (kW) | Used Exergy (kW) | Loss Exergy (kW) | Exergy Efficiency |
|--------------------------|------------------|------------------|------------------|-------------------|
| Heliostat field          | 747 884.4444     | 560 913.3333     | 186 971.1111     | 0.75              |
| Solar receiver           | 560 913.3333     | 311 188.1019     | 249 725.2315     | 0.554788206       |
| Steam generator          | 311 188.1019     | 283 855.8011     | 27 332.30081     | 0.912167912       |
| Steam turbine            | 283 855.8011     | 176 864.4237     | 106 991.3774     | 0.623078419       |
| Condenser                | 27 103.78914     | 0                | 27 103.78914     | 0.554788206       |
| LP Pump                  | 2824.250267      | 502.2832918      | 58 79286306      | 0.895214112       |
| Deaerator                | 19 656.89276     | 0                | 19 656.89276     | 0.554788206       |
| HP Pump                  | 2824.250267      | 2661.552801      | 162.697466       | 0.942392688       |
| Rankine cycle            | 283 855.8011     | 205 176.363      | 78 679.43802     | 0.722818989       |
| Power plant (overall)    | 747 884.4444     | 182 606.9631     | 565 277.4813     | 0.244164676       |

### FIGURE 3 The results of energy and exergy efficiencies of different components

The results of energy and exergy efficiencies of different components.
Finally, unlike the energy analysis, which indicates that the Rankine cycle is the main source of loss, the exergy analysis points out that the solar subsystem (especially the receiver) is responsible for the output loss (presented in Table 5). Stated before, the energy efficiency of the Rankine cycle was 37.98%, while the exergy efficiency obtained to be 72.281%. This fact indicates that even though not much work is extracted from the heat steam, however, from the extracted amount of work, a significant part of it transferred to useful work. Therefore, even though the significant energy loss occurred inside the turbine component (since no optimization technique is used such as reheat), almost all the extracted energy was transferred into the useful work. On the other hand, for the receiver, even though 90.02% of the energy was captured from the heliostats, nearly half of it (55.47%) was efficiently transferred to the molten salt. This leads to the conclusion that the solar subsystem was not as efficient as the energy analysis proved to be; however, this did not prove that the combination of solar and conventional subsystems was not efficient. Even a small improvement in receiver or heliostat efficiency would result in an efficient and environmentally clean power plant.

To discuss more, the effect of various parameters is explained within the subsequent sections. Moreover, for some parameters, which have maximum or minimum criteria, especial attention was toward finding an optimum value to minimize the losses.

### 3.3 Effect of direct normal irradiation (DNI)

The efficiency of a solar power plant highly depends on the incident of solar radiation. To evaluate its degree of sensitivity, a wide range of DNI values was investigated. As depicted in Figure 4, the increase in DNI resulted in increasing the exergy%energy efficiency of the power plant; however, the rate of increase has been approached to zero after a certain amount of DNI value (600 W.m\(^{-2}\)). To justify the reason, the efficiency of all cycle components has been investigated. Eventually, an extreme sensitivity regarding the variation of DNI captured from the central receiver. The same trend of energy%exergy efficiency was observed, indicating that the central receiver was the controlling factor influencing the overall efficiency of the cycle. As shown in Figure 4, increasing the DNI value up to 1200 W.m\(^{-2}\) results in the compensation of 90% of the energy and 40% of the exergy loss.

#### Table 5

| Irreversibility Component system | Exergy loss (kW) |
|---------------------------------|------------------|
| Solar heliostat field           | 1869.711111      |
| Solar receiver                  | 2497.252315      |
| Steam generator                 | 273.320081       |
| Steam turbine                   | 1069.913774      |
| Condenser                       | 271.0378914      |
| LP Pump                         | 58.79286306      |
| Deaerator                       | 196.5689276      |
| HP Pump                         | 162.697466       |
| Rankine cycle                   | 786.7943802      |
| Power plant                     | 5652.774813      |
Same as overall (energy%exergy) behavior, the efficiency of the receiver approached a constant value after the DNI value reached 600 W·m⁻².

To understand how the loss is occurred inside the receiver, further details of the heat transfer phenomenon were investigated. Based on Figure 5, it was found out that by increasing the DNI value, the portion of the loss that was associated to the reflective heat transfer (loss) was intensively increased, while all other forms of the energy (emission, convection, and conduction) loss were observed to reduce. This is reasonable since more emission from the sun (more DNI) leads to a higher amount of energy (loss) reflected in the environment. A valuable conclusion out of this analysis is that for a solar power plant with an average DNI value of 600-800 W·m⁻², only by optimization on the reflective loss, nearly 18% of the energy and 15% of the overall loss could be removed from the cycle, therefore increasing the net output of the solar power plants.

Finally, a brief investigation was carried out to evaluate the influence of DNI on the net output of the cycle as well as the receiver surface temperature (RST). As shown in Figure 6, increasing DNI boosted both of the studied parameters linearly, since it would indicate that utilizing more heliostats (more DNI captured) linearly increases the output of a solar-based power plant.

### 3.4 Effect of molten salt Outlet temperature

In this section, the performance of the cycle regarding the transfer of that heat to the molten salt fluid (on the solar sub-system) is investigated. To accomplish this purpose, a wide range of temperatures (500-800°C) for the molten salt at the outlet of the receiver was considered. As shown in Figure 7, while the outlet temperature of molten salt is increased (MSOT), the total energy and exergy efficiency of the cycle would be decreased by 2%-3%. Moreover, other findings indicated that 10°C increase in temperature reduced the cycle’s efficiency by 0.033%.

Further investigation revealed that despite the DNI section, the energy and the exergy efficiencies of the receiver were responding in the opposite direction. This observation leads the subsequent efforts toward finding an optimum value of MSOT that not only minimizes the energy loss but also gives us the maximum possible exergy efficiency. As depicted in
Figure 8, the optimum value was found to be 650°C, which was the crossing point of energy and exergy efficiency curves. Afterward, a detailed investigation was carried out to identify the loss sources within the central receiver. To do so, all the possible forms of the heat transfers that would cause energy-exergy loss were considered. As shown in Figure 9, despite the previous (DNI) analysis, the reflective loss was not the main effective source of waste since its degree of influence was decreasing as the MSOT values increased. It was observed that the most contributing form of loss was the emission loss which was increased by 15% as the MSOT increased. Moreover, other forms of heat loss (such as convection and conduction loss) were not quite influenceable.

As the final test, the effects of MSOT on the net output of the cycle and RST were studied. As depicted in Figure 10, by increasing the MSOT, the net output of the cycle decreased. This was previously predicted since the overall energy-exergy efficiency of the cycle was decaying per increase of MSOT. Additionally, from the crossing point of RST and net output curvatures, the optimum value of MSOT is found 650°C (from this figure) which was the same as the value calculated from Figure 7. The reason is laid upon the fact that the lowest amount of RST and the highest amount of net outputs are equivalent of minimize the energy loss while maximizing the exergy efficiency.

3.5 Effect of molten salt velocity (MSV) inside the pipe

The velocity of the molten salt inside the pipelines of the solar subsystem was also another parameter that was not studied throughout the literature. To this aim, a wide range of 1-10 m/s velocities was taken into consideration. Depicted in Figure 11, increasing the MSV results in lowering the losses occurred inside the system; therefore, the overall efficiency was increased. By increasing the MSV (from 1 to 10 m/s), the overall energy efficiency increased from 22.2% to 22.9%, and the overall exergy loss efficiency decreased from 23.2% to 24.5%. Further investigations revealed that the central receiver was the main deficiency of the cycle; therefore, only the loss performance of the receiver was depicted in Figure 11. Unfortunately, several important considerations in the receiver’s performance proved to be a clear restriction for further analysis. For instance, for MSV lower than 3 m/s, solidification is assumed to arise an issue while for MSV higher than 7 m/s the adequate time for the heat exchanging process inside the heat exchanger might be shortened; resulting in, the efficiency decrease of the cycle. By knowing this, the subsequent investigations were focused to identify the optimum value of MSV inside the cycle.

Presented in Figure 12, several notable conclusions could be achieved. First, expectedly, by increasing the MSV, the net output of the cycle was increased to a certain value. This was reasonable since higher MSV resulted in higher amounts of steam for the Rankine cycle. On the other hand, increasing the MSV resulted in a decrease of RST values. The reason was found to be that increasing the MSV shortened the time required for the heat transfer between the receiver and the molten salt.

Second, the rate of change for both parameters was approaching zero for velocities higher than 8 m/s; proving that there was a limited effective range for optimizing the parameters. Moreover, indicating that further alteration of MSV has a small impact on the cycle’s performance.

Finally, based on the crossing points of the net output and RST curves, the optimum value for the MSV was found to be 2 m/s. Critic readers might question the reason(s) to select net output and the RST as the cycle’s responding attributes to the variation of MSV.

First, the net output of the cycle is a common parameter that is targeted to be increased in almost all power plant. Here also, maximizing the net output (without changing Rankine-based parameters) was indeed the final goal. Second, as discussed in previous sections, increasing RST would inevitably increase the loss through various heat transfer mechanisms (mainly reflective loss and emissive loss). Therefore, minimizing the RST would inevitably reduce the loss. Finally, the selection of these parameters allowed us to investigate to lower a parameter corresponding to the losses (RST) while maximizing a parameter corresponding to the efficiency (achieving maximum net output). In this regard, the optimum value of MSV was shown to be 2 m/s (Figure 12).

Interestingly, for the previous investigation, the optimum RST was found to be 550°C (Figure 10). Here again, for the
optimum MSV, the RST value was found to be 550°C which points out the validity (and accuracy) of the current optimization. Afterward, the only remaining task was to investigate the heat loss mechanisms inside the receiver. As depicted in Figure 13, increasing the MSV decreases the emission loss significantly (8.2% increasing), while the reflective loss was increased intensively (9%). The reason was corresponding to the fact increasing MSV results in decreasing of RST (discussed before in Figure 12). Since the emission is significantly related to RST (power 4), therefore by lowering RST, emission loss would have a less portion loss mechanism compared to the reflective loss. Moreover, other loss mechanisms such as convection and conduction loss were also increased (3% each), since a higher amount of MSV yields to higher amounts of heat transferred through convection%conduction to the environment.

### 3.6 Effect of Wind Speed

Another critical parameter affecting the receiver efficiency was wind speed. To our knowledge, this parameter was not investigated in any other study. Therefore, the results might
introduce much more insights into the phenomenon. To this aim, a wide range of wind speed from 0 to 15 m/s was studied.

It must be noted that 10-15 m/s is quite high values for wind speed (rare case); however, a wide range of wind speed was utilized to magnify the cycle’s response so that the physical phenomenon was better understood. As illustrated in Figure 14, the results indicated that efficiency decreased by almost 4% for energy and 2% for exergy inside the receiver meaning that by increasing the wind speed by 1 m/s, 0.25% decrease in energy and 0.12% decrease in exergy efficiency has occurred. It was also observed that wind speed was less effective on overall efficiencies. In this regard, energy efficiency decreased from 24% to 22%, and the exergy efficiency decreased from 26% to 24%. It means that by 1 m/s increase in wind speed, a 0.12% decrease of energy/exergy efficiency has been observed.

Same as before, subsequent investigations were to analyze the loss of the receiver respect to the various wind speed. As shown in Figure 15, the convection and conduction loss were increased 14.8% and 9.6% respectively, while other sources of losses (reflecting and emission) were decreasing. This was expected since the overall heat transfer coefficient (mainly convective) entirely relied on wind speed and as the wind speed increased, the related losses increased. To this end, it was found that a 1.05% increase in convection loss and a 1% decrease in other sources of losses occurred per 1 m/s changes due to the wind speed. Therefore, as a valuable take out, it is evident that to improve the performance of the solar cycle within a windy climate, a block body mechanism to reduce the effect of wind speed would increase the receivers’ performance. As the final investigation, the effect of wind speed on the net output of the power plant is investigated.

Illustrated in Figure 16, the net output of the cycle decreased (close to a linear fashion) from 2.08 to 2.01 MW as the wind speed increased (467 kW net output decreasing per 1 m/s increasing of wind speed). Therefore, wind speed is quite an impression on the net output of the cycle.
3.7 Effect of Ambient Temperature

The final environmental parameter that was considered for the investigation was the ambient temperature. To evaluate the sensitivity of the cycle to this parameter, a wide range of temperatures from 5 to 50°C was utilized. According to Figure 17, the energy efficiency decreased from 90.12% to 90.8% (about 0.04%), and the exergy efficiency decreased by 6% from 58% to 52% for the receiver. The overall cycle’s performance exhibited even less variation regarding this parameter. Further investigations regarding the loss sources depicted in Figure 18 also revealed that both reflective and emission losses were increasing by 4% while the convection and conduction loss were decreasing again by 4%. The rate of changes for a wide range of temperature was quite low (compared with the previous cases), meaning that heat loss is almost independent respect to the variations of ambient temperature.

4 CONCLUSION

In this research, by considering exergy%energy perspective, performance of the solar-based cycle was investigated via a (molten salt) steam generator instead of a conventional boiler. For this purpose, a numerical code was developed to cover both the thermodynamic properties and the calculations of the cycle components. Based on the analyses, the central receiver was proven to have a significant contribution to the losses since only half of the captured energy from the heliostats was successfully transferred into the molten salt pipes. Moreover, several investigations regarding the sensitivity of the cycle efficiency respect to parameters such as DNI, MSOT, MSV, wind speed, and ambient temperature were carried out. Regarding the influence of DNI, it was proven that although increase in DNI improved overall efficiency of both cycle and the receiver, nevertheless, the rate of changes approached to zero as the DNI increased. Further investigations also indicated that the increase of DNI results in higher contribution of the emission loss compared to other forms of loss. Regarding the influence of MSOT and MSV, it was proven that increasing the MSV leads to the enhancement of the cycle efficiencies while on the contrary increasing MSOT results in somewhat decrement in the cycle energy%exergy efficiency. To remove the ill-influence of the noted parameters, the optimum value of MSV and MSOT was found to be 2 m%s and 650°C, respectively. Moreover, the highest loss was found to be reflective and emitting loss in these cases. Regarding the influence of wind speed, it was found out that due to the increase of convection loss (14.8%), the receiver energy and exergy efficiencies were reduced by 4% and 2%, respectively, as the wind speed increase. Last but not least, it was found out although the cycle’s energy response was quite low to the impact of ambient temperature, it had an obvious negative influence on the receiver’s exergy efficiency that was reduced from 57% to 52%.

Nomenclature

\[ A \quad \text{Area (m}^2\text{)} \]
\[ C \quad \text{Concentration ratio} \]
\[ d \quad \text{Diameter (m)} \]
| Symbol | Description |
|--------|-------------|
| $\dot{I}$ | Rate of irreversibility (W) |
| $Fr$ | View factor |
| $h$ | Specific enthalpy (J%(kg.k)) heat transfer coefficient (W%(m².K)) |
| $H$ | Rate of heat (W%(K)) |
| $HPP$ | High-pressure pump |
| $LPP$ | Low-pressure pump |
| $m$ | Mass flow rate (kg%s) |
| $MSV$ | Molten salt velocity (m%s) |
| $MSOT$ | Molten salt outlet temperature (°C) |
| $RST$ | Receiver surface temperature (°C) |
| $Nu$ | Nusselt number |
| $Pr$ | Prandtl number |
| $En$ | Energy (W) |
| $Ex$ | Exergy (W) |
| $T$ | Temperature (°C) |
| $s$ | Specific entropy (J%(kg.k)) |
| $S^*_{gen}$ | Rate of generation entropy (W%K) |
| $SIR$ | Irradiation of the sun (W%m²) |
| $\dot{W}$ | Work (J%(kg K)) |

**Greek symbols**

| Symbol | Description |
|--------|-------------|
| $\eta_{en}$ | Energy efficiency |
| $\eta_{ex}$ | Exergy efficiency |
| $\varepsilon$ | Receiver surface emissivity |
| $\delta$ | Thickness (m) |
| $\lambda$ | Thermal conductivity (W%(m.K)) |
| $\rho$ | Density (kg%m³) |
| $\sigma$ | Stefan Boltzmann constant 5.67e-8 (W%(m².K⁴)) |

**Subscripts**

| Symbol | Description |
|--------|-------------|
| $a$ | At the receiver inlet |
| $b$ | At the receiver outlet |
| $amb$ | Ambient |
| $con$ | Conductive heat loss |
| $conv$ | Convective heat loss |
| $em$ | Emissive heat loss |
| $force$ | Force convection |
| $free$ | Free convection |
| $hel$ | Heliotat |
| $i$ | Inlet |
| $inlet$ | At the inlet |
| $insu$ | Insulation |
| $loss$ | Heat loss |
| $ms$ | Molten salt |
| $o$ | Outlet |
| $outlet$ | At the outlet |
| $rec$ | Receiver |
| $ref$ | Reflective heat loss |
| $sg$ | Steam generator |
| $surf$ | Surface |
| $turb$ | Turbine |
| $use$ | Useful fraction of energy%exergy |
| $w$ | Wall surface |

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