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## Article

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## Research paper

# Exergy and economic analyses of replacing feedwater heaters in a Rankine cycle with parabolic trough collectors

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## HIGHLIGHTS

- Exergy and economic analyses are performed on a solar implemented Rankine cycle.
- A thermal storage system is added to the cycle to enable it for 24 hour operation.
- Total power generated of the system increases by 8.14% compared to base case.
- Exergy analysis revealed that boiler has the highest rate of exergy destruction.
- Payback time of the proposed system is equal to 1.5 year.

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## ABSTRACT

Fossil fuels are exhaustible and their consumption causes environmental problems. As a result, renewable energy resources, specifically solar energy, should be utilized more to overcome the aforementioned issues. The biggest problem with renewable energy utilization is that their capital cost is very high. To tackle this problem, renewable energies should be coupled with conventional energy systems to lower the initial investment. In this paper, a parabolic trough collector is coupled with a conventional Rankine cycle to increase output power of the system by replacing its closed feedwater heaters. Also, to be able to use the system during the nights, a thermal storage system is added to the cycle. A complete energy, exergy and economic analyses are performed on the system and the results are compared with the base case condition. The results show that by using the proposed system, net generated power of the plant increases by 8.14%. Also, exergy analysis shows that in both cases boiler has the highest rate of exergy destruction. In general, due to huge amount of losses in the collector, exergy efficiency of the system decreases. Finally, economic analysis shows that simple payback time of the system equals to 1.5 years which is very low.

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## 1. Introduction

In order to evaluate the energy systems, exergy analysis is a great method which discloses the inefficient thermodynamic processes. In recent years, exergy analysis has turned into a fundamental issue in presenting a superior comprehension of the processes, to evaluate the inefficiency's sources and to discriminate quality of energy consumption (Dincer and Al-Muslim, 2001; Jin et al., 1997;

Kotas, 2013). In order to determine the type, location and genuine size of exergy losses (or destruction), it is notable that exergy can be utilized. Therefore, a crucial role in progressing approaches and in presenting guidelines for more efficient utilization of energy has been played by the exergy in the existing power plants (Jin et al., 1997). Additionally, the origin of the exergy loss is another imperative issue for enhancing existing systems. Consequently, a conspicuous picture, instead of only the magnitude of exergy loss in each section, is required. Moreover, the utilization of exergy analysis in evaluating the thermal power plants is predominantly acceptable. In order to examine the effect of reheat temperature and pressure on the performance of regenerative cycle, an analysis of the second law on a steam turbine power cycle has been performed by Sciubba

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### Nomenclature

$\dot{E}_x$	Exergy flow rate (kW)
$\dot{E}_x^D$	Exergy destruction
$\dot{E}_x^Q$	Exergy associated with heat
$\dot{E}_x^W$	Exergy associated with work
$ex$	Specific exergy (kJ/kg)
$ex_i^{ch}$	Standard chemical exergy of <i>i</i> th component
$h$	Specific enthalpy (kJ/kg)
LHV	Lower heating value of fuel (kJ/kg)
$\dot{m}$	Mass flow rate (kg/s)
$P$	Pressure (bar)
$\dot{Q}$	Heat transferred (kW)
$R$	Gas constant (kJ/kg K)
$s$	Specific entropy (kJ/kg K)
$T$	Temperature (K)
$T_r$	Heat transfer temperature
$\dot{W}$	Work rate (kW)
$y$	Mole fraction
$Y$	Share of exergy destruction

### Greek symbols

$\psi$	Exergy efficiency
$\xi$	Chemical exergy/Energy ratio

### Subscripts

0	Reference environment condition
ch	Chemical
cond	Condenser
c.v.	Control volume
FWH	Feed water heater
in	Inlet stream
ph	Physical
out	Outlet stream
ST	Steam turbine

and Su (1986). An analysis of the optimal performance on a combined Carnot cycle (two single Carnot cycles in cascade), containing internal irreversibilities for steady-state operation has been carried out by Şahin and Kodal (1995). Furthermore, the maximum power and efficiency have been analytically achieved and the influences of irreversibility parameters on maximum power output have been demonstrated.

Rosen and Dincer (2001) presented exergy analysis as the exceptional method for determining on optimization of the cycle regarding the input information. Additionally, the boiler's energy and exergy analyses in a steam power plant have been performed by Amir (2012). According to the results, reducing the fraction of combustion excess air from 0.4 to 0.15 leads to 0.19% and 0.37% increase in energy and exergy efficiencies, respectively. Furthermore, the mentioned efficiencies respectively enhance to 0.84% and 2.3% by decreasing the temperature of the exhaust gases leaving the chimney from 137 °C to 90 °C. Also, with a specific end goal of developing an optimization plan, the energy and exergy analyses for every part of a heat recovery steam generator in combined-cycle power plant have been provided by Kaviri et al. (2013). Drum pressure and arrangement of heat exchangers in the heat recovery steam generator with respect to high and low-pressure parts were the examined parameters in the research. Additionally, the exergy analysis of an existing 27 MW binary geothermal power plant in which isobutene operates as the working fluid and liquid-dominated heat source at 160 °C has been carried out by Kanoglu

and Bolatturk (2008). According to the results, the exergy and energy efficiencies were 33.5% and 10.2%, respectively. Also, an exergoeconomic analysis on a steam power plant has been performed by Bolatturk et al. (2015) in Turkey. By considering both design and off-design conditions of a steam power plant, Ray et al. (2010) presented exergy analysis of the power plant. As the results indicated, a superior principle to reflect the degradation of the system is the second law of thermodynamics. The location where the most exergy destruction occurs is combustion chamber as the result of chemical reaction circumstance. Furthermore, the investigation of the combustion processes and its exergy analysis have been provided by Taniguchi et al. (2005). The simulation of gas turbine power plant in Mahshahr and its optimization with respect to economic and exergy have been presented by Almasi (2011). Kaviri et al. (2012) modeled a dual pressure combined cycle power plant equipped with a duct burner and optimized the plant based on exergy and economic objective functions. According to the results, the performance of the plant has been considerably affected by compressor pressure ratio, gas turbine inlet temperature and pinch point temperatures.

In addition to the benefits of exergetic performance analysis in determining the magnitudes, location and causes of irreversibilities in the plants, it provides more considerable evaluation of the individual elements efficiency of the plant (Kaushik et al., 2011). A review of energy and exergy analyses for distinct power plants has been presented by Reddy et al. (2010). Also, an analysis of the second law founded on the exergy concept for a solar thermal power system has been provided by Kaushik et al. (2000). In order to demonstrate the different thermodynamic and thermal losses, related exergy and energy flow diagrams have been drawn. According to the results, the maximum energy loss happens at the condenser of the heat engine part, while the main exergy loss happens at the collector–receiver assembly. Additionally, analyses of energy and exergy for the distinct elements of a presented theoretical direct steam generation solar thermal power plant have been carried out by Gupta and Kaushik (2010). As the results indicated, the main energy loss occurs in the condenser followed by the solar collector field, whereas the main exergy loss takes place in the solar collector field and the exergy loss in other elements of the plant is insignificant. Also, a thermodynamic assessment in various designs of coupling parabolic-trough solar power plants and desalination facilities in a dry location have been presented by Palenzuela et al. (2011). Blanco-Marigorta et al. (2011) performed an investigation of two distinct cooling technologies and an exergy analysis for the power cycle of a 50 MW<sub>e</sub> solar thermal power plant. Furthermore, the impact of ambient temperature on the exergy efficiency of Catalagzi power plant in Turkey has been investigated by Kopac and Hilalci (2007). According to the results, the maximum exergy losses take place in the boiler, while the main energy losses occurs in the condenser. They also discovered that an increment in ambient temperature reduces the exergy efficiency of all the components of a power plant except the condenser. In this study, the condenser pressure at various ambient temperature has been assumed to be constant, while is not consistent in the actual situation. Farhad et al. (2008) designed a method to reduce irreversibility of the feedwater heaters network in steam power plants based on pinch and exergy analyses. Applying this method leads to increase the exergy efficiency of the system components and the total system. Popov (2011) used Fresnel collectors to directly increase temperature of boiler's feedwater and therefore eliminating feedwater heaters. Four different scenarios were considered to couple the Fresnel collector with the fossil fuel power plant and the best one was selected. Reddy et al. (2012) coupled Fresnel collectors with a combined cycle power plant and performed energy and exergy analyses on it. They showed that by using the proposed system, net output power of the system increases by

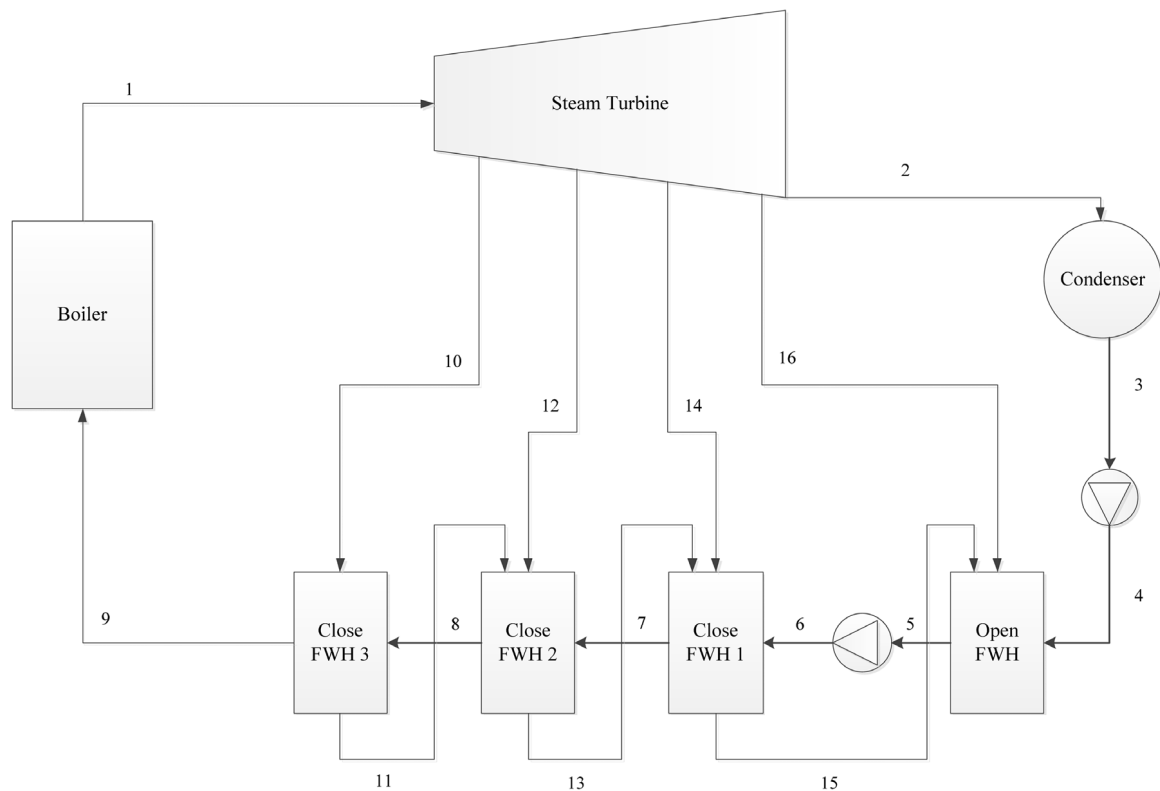


Fig. 1. Schematic diagram of the power plant.

approximately 10%. Hu et al. (2010) integrated a solar collector to a conventional coal-fired power station. The techno-economic study of the plant demonstrated a successful operation of about 354 MW capacity solar electricity generation scheme in California, USA (Price et al., 2002).

As mentioned above, there are some published papers about integrating solar collectors to power cycles and most of them used either Fresnel or flat plate collector. The most important drawback of these systems is that they only operate during the day and during the nights in which there is no radiation, they should be bypassed. Therefore, their availability is low. Also, temperature range of these types of collectors is not high. To resolve these issues, we propose to integrate parabolic trough collector with thermal storage to a simple Rankine cycle to increase its performance. To analyze system performance, energy, exergy and economic analyses are performed in the system and the results are compared with the base case condition.

## 2. System description

Zarand power plant located in Iran selected as the case study to perform the analysis. Fig. 1 shows the schematic diagram of this power plant. It is a steam power plant with nominal capacity of approximately 30 MW. The water is converted into steam in a boiler which uses natural gas as fuel. The steam expands in a turbine and produces power. Turbine outlet stream condenses in a water cooled condenser and then goes through an open feedwater heater. The main objective of this component is to separate dissolved gases in water. After that, three close feedwater heaters are used to increase feedwater temperature. To do this, a proportion of steam is extracted from turbine and used in the close feedwater heaters as heat source. Although this reduces net output power of the cycle by decreasing steam mass flow rate in the turbine, it raises efficiency of the plant by reducing its fuel consumption in the boiler.

To boost net output power of the plant, close feedwater heaters are replaced with a series of solar collectors. There are three types of collectors that can be integrated with Rankine cycle, including flat plate collectors, linear Fresnel collectors and parabolic trough collectors. In this paper, parabolic trough collectors are used. The main advantage of these collectors is that they can achieve higher temperatures compared with flat plates and linear Fresnel collectors. Another substantial difference is that linear Fresnel collectors are usually designed for heating water, while parabolic trough collectors have the capability to use a wide range of heat transfer fluids. This ability can be used to store heat in a thermal storage system and use it during the nights in which sun's radiation is not available. This is very important in integration of solar fields with the Rankine cycle. Because when no thermal storage system is available, the plant can use solar energy only during the day. In this condition, the surplus steam (which is not extracted from turbine) could damage the turbine by overloading its stages. To prevent this from happening, steam flow rate at turbine inlet has to be reduced so that the volumetric flow of steam in the last turbine stage is equal to its design point. As a result, adding a thermal storage system is very beneficial for the plant and plays a key role in enhancing its net output power and its efficiency. Fig. 2 shows the new schematic diagram of the plant after adding solar field and thermal storage system. In this situation, only one stream is extracted from the turbine to provide the heat source for open feedwater heater. The three close feedwater heaters are replaced with heat exchangers which use heat transfer fluids as their heat source. In this condition, the steam which was used to increase temperature of the feedwater expands in the turbine to produce more power. Also to store heat, a thermal storage system is utilized in the system which is comprised of a hot and a cold tank. The heat stored in these tanks is transferred to the heat transfer fluid through a heat exchanger. Therminol 66 is used as heat transfer fluid in the cycle.

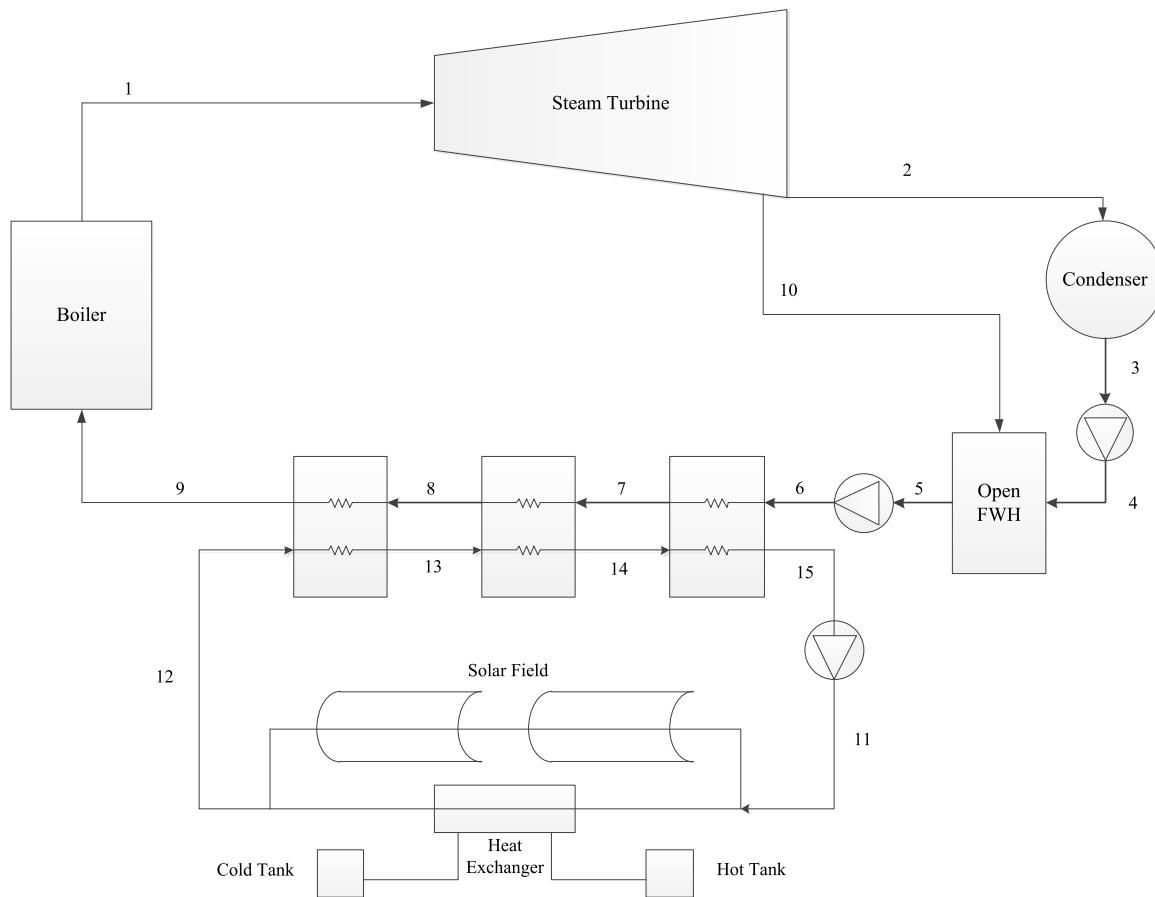


Fig. 2. Schematic diagram of the plant after adding solar collectors and thermal storage system.

### 3. Exergy

Exergy is the maximum work which can be obtained from a given form of energy using the environmental parameters as the reference state (Kotas, 2013). It is a better criterion to evaluate the performance of systems than energy analysis, because energy analysis considers energy transfer to the environment as the only source of inefficiency, whereas exergy analysis considers internal irreversibility of the system as well as the losses. In general, exergy balance for a control volume is represented as follows:

$$\dot{E}x^Q + \sum \dot{m}_{in} ex_{in} = \dot{E}x^W + \sum \dot{m}_{out} ex_{out} + \dot{E}x^D \quad (1)$$

Based on this equation, exergy could be transferred by work, heat and mass. In Eq. (1),  $\dot{E}x^D$  represents exergy destruction and  $\dot{E}x^Q$  and  $\dot{E}x^W$  are the exergy associated with heat and work, respectively and defined as follows (Bejan and Tsatsaronis, 1996; Dincer and Cengel, 2001):

$$\dot{E}x^Q = \left(1 - \frac{T_0}{T_r}\right) \dot{Q} \quad (2)$$

$$\dot{E}x^W = \dot{W}_{c.v.} \quad (3)$$

where  $T_0$  shows ambient temperature and  $T_r$  is the temperature in which heat transfer is occurred. The other parameter in Eq. (1) is exergy transfer through mass transfer. Neglecting the potential and kinetic exergies, exergy associated to mass transfer is comprised of

physical and chemical exergies as follows:

$$ex = ex_{ph} + ex_{ch} \quad (4)$$

Physical exergy is defined as:

$$ex_{ph} = (h - h_0) - T_0 (s - s_0) \quad (5)$$

where  $h$ ,  $s$  and  $T$  are enthalpy, entropy and temperature, respectively. Also "0" refers to the ambient condition. Chemical exergy is calculated using Eq. (6):

$$ex_{ch} = \sum_{i=1}^N y_i ex_i^{ch} + RT_0 \left( \sum_{i=1}^N y_i \ln(y_i) \right) \quad (6)$$

where  $y_i$  is the molar fraction and  $ex_i^{ch}$  is standard chemical exergy of each component. Also  $R$  is the universal gas constant. Calculating chemical exergy of fuels using the above equation is too complex. For simplicity, Eq. (7) is used for gaseous fuels (Dincer and Cengel, 2001):

$$\xi = \frac{ex_{fuel}}{LHV_{fuel}} \quad (7)$$

Normally the value of  $\xi$  is close to unity for the most of commonly used fuel gases. For example (Dincer and Cengel, 2001):

$$\begin{aligned} \xi_{CH_4} &= 1.06 \\ \xi_{H_2} &= 0.985 \end{aligned} \quad (8)$$



**Table 1**  
Exergy destruction and efficiency of each component of the cycle (Ameri et al., 2008).

Component	Exergy destruction	Exergy efficiency
Feed water heater	$\dot{E}x_{FWH}^D = \sum \dot{E}x_{in} - \sum \dot{E}x_{out}$	$\psi_{FWH} = \frac{(\dot{E}x_{out} - \dot{E}x_{in})_{feedwater}}{(\dot{E}x_{in} - \dot{E}x_{out})_{steam}}$
Steam turbine	$\dot{E}x_{ST}^D = \dot{E}x_{in} - \dot{E}x_{out} - \dot{W}_{ST}$	$\psi_{ST} = \frac{\dot{W}_{ST}}{\dot{E}x_{in} - \dot{E}x_{out}}$
Condenser	$\dot{E}x_{Cond}^D = \sum \dot{E}x_{in} - \sum \dot{E}x_{out}$	$\psi_{cond} = \frac{(\dot{E}x_{out} - \dot{E}x_{in})_{cooling\ water}}{(\dot{E}x_{in} - \dot{E}x_{out})_{steam}}$
Pump	$\dot{E}x_{pump}^D = \dot{E}x_{in} - \dot{E}x_{out} + \dot{W}_{pump}$	$\psi_{pump} = \frac{\dot{E}x_{out} - \dot{E}x_{in}}{\dot{W}_{pump}}$
Boiler	$\dot{E}x_{Boiler}^D = \sum \dot{E}x_{in} + \dot{E}x_{fuel} - \sum \dot{E}x_{out}$	$\psi_{boiler} = \frac{\dot{E}x_{out} - \dot{E}x_{in}}{\dot{E}x_{fuel}}$
Collector	$\dot{E}x_{col}^D = \dot{E}x_{in} - \dot{E}x_{out} + \dot{E}x_{sun}$	$\psi_{boiler} = \frac{\dot{E}x_{out} - \dot{E}x_{in}}{\dot{E}x_{sun}}$

Also to calculate exergy destruction of the collector, exergy of sun’s radiation should be calculated using Eq. (9) (Petela, 2005):

$$\dot{E}x_{sun} = A_a I \left( 1 + \frac{1}{3} \left( \frac{T_{amb}}{T_{sun}} \right)^4 - \frac{4}{3} \left( \frac{T_{amb}}{T_{sun}} \right) \right) \quad (9)$$

where  $A_a$  is the aperture area of the collector and  $I$  is the direct normal irradiance. Also  $T_{sun}$  is the temperature of sun’s surface which is assumed to be 6000 K. Using Eqs. (1) to (9), exergy destruction of each component could be easily computed. Table 1 shows required equation to calculate exergy destruction and efficiency of each component of the cycle.

After calculating exergy destruction of each component, total exergy destruction of the plant can be computed. Using Eq. (10), contribution of each component in total plant’s exergy destruction can be calculated.

$$Y_i = \frac{\dot{E}x_i^D}{\dot{E}x_{total}^D} \quad (10)$$

Also total exergy efficiency of the plant is defined as follows:

$$\psi = \frac{\dot{W}_{net}}{\dot{E}x_{fuel}} \quad (11)$$

#### 4. Economic analysis

Economic analysis is the most important aspect of each engineering project. It is an indicator which shows whether a project is feasible or not. There are many ways to perform this analysis. One of the simplest, yet practical indicators in economic analysis is the simple payback time. The payback time is defined as the length of time required for the cash inflows received from a project to recover the original cash outlays required by the initial investment (Bejan and Tsatsaronis, 1996). Since, in this paper, solar collector and thermal storage system should be added to the cycle, initial investment of the system is defined as the capital cost of the solar collector and thermal storage system. Also the product of the system which is obtained by integration of solar collector and Rankine cycle is the surplus power generated in turbine by not extracting steam from it. Therefore, payback time of the plant is defined as follows:

$$Pb = \frac{Z_{col} + Z_{TS}}{\dot{W}_{surplus} Hr C_{elec}} \quad (12)$$

In this equation,  $Z_{col}$  and  $Z_{TS}$  are capital cost of collector and thermal storage system. Also  $\dot{W}_{surplus}$ ,  $Hr$  and  $C_{elec}$  are surplus output power, total working hour of the plant in a year and cost of electricity, respectively. Value of these parameters can be calculated using data provided in Table 2.

#### 5. Result and discussion

To perform a comparison between the base case power plant and the new proposed one, energy analysis is applied on both

**Table 2**  
Economic data used in the paper (Hu et al., 2010; Palenzuela et al., 2015).

Parameter	Unit	Value
Capital cost of collector	\$/m <sup>2</sup>	150
Capital cost of heat transfer fluid and hydraulic circuit	\$/m <sup>2</sup>	90
Capital cost of thermal storage	\$/kWh	35
Operating hours per year	hr/year	8760
Cost of electricity	\$/MWh	370

cycles using Thermoflex software. This software is specially designed by Thermoflow Inc. for simulation and analysis of all types of power plants. Then the results of this step are used in Matlab software to carry out exergy analysis. Table 3 lists constant parameters used in modeling of solar collector in Thermoflex software.

Table 4 shows thermodynamic properties of each stream of the base case power plant.

After replacing feedwater heaters with solar field, extraction streams are eliminated and the mass flow rate of these streams expands in the turbine. Thermodynamic properties of each stream in this situation are represented in Table 5.

As can be seen in Tables 4 and 5, thermodynamic properties of turbine inlet stream are equal in both cases. The only difference between these two cycles is the heat source for preheating feedwater heater. Heat exchangers in solar implemented power plant are designed in a way that the temperature of outlet feedwater is kept unchanged in comparison with the base case condition. Under these circumstances, total generated power of the plant increases. Table 6 compares the performance evaluation criteria of the two power plants. It is clear that because of eliminating close feedwater heaters more steam is available in the turbine to expand and as a result turbine generated power increases by 2753 kW. On the other hand, consumed power by pumps also increases and another pump is added to the cycle to circulate the heat transfer fluid in the solar field. In general, net output power of the plant increases by 2578.31 kW which shows 8.14% augmentation. Although adding solar field to a Rankine cycle increases its output power, it leads to reduction in exergy efficiency of the system. This is due to huge exergy destruction which occurs in solar collector. In general, solar systems have low efficiencies compared with energy systems which consume fossil fuels. But, since solar energy is free and do not produce any contaminants, it is feasible to use it instead of fossil fuels. As can be seen in Table 6, exergy efficiency of the plant after installing solar field from 30.05% decreases to 26.48%. There is only one disadvantage to this new proposed cycle. Since mass flow rate of turbine outlet stream increased, a higher amount of cooling water is needed to condense the turbine outlet stream. This could damage the environment by consuming its underground water; especially in the place where the power plant is located which suffers from water shortage problems. Another noticeable point is solar collector area which is needed to provide the heat source for preheating feedwater. Under the design condition which was shown in Table 1, 32804 m<sup>2</sup> active aperture area is required.

**Table 3**  
Constant parameters of solar collector.

Parameter	Value
Direct normal irradiance ( $W/m^2$ )	800
$\rho_{\text{mirror}}$	0.94
$\square_{\text{shadowing}}$	0.98
$\square_{\text{geometry}}$	0.93
$\square_{\text{unaccounted}}$	0.96
Collector optical efficiency	0.95
Collector exit temperature (C)	230
Receiver tube outer diameter (mm)	70
Reflector aperture width (m)	1
Reflector row pitch/Aperture width	2.5
Reflector cleanliness factor	0.95
Receiver tube emissivity	0.14
Receiver glass envelope emissivity	0.92
Convective heat transfer coefficient Outside glass envelope ( $w/m^2 \text{ } ^\circ C$ )	56.78
Mass flow rate of thermal oil (kg/s)	45.36

**Table 4**  
Thermodynamic properties of each stream of the base case power plant.

Stream	Working fluid	$P$ [bar]	$T$ [C]	$h$ [kJ/kg]	$s$ [kJ/kg K]	$\dot{m}$ [kg/s]
1	Water	45.2	490	3416.82	7	33.2
2	Water	0.077	40.79	2336.93	7.48	25.93
3	Water	0.4359	40.8	170.83	0.58	25.93
4	Water	1.976	40.73	170.69	0.58	26.01
5	Water	1.976	92.78	388.78	1.22	33.28
6	Water	50.53	93.49	395.49	1.23	33.28
7	Water	50.12	129.01	545.35	1.62	33.28
8	Water	49.7	148.02	626.45	1.82	33.28
9	Water	49.29	178.02	756.41	2.11	33.28
10	Water	11.72	313.44	3076.22	7.10	1.78
11	Water	11.72	153.04	645.69	1.87	1.78
12	Water	5.285	226.88	2911.7	7.15	1.09
13	Water	5.285	134.03	563.68	1.68	2.87
14	Water	3.09	175.43	2814.53	7.19	1.90
15	Water	3.09	98.49	412.89	1.29	4.76
16	Water	0.78	92.81	2606.1	7.28	2.51

**Table 5**  
Thermodynamic properties of each stream of the power plant after implementing the solar field.

Stream	Working fluid	$P$ [bar]	$T$ [C]	$h$ [kJ/kg]	$s$ [kJ/kg K]	$\dot{m}$ [kg/s]
1	Water	45.2	490	3416.82	7	33.2
2	Water	0.077	40.79	2337.32	7.48	30.23
3	Water	0.4359	40.8	170.83	0.58	30.23
4	Water	1.976	40.82	171.08	0.58	30.31
5	Water	1.976	92.78	388.78	1.22	33.28
6	Water	50.19	93.72	396.42	1.23	33.28
7	Water	49.59	129.01	545.32	1.62	33.28
8	Water	49.18	148.02	626.42	1.82	33.28
9	Water	48.71	178.02	756.38	2.11	33.28
10	Water	0.78	92.81	2606.59	7.29	2.975
11	Oil	10.171	117.85	633.07	0.50	45.36
12	Oil	5	230	897.24	1.00	45.36
13	Oil	4.763	191.25	800.92	0.83	45.36
14	Oil	4.549	165.9	740.82	0.72	45.36
15	Oil	4.378	116.65	630.47	0.49	45.36

**Table 6**  
Performance evaluation criteria of the two power plants.

Parameter	Base case power plant	Solar implemented power plant
Turbine generated power (kW)	31 903	34 656
Pump 1 power (kW)	6.22	18.71
Pump 2 power (kW)	241.6	274.8
Pump HTF power (kW)	0	129
Net output power (kW)	31 655.18	34 233.49
Exergy efficiency (%)	30.05	26.48
Cooling water mass flow rate (kg/s)	1330.9	1551.2
Total active aperture area ( $m^2$ )	0	32 804

Table 7 represents exergy destruction of each component. In both cycles, boiler has the highest rate of exergy destruction with approximately 64 MW. In base case condition, turbine and

condenser have the second and the third place in exergy destruction with 4.4 MW and 1.89 MW, respectively and other components have a minor contribution in total exergy destruction,

**Table 7**

Exergy destruction of each component in both cases.

Component	Exergy destruction in base case power plant	Exergy destruction in solar implemented power plant
Boiler	64012.5	63379.71
Turbine	4464.2	4869.124
Condenser	1890.28	2204.4
Pump 1	2.09	13.10648
Open feedwater heater	354.64	410.1479
Pump 2	62.39	90.01992
FWH 1	200.22	816.4889
FWH 2	75.50	429.0385
FWH 3	181.99	659.533
Collector	–	19227.57
HTF pump	–	84.5818
Total	71243.81	92183.72

**Table 8**

Share of exergy destruction of each component in the plant's total exergy destruction.

Component	Share of exergy destruction in base case power plant	Share of exergy destruction in solar implemented power plant
Boiler	89.85	68.754
Turbine	6.266	5.282
Condenser	2.653	2.391
Pump 1	0.003	0.014
Open feedwater heater	0.498	0.445
Pump 2	0.088	0.098
FWH 1	0.281	0.886
FWH 2	0.106	0.465
FWH 3	0.255	0.715
Collector	–	20.858
HTF pump	–	0.092
Total	100	100

therefore any effort to improve performance of the plant should be focused on these components. For example, preheating air and reducing the amount of excess air can be suggested to decrease boiler exergy destruction. Also utilizing a steam turbine with higher isentropic efficiency can decrease exergy destruction in turbine. It is worth mentioning that between feedwater heaters, the open one has the highest rate of exergy destruction. When the solar field is added to the cycle, it increases total exergy destruction of the plant. This is because a huge amount of solar energy is dissipated and lost in the collector, but still its exergy destruction is lower than the boiler. A comparison between the results shown in Table 7 shows a rise in exergy destruction of all components except the boiler. Small reduction in exergy destruction of boiler is due to insignificant changes in feedwater pressure, because of replacing feedwater heaters with heat exchangers. On the other hand, turbine's destruction increases because of increment in its steam mass flow rate. This increase in steam mass flow rate leads to augmentation of condenser heat load and as a result mass flow rate of cooling water increases as indicated in Table 6. This raises exergy destruction of the condenser. Also all four feedwater heaters experience an increment in their exergy destruction. But unlike the previous situation in which open feedwater heater had the highest rate of exergy destruction, in the solar implemented power plant close feedwater heaters have higher rate of exergy destruction. This is because of higher temperature difference between hot and cold streams. All these changes results in almost 30% increment in total exergy destruction of the plant.

To get a better insight, Table 8 provides information about share of each component in plant's total exergy destruction. As can be seen, in base case condition boiler is responsible for about 90% of plant's total exergy destruction and contribution of turbine and condenser is 6.2% and 2.6%, respectively. Share of all the other components equals to 1.2% altogether. Introducing the solar field to the plant changes this contribution completely. In the new proposed system boiler still has the highest share, but its contribution reduces to 68%. After boiler, solar collector has the second highest share with 20.8%. Also turbine and condenser and open feedwater

**Table 9**

Results of economic analysis.

Parameter	Unit	Value
Capital cost of collector and heat transfer fluid	mm\$	7.872960
Capital cost of thermal storage system	mm\$	4.613000
Cost of surplus electricity produced	mm\$	8.3558136
Simple payback time	year	1.49
CO <sub>2</sub> emission avoided	tons/year	12497.4

heater saw a decrease in their contribution, while share of close feedwater heater increases.

As mentioned in Section 4, economic analysis should be considered in each engineering project. Table 9 shows economic results of the solar implemented power plant. Total active aperture area of the collector is equal to 32804 m<sup>2</sup>. This is the required area to produce necessary heat. Also required volume of the storage system is 2576 m<sup>3</sup>. As can be seen, payback time of the project equals to approximately 1.5 year which is very feasible. Another important aspect of the project is that it can increase output power of the plant without consuming more fuel. As a result, a huge amount of carbon dioxide would be avoided as shown in Table 9. This fact can be used to receive subsidy from the government to increase economic feasibility of the plant.

To gain more insight about the effect of adding a solar collector to a Rankine cycle, an analysis is performed on the most important parameters of the collector. Fig. 3 shows the variation of collector area and exergy efficiency of the plant with direct normal irradiation. It is assumed that the outlet temperature of collector is constant. As a result, when direct normal irradiation changes, a different collector area is needed to provide the required temperature. As shown in Fig. 3, when DNI increases, collector area reduces. Because there is more energy exists in sun's radiation and a lower collector area can deliver the same outlet temperature. Also when collector area decreases, exergy destruction in solar field falls too and this leads to higher exergy efficiency of the system.

Another important parameter which has a huge impact on plants performance is outlet temperature of collector. Fig. 4 shows



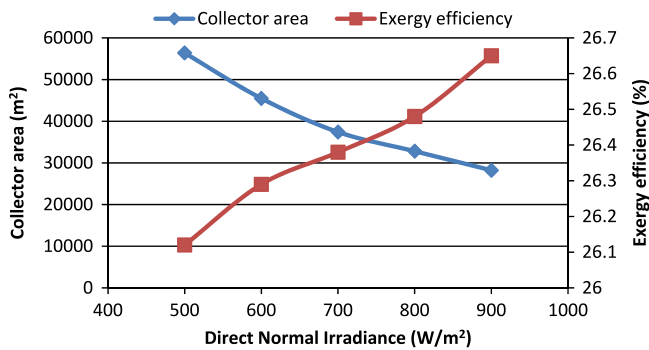


Fig. 3. Effect of direct normal irradiance on the collector area and exergy efficiency of the plant.

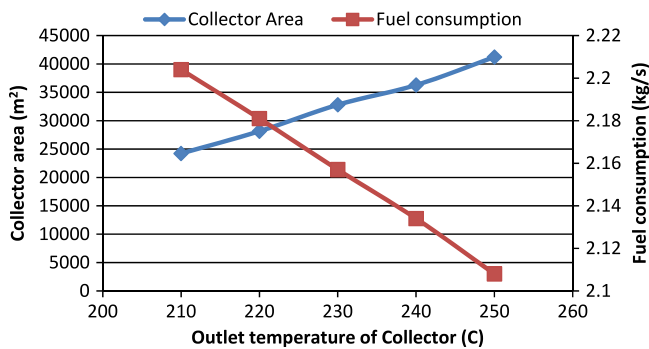


Fig. 4. Variation of collector area and fuel consumption with outlet temperature of collector.

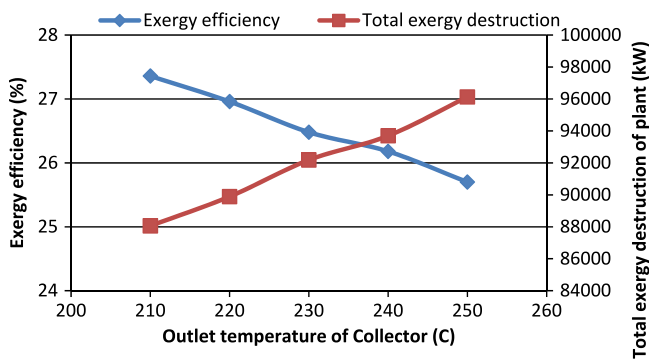


Fig. 5. Variation of total exergy destruction and exergy efficiency with outlet temperature of collector.

the effect of this parameter on collector area and fuel consumption of the plant. As known, higher collector area is needed to provide higher outlet temperature. Also, when collector outlet temperature increases, higher amount of energy is available to preheat the feedwater. Therefore, inlet feedwater temperature to the boiler increases and consequently, lower amount of fuel should be burnt. Higher collector area and lower fuel consumption have opposite effect on total exergy destruction of the plant. While the first increases exergy destruction, the latter reduces it. In general, higher collector area dominates and total exergy destruction of the plant increases. This increment in total exergy destruction leads to reduction in exergy efficiency of the plant as shown in Fig. 5.

## 6. Conclusion

In this paper, closed feedwater heaters of a Rankine cycle are replaced with heat exchangers which use solar energy as heat source.

This combination allows reducing mass flow rate of extractions streams from steam turbine and therefore increases net output power of the system. A complete energy, exergy and economic analysis are performed on the system. Also, a sensitivity analysis is performed on the two most important parameters of the solar field. The main conclusions are summarized as follows:

- Energy analysis showed that net generated power of the system increases by 8.14%. This is due to expansion of steam that is used to be extracted from the turbine to preheat feedwater.
- Exergy analysis showed that highest exergy destruction before and after adding solar field occurs in boiler. The only difference is that after adding solar collector, boilers share of exergy destruction reduces. Therefore the boiler should be modified to reduce its exergy destruction. In general, due to huge amount of losses in collector, adding solar collector reduces exergy efficiency of the system.
- Economic analysis showed that compared to a complete solar energy field, the proposed system in this paper is far economical and its payback time equals to only 1.5 years.
- Sensitivity analysis showed that although increasing outlet temperature of solar collector reduces fuel consumption of the system, it increases aperture area of the collector which leads to reduction in exergy efficiency of the system and increment of payback time.

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