Analysis of Thermal Storage System Design Simulation for Yazd Integrated Solar Combined Cycle (YISCC)

Ramin, Haghighi Khoshkhoo[a], Ali Akbar, Fazli[b]

Abstract**—** During Yazd Solar Power Plant's construction in 1990s, due to the low prices of natural gas, the usage of an integrated solar combined cycle was an economically attractive option in order to provide steady power at nights or intermittent solar radiation hours. As fuel prices have been escalating throughout these past few years, the usage of an independent solar thermal power plant seems both economically and environmentally desirable. In this paper, by generating a MATLAB program, we study the feasibility of implementing a solar thermal storage system on the solar field to bring the combined cycle's operation to a minimum. The important technical design parameters such as: molten salt storage system sizing, variations of mass flow rate in charge/discharge mode and variations of inlet and outlet temperatures in different modes are calculated by this code. This study shows that while applying a solar thermal energy storage system will increase the investment costs, it can also increase the solar plant's output by nearly 70%, while decreasing LCOE by 20%.

Index Terms**—** Solar thermal power plant, Thermal energy storage, Heat transfer fluid, Molten salt.

I. Introduction

Contrary to conventional fossil power plants, CSP (Concentrating Solar Power) plants do not produce CO_2 during operation, and are therefore suitable to meet the challenges of modern life standards without compromising environmental issues. A MW of installed CSP avoids 688 t of CO_2 emissions compared to a combined cycle conventional power plant and 1360 t of CO_2 emissions compared to a conventional coal/steam power plant. [1]

The review of integrated solar combined cycle system (ISCCS) with a parabolic trough technology from 1990 to 2014 was demonstrated by Behar O, et al. The status of operational, under construction and planned power plants of ISCCS, with related the major results of R&D activities and published studies have reviewed and summarized by them. [2]

The ability to provide dispatch ability on demand is what makes CSP stand out among other renewable energy technologies such as PV (Photovoltaic) or wind. Even though the sun is an intermittent source of energy, CSP systems with a storage system offer the advantage of being able to run the plant continuously at a predefined load.Zhang, et al., (2013) review the optimum design and operation of the CSP throughout the year, whilst defining the required thermal energy storage (TES) and/or backup systems (BS), an accurate estimation of the daily solar irradiation is needed. [3]

Intermittency and daytime availability of solar radiation are the main restraints on solar energy. Another limitation for solar parabolic systems is their ability to only absorb direct solar radiations thus the sky needs to be clear because clouds might cause deflection of the solar rays. Thermal energy storage systems can store excess heat collected by the solar field, therefore implementing a thermal storage system can clear these restraints. Thermal storage systems have two main advantages: [4]

A) The ability to provide thermal energy during hours with no direct solar radiation. This means that there would be no reason for the energy to be captured and used at the same time.
B) The solar field's inlet can be isolated from possible disturbances in the outlet. Because the storage system behaves as a good thermal control and stops the disturbance feedback from affecting the solar field's outlet temperature.

As mentioned by Ming L, et al., TES improves the dispatch ability of a CSP plant. Heat can be stored in either sensible, latent or thermochemical storage. Commercial deployment of CSP systems have been achieved in recent years with the two-tank sensible storage system using molten salt as the storage medium. The developments in high temperature TES over the past decade with a focus on sensible and latent heat storage was reviewed by them. [5]

A typical storage system consists of two storage tanks containing storage medium at different temperatures. Storage system keeps the plant running at full-load condition for a longer period. Thermal energy storage allows the plant control system to determine when energy supply through the solar field should stop and storage system must continue the stable operation on its own. Such a system is studied in this paper. [6]

Stuetzle (2004) developed a thermodynamic solar trough model focused on solar field control. Maintaining a constant solar field outlet temperature through HTF mass flow rate's automatic control in the solar field was the purpose of this work. The results showed that automatic control will not improve the plant's gross output significantly as opposed to what was normally achieved by a plant operator. [7]

The Solar Advisor Model, SAM, is modeling software developed by the National Renewable Energy Laboratory. But it isn't good software for modeling a complete thermo physical

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of a real Yazd Integrated Solar Combined Cycle thermal power plant (YISCC). [8]

Angela Patnode (2006) has simulated the SEGS VI by EES, and TRNSYS. In this paper, the equations of Patnode were used for modeling the parabolic trough solar thermal power plant with thermal energy storage system. [9]

In this paper a simulation MATLAB base code is written to calculate the important necessary inputs and operational parameters such as different data about solar field sizing and power block specification such as superheater and reheater. Indeed the initial case with no TES is modeled using design parameters from the Yazd solar power plant. Also the effects of different solar field sizing conditions are analyzed. The YISCC without and with TES are simulated and the effects of different parameters such as variation of solar radiation in one year, variations of mass flow rate in charge/discharge mode, variations of inlet and outlet temperatures and the power output in megawatt in different modes are calculated. This study shows that while applying a solar thermal energy storage system will increase the investment costs, it can also increase the solar plant's output by nearly 70%, while decreasing LCOE by 20%.

II. Oil-Water storage (steam generation with oil)

The study of this paper is focused on the oil-water system, as it is the most commercially acceptable thermal storage system available.

The main operating strategy is that, the thermal storage system should be charged when the flow rate of the heat transfer fluid (HTF) becomes greater of the nominal value needed for steam generation. At this case the extra flow rate flow of HTF charge the molten salt storage media in the heat exchanger. Therefore cold molten salt captures heat from HTF in the heat exchanger and enters the hot tank. [10] It must be noticed that HTF is always flowed in the tube side of the heat exchanger. [1] The ideal HTF flow rate must be maintained throughout the operation, like providing a stable steam generation in the steam train, heat exchangers (Economizer, Evaporator and super heater) and also steam reheat process at the high pressure steam turbine outlet.

The desired molten salt volume is considered to be the amount of molten salt required to fill one tank. Obviously any of cold and hot tanks have a minimum volume of salt every time [11]. Design temperature is calculated by assuming a maximum temperature of 386˚C for the hot salt tank and 293˚C for the cold salt tank. The tanks must be thermally homogeneous.

To charge the HTF by hot molten salt (discharge mode of the storage system), the insufficient HTF flow of the solar field is mixed with the complementary HTF flow rate is heated in the hot tank molten salt. Considering the outlet temperature being fixed on 393˚C, and 4 MWe being the minimum amount of power assumed necessary for electricity generation, minimum oil flow rate would be 54 kg/s. if even after storage discharge, HTF cannot reach this flow rate (54 kg/s), molten salt would remain in the hot tank and no power would be generated. [12]

III. Solar Field And Storage System Design Strategies

The power plant and its related thermodynamic cycle consist of solar field, storage system, heat exchanger and power block that are shown in Fig. 1. It must be noted that this solar field consists of parabolic trough collectors tracking the sun in one direction from east to west while rotating on a north-south axis. The inputs required for the solar field and storage system model are:

- The flow rate of heat transfer fluid (HTF) [m3/s];
- Inlet HTF temperature [°C];
- DNI of the solar field [W/m2];
- Site air condition temperature [°C];
- Wind speed [m/s].

The basic operating strategies for an Oil-Water storage system are shown in Fig. 2. Advanced storage control systems can be very important for the analysis of parasitic calculations. For example, the minimum discharge volume necessary to generate nominal power could be determined.

Solar field's outlet HTF mass flow calculation process can be broken down into three main steps. First, the absorbed energy ($\dot{Q}_{absorbed}$) is calculated on the basis of equations 1 to 5. Next, by using the equations 6 to 9, the receiver heat loss ($\dot{Q}_{heatloss}$) is calculated. Finally the effective energy gain of the HTF,$\dot{Q}_{collected}$, is calculated on the basis of equations 10 to 13. Knowing the useful energy absorbed by the HTF and the inlet fluid enthalpy makes it possible to determine the outlet HTF enthalpy. Since the outlet temperature is fixed on 393℃, the mass flow rate of outlet HTF can be calculated from the equation 12 (energy balance).

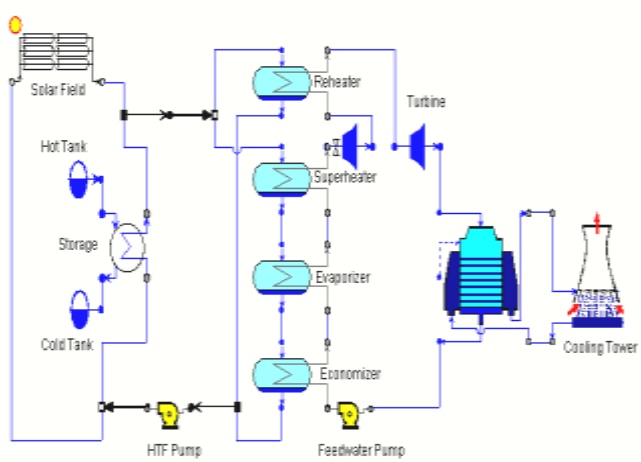

Fig. 1. Schematic of the CSP plant with storage

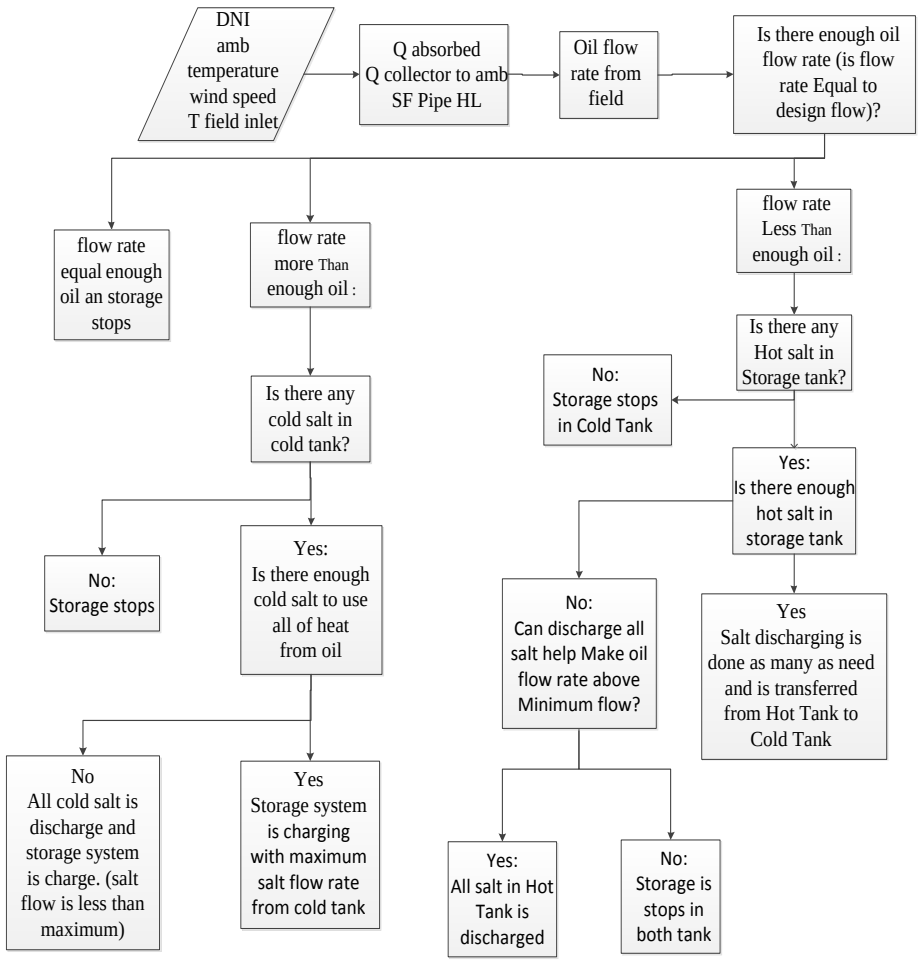

Fig. 2. Storage Controls for Oil-Water Storage model

Calculation process is described further in the following sections. Absorbed solar radiation equation is: [9].

$$\dot{Q}_{absorbed} = DNI \times \cos(\emptyset) \times K(\emptyset) \times Rowshadow \times Endloss \times \eta_{opt,0^\circ} \times F_e \quad (1)$$

The quantity of direct normal insolation (DNI), incidence angle ($\emptyset$), and soiling factor (Fe) are derived from the Yazd solar power plant documentations.

In equations (2) to (5) incidence angle modifier ($K(\emptyset)$), performance factor that accounts for mutual shading of parallel collector rows (Row Shadows), performance factor that accounts for losses from the edges of HCEs (End Loss), and peak optical efficiency (optical efficiency with an incidence angle of 0° ($\eta_{opt,0^\circ}$)) are calculated. [1]

$$K(\phi) = 1 - 2.23073 \times 10^{-4} \times \phi - 1.1 \times 10^{-4} \times \phi^2 + 3.18596 \times 10^{-6} \times \phi^3 - 4.85509 \times 10^{-8} \times \phi^4 \quad (2)$$

$$Row\ Shadow = \frac{W_{eff}}{W} = \frac{L_{spacing}}{W} \cdot \frac{\cos(\emptyset_z)}{\cos(\emptyset)} \quad (3)$$

$$End\ Loss = 1 - \frac{f \cdot \tan(\phi)}{L_{SCA}} \quad (4)$$

$$\eta_{opt,0^\circ} = \rho \times \gamma \times \tau \times \alpha|_{\Phi=0^\circ} \quad (5)$$

The quantities of ϕ and $\emptyset_z$ in the above equation as well as the other parameters are published by Fichtner for Yazd solar power plant and are given in Table 1. [12]

TABLE1: CHARACTERISTIC OF SOLAR FIELD [12]

L_{SCA}	f	W	$L_{spacing}$	α	τ	γ	ρ
148.5	1.71	5.76	17.3	0.95	0.93	0.95	0.93

IV. HEAT LOSS FROM COLLECTORS AND PIPING

The parabolic trough collector total heat loss, $Q_{\text{collector}\to\text{ambient}}$ is composed two important parts: radiation heat loss part, $Q_{\text{absorber}\to\text{ambient}}$, and convective and conductive heat losses part, $Q_{\text{absorber}\to\text{glass}}$. It is possible to calculate the amount of thermal loss from all three mechanisms by the heat loss coefficient, $U_{L)abs}$ as [13]:

$$Q_{,\text{collector}\to\text{ambient}} = U_{L)abs} \times \pi \times d_0 \times L \times (T_{abs} - T_{amb}) \quad (6)$$

Where Tabs is the mean absorber tube temperature, Tamb is the ambient air temperature, d0 is the outer diameter of the absorber tube, and L is the absorber tube length. The heat loss coefficient is given in (W/m2.K) units per square meter of the steel absorber tube surface is calculated based on equation (7). Different heat loss coefficients in contrast with the receiver pipe temperature can usually be expressed with a second-order polynomial equation (Equation (8)), with coefficients a, b, and c derived experimentally [1]:

$$U_{L)abs} = a + b \times (T_{abs} - T_{amb}) + c \times (T_{abs} - T_{amb})^2 \quad (7)$$

The value of a, b and c coefficients for a SKAL-ET150 collector is given in Table 2.

Thermal losses from the inlet and outlet HTF piping in the solar field are calculated by the following experimental equation:

$$SfPipeHl = 0.01693 \times \Delta T - 0.0001683 \times \Delta T^2 + 6.78 \times 10^{-7} \times \Delta T^3 \quad (8)$$

SfPipeHl is the solar field pipe heat [W/m2], and ΔT [°C] is described by equation (9).

$$\Delta T = \frac{T_{fieldoutlet} + T_{fieldinlet}}{2} - T_{ambient} \quad (9)$$

V. HTF Energy Absorption and the Outlet Mass Flow of the HTF

The net energy absorbed ($\dot{Q}_{collected}$) with HTF throughout in the solar field equals the difference between the total of energy gained by the absorber tubes ($\dot{Q}_{absobed}$, Equation 1) and the total of heat loss from the receivers ($Q_{,collector \to ambient}$, Equation 6) and solar field piping (SfPipeHl, Equation 8):

$$\dot{Q}_{collected} = \dot{Q}_{absorbed} - (SfPipeHl + (Q_{,collector \to ambient})/\pi d_0 l) \quad (10)$$

Table 2: Values of Coefficients A, B, and C for a SKAL-ET150 Collector [1]

T_{abs} [°C]	A	B	C
<200	0.687257	0.001941	0.000026
200<<300	1.433242	- 0.00566	0.000046
>300	2.895474	- 0.0164	0.000065

The outlet temperature of the solar field, T_{out}, is fixed at 393˚C, while the inlet temperature, T_{in}, will vary based on the last iteration while the mass flow rate is calculated by Equation 11:

$$\dot{m} = \frac{\dot{Q}_{collected}}{c_p \times (T_{out} - T_{in})} \quad (11)$$

VI. Superheater and Reheater

A single model is used for both superheater and reheater in the solar field for increasing the temperature of the inlet steam. Fig. 3 shows the schematic for this equipment.

It is essential to calculate the heat transfer rate inside the heat exchangers which is performed by an energy balance equation and the NTU method. The energy balance across the heat exchanger can be stated by:

$$\dot{m}_1 \times cp_1 \times \Delta T_1 = \dot{m}_2 \times cp_2 \times \Delta T_2 \quad (12)$$

The overall heat transfer coefficient, UA, can be determined from the design conditions prepared in the power plant technical specifications by Fichner. At full-load conditions the UA for the superheater and reheater is 225 [kW/K] and 354 [kW/K], respectively. At reference state, the heat exchanger's UA is provided as a parameter for the exchanger model. It must be noticed that at partial loads, decreasing the streams' flow rate would result in reduction of the UA. The relationship between the reference UA/flow rate and a reduced UA/flow rate is presented in the following equation. [9]

$$\frac{UA}{UA_{ref}} = \left(\frac{\dot{m}_o}{\dot{m}_{o,ref}}\right)^{0.8} \quad (13)$$

The heat exchanger design point inlet and outlet temperature, enthalpy and mass flow rate for HTF and working fluid are given in Table 3.

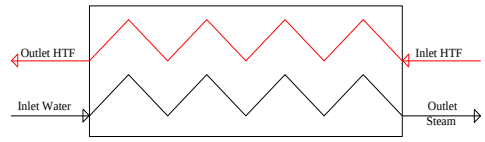

Fig. 3. Flow diagram of superheater/reheater

Table 3: The quantity of heat exchanger at reference state [12]

	Working fluid	HTF
Inlet temp (°C)	205.18	393
outlet temp (°C)	381	295
Inlet enthalpy (kJ/kg)	864.3	778.18
Mass flow (kg/s)	2705.1	544.91
outlet enthalpy (kJ/kg)	26.52	218

VII. Storage Tanks and Storage Controls Logic

The storage fluid's volume in each tank is enough for a thorough charge or discharge process.

There are two main limitations facing this system, one is that the tank cannot be filled completely; also the tank can never be fully discharged [8]. Both tanks have a diameter of 25 m and a maximum charging level of 7.5 m. Power losses were reported 107 kW and 99 kW for the cold tank and hot tank respectively. Given an area of 1571 m^2, the heat loss terms can be expressed as 63 W/m^2 and 67.7 W/m^2. The thermal energy storage tanks specifications based on the model of simulation of YISCC with TES are displayed in Table 4. The mass flow quantity is the value calculated repeatedly every second.

The energy balance for the tank mass is [9]:

$$M_{tank} = M'_{tank} + \dot{m}_{in} dt - \dot{m}_{out} dt \quad (14)$$

Where M'_{tank} the amount of mass left is in the tank from previous iterations, Q_{in} is the amount of heat entering the tank, and Q_{out} is the heat exiting the tank which can be calculated by:

$$Q = \dot{m} \times cp \times T \quad (15)$$

T in this equation is in Kelvin. The heat inside the tank can be calculated by:

$$Q_{tank} = Q_{in} - Q_{out} - Q_{loss} \quad (16)$$

Table 4: The thermal energy storage tanks specifications [12]

Mass flow (m^3/hr)	Discharge Time (hr)	Volume (m^3)	Diameter (m)	Height (m)	Area (m^2)	Mass (Ton)
728.2	5	3641	25	7.5	1571	6715

VIII. Simulation Results

Three typical days are chosen from the annual simulation, to present the behavior of a power plant, with and without storage, at different weather and climate conditions. The first analysis is for a summer day. The first day chosen for analysis is July 7th which has the most amount of solar radiation reaches the earth. August 6th is chosen to represent a summer day cloudy with low solar radiation along with high electricity demands which is basically due to air conditioning systems. The last day chosen for analysis is a day on December first. This day is chosen to represent an autumn day with ideal conditions.

Fig. 4 present the power output in the sunny day. A deploy angle of 10 degrees is considered for the solar field along with a stow angle of 170 degrees, which results in power generation delay in the morning and evening.

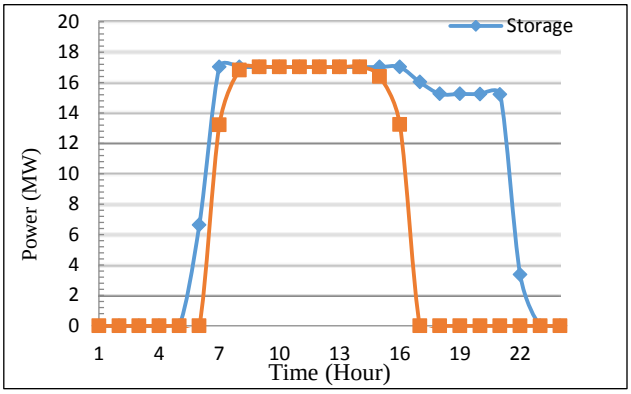

Fig.4. Total hourly energy generation at July 7th, (sunny day)

In Fig. 5, when clouds arise at noon, an accurate display of the weather is illustrated (In the early afternoon, power generation is significantly reduced as a result of lower DNIs). Increasing the solar multiple would not be beneficial throughout the entire year. Even at a solar multiple of one, some part of the thermal energy is going to be wasted during the summer. However, the relationship between costs and revenues of implementing a thermal storage system must be considered. These revenues are due to saving the amount of potential energy which is lost in a conventional solar power plant. However, the potential amount of wasted power for a solar multiple of one cannot justify thermal storage. As for this study, the solar multiple is considered to be 1.8 for a power plant with thermal energy storage.

According to Fig. s 4 and 5, there are clear differences between power generated by a solar power plant with or without storage. The amount of power generated for both plant cases of with/without storage are given in Table 5.

TABLE 5: THE AMOUNT OF POWER GENERATED

	Storage	No storage
July 7th	257 MW-h	161 MW-h
August 6th	221 MW-h	126 MW-h

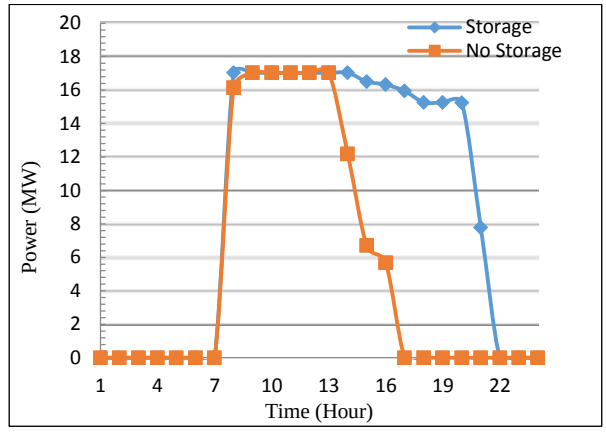

Fig.5. Total hourly energy generation at August 6th, (cloudy day)

The amount of monthly power generation for both cases is shown in Fig. 6. A considerable amount of energy can be generated when thermal energy storage system is implemented on the plant (about 49Gwh) while without storage it would be about 29Gwh which is evident in Fig. 6.

IX. Economic Analysis of Solar Thermal Power Plant with Thermal Storage System

Investment cost and electricity generation increase when solar field size is greater. But cost of energy not only depend on investment cost and annual energy but also depend to interest rate, O&M cost and life of project. Economical size of solar thermal power plant with thermal storage system means the solar and power block field with minimum cost of energy, as secure the guaranteed value of thermal power. Levelized cost of energy (LCOE) is an excellent parameter for realizing optimum size of field, is given by Equation 17 [13].

$$LCOE = \frac{\sum_{t=1}^{n} I_t + M_t + F_t}{\sum_{t=1}^{n} \frac{E_t}{(1+r)^t}} \tag{17}$$

Where I_t is the annual plant investment; M_t, the annual operation and maintenance costs; F_t, the annual fuel costs; E_t, the annual net electricity generated; r, the discount rate and n, plant lifetime.

The economic analysis is performed for a thirty year lifetime of plant. It is considered in this paper that the electricity generated in both plants will be offered to the national grid at the same price. The prices and costs applicable for this economic analysis are shown in Table 6. These prices are based on the Sandia International Laboratory documents as a reference [8]. Since the transportation and installation costs are not accessible for Iran territory, and just a comparison between two identical plants with and without the storage system is intended in this paper, these costs are not mentioned in this analysis and the plant is considered to be installed in Europe.

Fig. 7 shows the amount of LCOE for the plants with and without the storage system, which is calculated based on equation (14) and the prices indicated on Table 6.

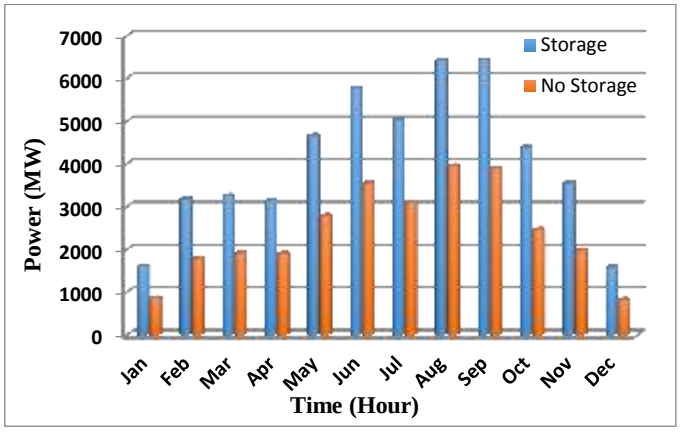

Fig.6. Net power for different month

X. Conclusion

Comparing the results from the specified plant with and without the storage system shows that installing a storage system not only allows the plant to perform longer hours at nameplate capacity, but it also acts as a buffer system at transition states like eliminating effects of a cloudy sky on plant output. According to Fig. 5 implementing a storage system increases the plant energy output to almost 1.7 times the previous output. It also appears that the levelized cost of electricity decreases more than 20%, which makes implementing the storage system a desirable option.

Table 6: Estimated Current Costs For Parabolic Trough Systems

Investment		
Solar field	295 $/m^2	30868800
HTF System	90 $/m^2	9417600
Preheater	2.464 $/kWe	41888
Evaporator	16.72 $/kWe	284240
Superheater	2.6 $/kWe	44200
Reheater	6.754 $/kWe	114811.2
Power Block	940 $/kWe-gross	15980000
Installed Cost (no storage)	5.3 $/W	90100000
O&M (no storage)	66 $/kWe-year	1122000
Thermal Storage cost	80 $/kWh	21040000
Installed Cost (storage)	7.2 $/W	122400000
O&M (storage)	70 $/kWe-year	1190000
Insurance	1%	901000

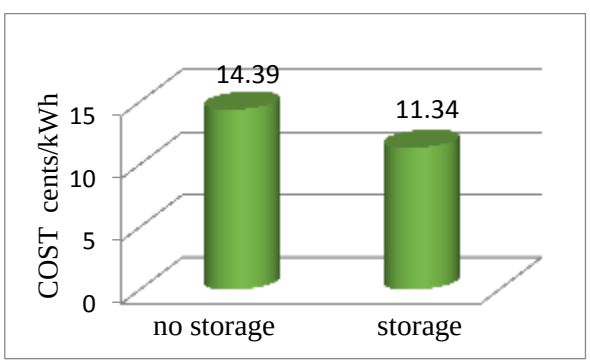

Fig.7. LCOE comparison for plant with/without storage

Nomenclature

$\emptyset$: Incidence angle [degree].
$\emptyset_Z$: Zenith angle [degree].
W_{eff}: Mirror aperture's effective width [m].
$L_{spacing}$: spacing length between the troughs [m].
W: Collector aperture width [m].
f: Focal length of the collectors [m].
L_{SCA}: Length of a single solar collector assembly [m].
P: Reflectivity of the collector reflecting surface [%].
γ: Intercept factor [%].
τ: Transmissivity of the glass tube [%].
α: Absorptivity of the absorber selective coating [%].

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