Desalination and Water Treatment
www.deswater.com

doi: 10.1080/19443994.2013.797674

52 (2014) 9–26
January

Taylor & Francis
Taylor & Francis Group

Solar cogeneration power-desalting plant with assisted fuel

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Received 26 February 2013; Accepted 11 April 2013

ABSTRACT

Solar power plants (SPP) using parabolic trough solar collectors operating Rankine steam cycle are well proven and most widely used worldwide. These plants have low-power cycle efficiency (30%) due to low throttling conditions of 350–375°C and 100 bar. Conventional steam power plants have high throttling conditions (535–560°C and 140–160 bar) and high efficiency (38–40%). Qatar is endowed with abundance of natural gas (NG) resources and production. Qatar's power plants use natural gas-operated gas turbines. However, resources are finite and domestic consumption is rising because of rapid economic and population growth. As in many countries, SPPs are considered to take share in electric power and desalted seawater productions. This elongates the life of the NG resources, and keeps the income return from its exporting. This also limits the emission of greenhouse gases and air polluting gases due to NG combustion, which badly affects the environment. While NG is used in SPP to compensate solar energy intermittent nature and keeps operation during non-sunshine hours, the main purpose of using NG here is to raise the SPP throttling temperature, and thus increasing steam cycle efficiency, even during full sun shine. It also lowers using expensive land where the SPP is planned. This paper studies the feasibility of utilizing NG to superheat the steam leaving the SPP solar collector field, and to heat the feedwater to the collector. This drastically increases both power output and efficiency. Modifications of SPP power cycle to become cogeneration power desalting plants are presented.

Keywords: Solar power plant SPP; Natural gas; Steam turbine; Heat recovery super heater-feedwater heater; Efficiency; Parabolic solar collectors

1. Introduction

Fossil fuels resources, natural gas (NG) and crude oil, in the Gulf cooperation countries (GCC) are plentiful. However, these resources are finite and their consumptions are excessively on the rise. All produced NG are consumed locally in all GCC, except in Qatar. The cost of crude oil and its refineries became too expensive to be burned in power plants (PP). So, GCC have to diversify the fuel used in PP, which is about half of the consumed fuel for all purposes in a country like Kuwait.

Solar power plants (SPPs) are expected to carry a share in satisfying future electric power (EP) needs in many countries. Even in fuel exporting countries like the GCC, there is great interest to apply solar energy to generate EP and desalted seawater (DW) in what is called solar cogeneration power desalting plants

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(CPDP). This elongates the life of the available oil and NG resources, or at least keep the return income from exporting these fuels. It also limits the emissions of greenhouse gases and air polluting gases due burning the fossil fuel, which negatively affect the environment.

Besides, all utilities common goal is to increase the efficiency of converting thermal energy obtained either from fossil fuel or solar energy to EP. In Qatar, the use of SPP is seriously considered to satisfy its increasing demands of both EP and DW. In this paper, an overview of the current practice of SPP is given. Modifications to transfer SPP to solar CPDP and use of NG-assisted fuel to raise the efficiency and reduce the expensive solar collectors' area are outlined. An economic analysis is presented for SPP and NG fuel-assisted solar plants.

1.1. The SPP using parabolic trough collectors (PTC) and Rankine steam cycles

Thermal SPPs use several technologies to concentrate direct normal solar energy to have high temperature heat source, which can operate several heat engine power cycles. These include: parabolic troughs and linear Fresnel solar collectors concentrating solar rays on line collectors; and central receivers (towers) and parabolic dishes concentrating solar rays on point collectors. The SPP using solar PTC and steam Rankine heat engine is the well proven and mostly used type. In this SPP, troughs-curved mirrors, Figs. 1a and 1b, reflect direct solar radiation onto collector tubes (called a receiver, absorber or collector) containing running fluid along the length of the trough and positioned at the focal point of the reflectors, Fig. 2. The trough is parabolic along one axis and linear in the orthogonal axis. To keep the sun daily position perpendicular to the receiver most of the time, the trough tilts east to west to keep the direct radiation focused on the receiver. So, the trough design for practical reasons does not use tracking on a second axis. The receiver may be enclosed in a glass vacuum chamber as given in Fig. 2. The vacuum reduces convective heat loss significantly. The fluid passing through the receiver is called heat transfer fluid (HTF). Its temperature is raised to high temperatures (390°C). Common fluids are synthetic oil, molten salt, and pressurized water to be converted to steam. The hot HTF is used in heat exchangers (HX) to generate steam to operate Rankine steam power cycle.

Full-scale PTC systems consist of many such troughs laid out in parallel over large land area (3–4 times the solar collectors' area). The SPPs using PTCs,

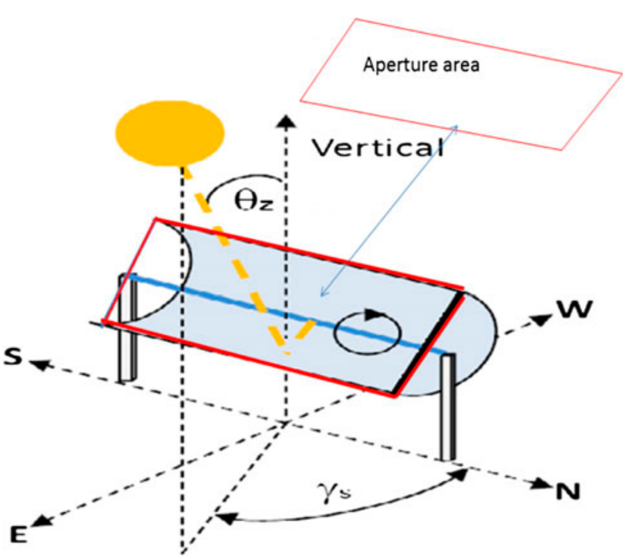


Fig. 1a. Schematic diagram of parabolic solar collector showing sun angles and aperture area [1].

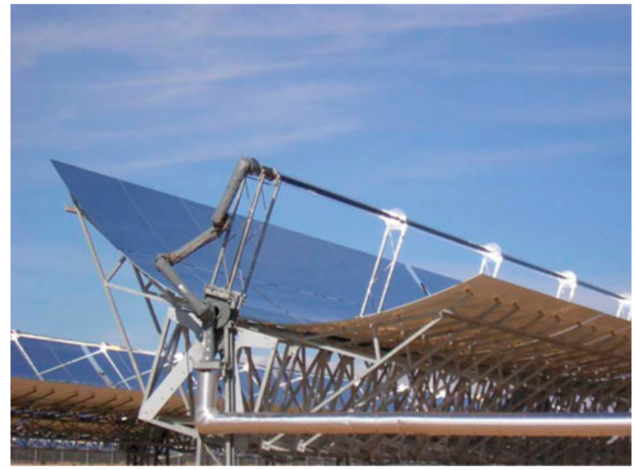


Fig. 1b. End of a row of parabolic trough SCAs of a SEGS VI SPP Plant [1].

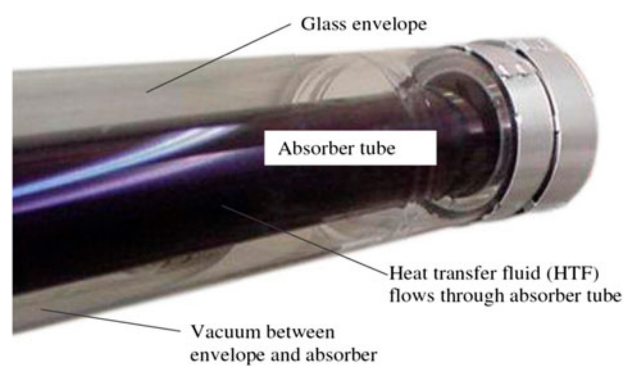


Fig. 2. HCE (Photo source: Solel UVAC, 2004) [1].

known as solar electric generation system (SEGS), have been in full operation in California, USA since 1985. The gained long time operation and experience by the SEGS make the parabolic trough SPP well proven and the most used type. Parabolic trough technology accounted for about 96% of global concentrated solar power (CSP) capacity at the end of 2010; tower technology accounted for 3%, [2]. The SEGS consists of nine plants with total capacity of 354 MW. It is currently the largest operational solar system (both thermal and non-thermal). Newer plants are the 64 MW Nevada Solar One plant and 100 MW Shams I plant under construction in the United Arab Emirates (UAE). Data on some of the SEGS and Nevada solar 1 are given in Table 1, and on some of Spain parabolic trough SPP are given in Table 2, [3,4]. At the end of 2010, Spain accounted for about 57% of all global CSP capacity. The Andasol 1, Andasol 2, and Andasol 3 plants shown in the table are the first commercial CSP plants to feature thermal energy storage (TES), using two-tank molten salt system to store up to 7.5 h of peak-load energy, [5]. Two other large projects in USA are the Mojave Solar Park and Beacon solar project. The Mojave Solar Park is 553 MW parabolic trough power plant system situated in the Mojave Desert in California and is expected to be completed in 2014. It will cover up to 24 km^2 of land and use 1.2 million mirrors and 317 miles long of vacuum tubing. The Beacon Solar Project is 250 MW solar power plant in the Mojave Desert also. It is under reviewing. It would have parabolic trough solar collectors, and costing approximately 1 billion Dollars.

These SEGS data indicate that the specific field collector size per MW is in the range of 6,267–7,677 m^2/MW without TES; and the maximum solar field temperature is 390°C.

Many SPPs using PTC have been built in Spain. Most of these plants have capacity of 50 MW each. Some of these plants have 7.5 h of TES, and specific field collector size of per MW in the range of 10,000–11,000 m^2/MW. When no solar TES is used, the solar collector's areas per MW is in the range of 6,000 m^2/MW as shown from SEGS VI (6,267 m^2/MW) and the recently built Ibersol Ciudad Real in Spain (5,755 m^2/MW). The reported direct normal irradiance (DNI) in Spain is around 2,168 kWh/m^2/y (close to that in

Table 1
Basic characteristics of some SEGS parabolic trough SPP at Kramer Junction and Nevada Solar 1 [1]

Plant name	Start up year	Capacity (MW)	Solar field temperature (°C)	Solar turbine efficiency (%)	Solar field size m^2	Power cycle Pressure	Dispatch-ability
SEGS III	1987	30	349	30.6	230,300	40 bar	Gas boiler
SEGS VI	1988	30	390	37.50	188,000	100 bar, reheat	Gas boiler
SEGs IX	1991	80	390	37.60	483,960	100 bar, reheat	HTF heater
Nevada solar 1	2007	64	390	37.60	357,200	100 bar, reheat	None

Table 2
Basic characteristics of the parabolic trough SPP in Spain [3,4]

Name	Capacity MW	Storage hours	DNI kWh/m^2/y	Solar field area As/MW	Plant, 1,000 m^2/MW	Temperature Solar field in/out	efficiency
Alvarado1	50		2,174		27	?/393	
Andasol 1	50	7.5	2,136	10,202.4	40	293/393	16
Andasol 2	50	7.5	2,136	10,202.4	40	293/393	16
La Florida	50	7.5		11,055	40	298/393	14
Extresol-2	50	7.5	2,168	10,202.4	40	293/393	16
Extresol-1	49.9	7.5	2,168	10,222.85	40	293/393	16
Ibersol Ciudad Real	50	0	2061	5,755.2	30	304/391	
La Dehesa	49.9	7.5		11,077.15	40	29/393	14
Majadas	50		2,142				
Manchasol-1	49.9		2,208		40	293/393	16
Palma del Rio 2	50	0	2,291		27	?/393	

Qatar), while in California, it is reported around 2,700 kWh/m^2/y. The solar field has inlet and outlet temperature of 293 and 393°C, respectively, and thus the throttling condition of Rankine power cycle is about 370°C temperature. The steam pressure at the steam turbine inlet is 100 bar. These necessitate the use of reheat turbine to insure at least 88% dryness fraction at the turbine exit as required by the steam turbines industry. The cycle has an average annual solar to electricity efficiency of 16%.

2. SPP hybridization and adding fuel-assisted super heaters

Typical SPP is shown in Fig. 3. For the SPP to operate when the sun is not shining, backup fuel, typically NG is used, and/or TES. This is known as hybridization, which means that the SPP can be operated by using some backup fuel. All existing trough plants are hybrid plants. As shown in Fig. 3, backup NG fired boiler is used to generate steam to run the turbine, even when the solar collectors are not operating. Another auxiliary NG-fired heater is used to heat the HTF. In these systems, the maximum temperature of the HTF is about 390°C, and accordingly the maximum steam temperature is set equal to 370°C. Sensible, cost-effective operation of a hybridized solar plant dictates that NG will be used periodically only to supplement electrical production. The fossil energy would likely be used only for economic dispatch during on-peak or mid-peak periods. The system heat rate is high, since the NG is being used in a conventional steam power plant instead of a combined cycle. In Spain, the SPPs are limited by the Federal Energy Regulatory Commission "qualifying facility" status to cap NG use to 25% of energy input to the plant.

It is noticed that in Fig. 3 the steam super heater and reheater are mainly heated with HTF which has a maximum temperature of 390°C, which limits the maximum steam temperature to 370°C. This is far below the 535°C maximum steam temperature used by conventional steam power plant.

This maximum temperature of 535°C can be achieved by introducing NG heater just before the inlet of the high pressure (HP) turbine, and another NG heater between the HP and low pressure (LP) turbines as shown in Fig. 4. The effect of introducing these two super-heaters on the cycle will be clearly shown when analyzing the two cycles, when with and one without fuel assisted super heaters.

3. Reference SPP using PTC

A reference plant is chosen here similar to the SEGS VI of 30 MW located in California, Fig. 5. More data on the reference plant are given in Refs. [1,3]. As shown in Fig. 5, the plant has five closed feedwater heaters (FWH), and one open FWH (de-aerator). Three of the closed FWH as well as an open FWH are supplied with steam bled from the LP turbine, and the other two heaters are supplied with steam bled from the HP turbine.

The solar field, Fig. 6 consists of several rows of single-axis tracking collector troughs. SPPs using PTC

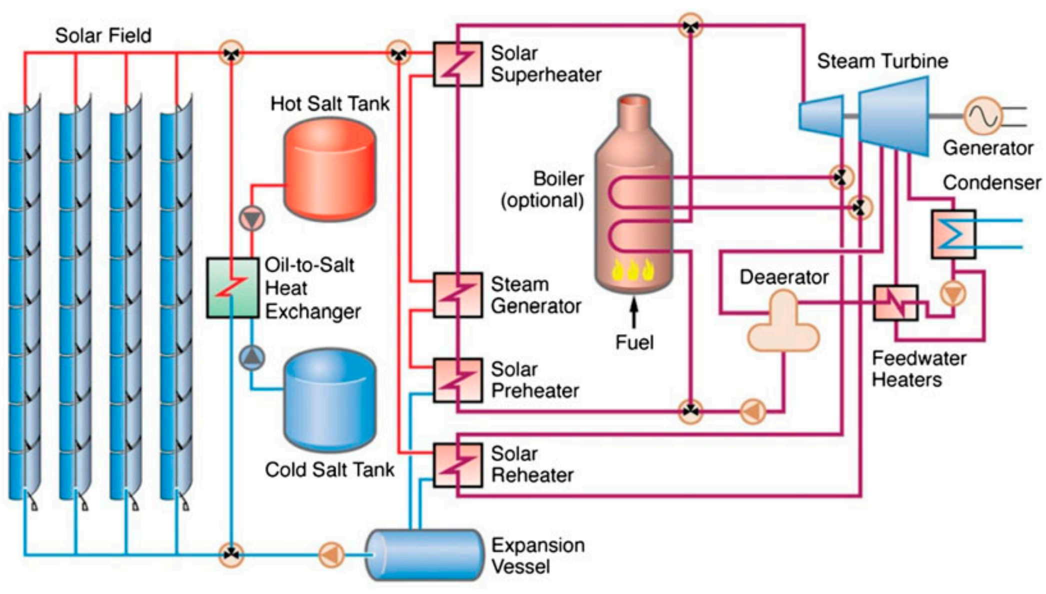


Fig. 3. Example of hybrid plant, where NG is used to support summer on-peak generation, sunshot vision study, US department of energy February 2012.

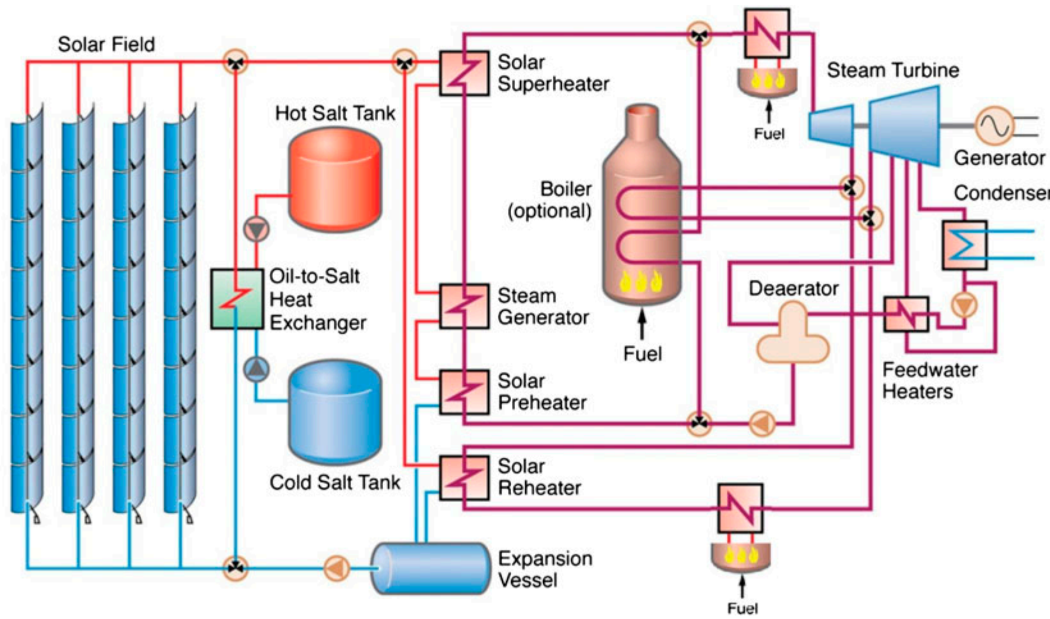
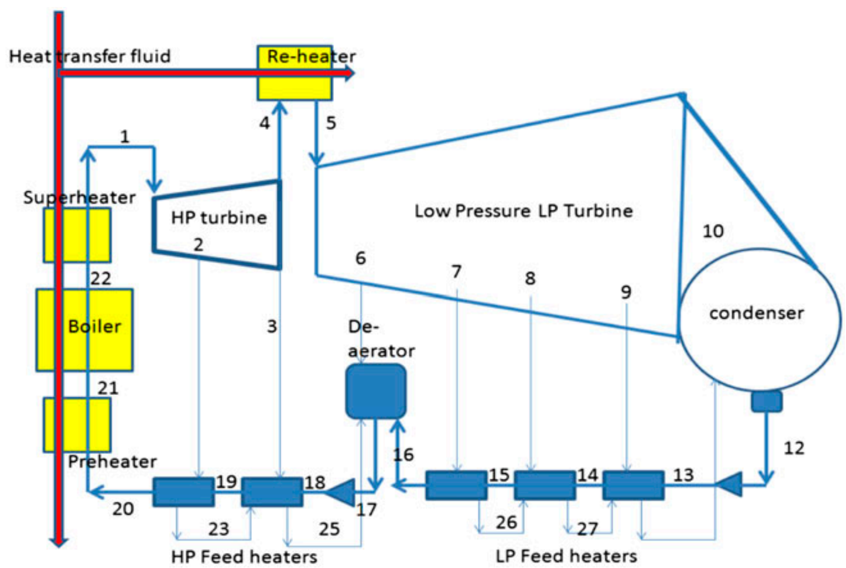


Fig. 4. Hybrid solar plant, where NG is used to support summer on-peak generation plus raising the steam temperature to the HP turbine to 535°C and LP turbines.



m_1=36, m_2=2.7, m_3=2.575, m_4=m_5=30.715,$m_{5\text{-}6}$=30.715, m_6=2.0245, m_7=1.634, m_8=1.498, m_9=1.021, m_{10}=24.5375, m_{13}=m_{14}=m_{15}=m_{16}=30.725, m_{17}=m_{18}=m_{19}=m_{20}=m_{21}=m_{22}=36

Fig. 5. The reference solar only steam power cycle showing the components of the cycle with state points [6].

have a proven track record for providing firm renewable daytime peaking generation. Trough plants generate their peak output during sunny periods when air conditioning loads are at their peak. Integrated NG hybridization and thermal storage have allowed the plants to provide firm power even during non-solar and cloudy periods.

Each trough is formed of float-formed, parabolic-curved mirrors that focus direct radiation from the sun onto a heat collection element (HCE) running through the focal line of each trough. The concentration ratio of the troughs is 71:1 for the used collector known as LS-2 collector model. The collectors are aligned on a north–south line, thus tracking the sun

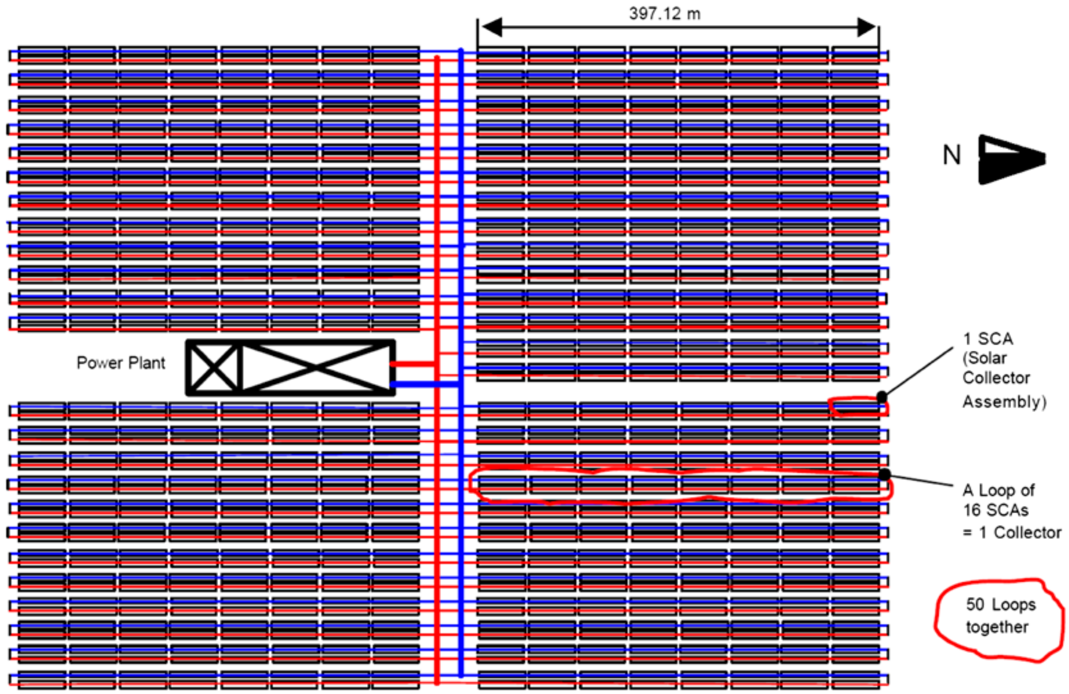


Fig. 6. Solar field of the reference steam power cycle [6].

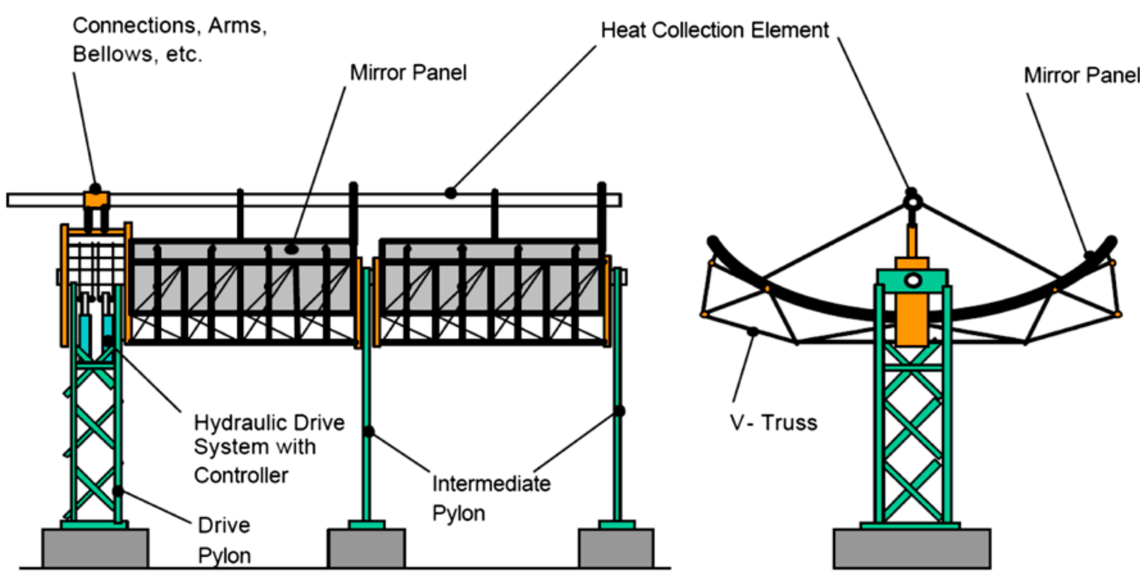


Fig. 7. Solar collector assembly [6].

as it traverses the sky from east to west. The reflectors are made up of a number of submodules each with a typical length of 12 m. The type 100 has an overall length of 100 m and 8 submodules. Larger parabolic trough reflector has a length of 150 m and an aperture width of 5.77 m (see Fig. 1b) and consists of 12 submodules [2]. There are 50 solar trough collector groups, with each group formed by a loop of 16 solar collector assemblies (SCA), Fig. 7. The parabolic troughs are fixed on central pylons that must be very sturdy and heavy in order to cope with the resulting central forces.

The entire SCA consists of six mirror panels. All the SCAs are controlled by a main process computer. The length of an entire collector mirror is the length of one mirror panel times the number of mirror panels in a single collector. The collector mirror length is 753.6 m, Fig. 4.

The HCE is a steel absorber tube 70 mm diameter and coated with either black chrome or a selective ceramic/metal surface coating (Fig. 2). The absorber tube is surrounded by a glass envelope; the space between the steel tube and glass is evacuated to limit heat losses from the absorber tube to the surrounding, and to save tube coating. The focused radiant energy from the sun is absorbed through the HCE and transferred to the HTF, which is synthetic oil such as a mixture of biphenyl and diphenyl oxide (Therminol VP-1) that is pumped through each HCE tube. The heated HTF is pumped after being heated to HX, where its thermal energy is transferred to water, Fig. 6. In the HX, water coming back from the steam cycle is preheated to its saturation temperature (in part of HX called preheater or economizer), transferred to vapor (in part of HX called boiler or steam generator), and then super heated (in part of HX called super heater), Fig. 6. Also the steam leaving HP turbine is reheated (in part of the HX called reheater) before returning back to LP turbine inlet. The thermal energy gained by water is the heat source for Rankine steam power cycle.

The steam leaving the steam turbine is condensed in a water cooled condenser, and is pumped back through the cycle FWHs to the cycle's steam generator. Heat absorbed by the condenser water is rejected to environment through an induced draft cooling tower. In the plant, there is an ancillary NG-fired boiler, which may be used to supplement solar steam production (up to 25%).

Heat exchanger	T_{in} (HTF)	T_{out} (HTF)	T_{in} (steam)	T_{out} (steam)
Preheater	317.8	298	234.83	
Steam generation	377.2	318		
Super heater	390.6	377		370
Reheater	390.6	294		370

Fig. 5 gives the reference state points of the steam cycle. This is required to check the cycle power output for assumed steam flow rate through the cycle components. Steam condition to the HP turbine inlet (point 1 in Fig. 5) is 370°C temperature, 100 bar pressure, and 36 kg/s steam flow rate (assumed and to be checked). The HP turbine has one extraction at 33.6 bar to HP feedwater heater (#6), point 2, and the steam is discharged at 18.58 bar (points 3 and 4). Part of the steam leaving the HP is directed to the HP feedwater heater (#5), and the balance (called cold reheat) is directed to the reheater. The hot reheat (at point 5) at 370°C from the reheater is directed to the LP turbine. The LP turbine has four extractions, one (at point 6) to a de-aerator, and 3 (at points 7, 8, and 9) to LP feed heaters #3, #4, and #5 respectively. The steam is discharged from the LP turbine (point 10) at 0.1 bar pressure. The values of temperature, pressure, enthalpy, and mass flow rates are given in Table 3.

Data given in Table 3 are used to calculate the HP and LP cycle work outputs $W_c(\text{HP})$, and $W_c(\text{LP})$, respectively, as:

$$W_c(\text{HP}) = m_1(h_1 - h_4) - m_2(h_2 - h_4) = 10,357 \text{ kW}$$

$$\begin{aligned} W_c(\text{LP}) = {} & m_5(h_5 - h_{10}) - m_6(h_6 - m_{10}) - m_7(h_7 - h_{10}) \\ & - m_8(h_8 - m_{10}) - m_9(h_9 - h_{10}) = 23,106.1 \text{ kW} \end{aligned}$$

$$\begin{aligned} W_c(\text{total}) &= W_c(\text{HP}) + W_c(\text{LP}) = 23,106.1 + 10,357 \\ &= 33,463.1 \text{ kW} \end{aligned}$$

The network (W_n) is less than cycle work output W_c due to end losses, frictions in turbines and genera-

Table 3
State points of the reference SPP steam cycle

State point	mass flow rate (kg/s)	Pressure (bar)	Temperature (°C)	Enthalpy (kJ/kg)
1	36	100	370	3,005
2	2.7	33.61	238	2,807
3	2.575	18.58	207	2,710
4	30.725	18.58	207	2,710
5	30.715	17.1	370	3,190
6	2.0245	7.98	278	3,016
7	1.634	2.73	168	2,798
8	1.498	0.96	99	2,650
9	1.021	0.29	70	2,500
10	24.5375	0.1	45.81	2,350
11	30.725	–	45	–
12	30.725	–	45	–
13	30.725	14.76	45	188
14	30.725	10	65	271.7
15	30.725	8.7	95	398.9
16	30.725	7.94	127	532.7
17	36	7.94	–	
18	36	125	170	722.5
19	36	112	–	873.2
20	36	103	–	1014.8
21	36	100	311	1407.8
22	36	100	311	2725.5

tor, and steam leakage from the turbine, and is assumed equal to $0.9W_c$(total), or $W_n = 0.9W_c(\text{cycle}) = 30,117\text{kW}$, very close to the nominal work of 30,000 kW.

So, the assumed flow rate to the turbine is suitable.

Concerning the solar energy gained from the HTF to the water in the HX connected to solar collectors, the heat gains to the water in the preheater Q(pre), boiler Q(boiler), super heater

Q(sup), and reheater Q(reh) are:

$$\text{heat gains by water in the preheater}, Q(\text{pre}) = m_1(h_{21} - h_{20}) = 14,148 \text{ kW}$$

$$\text{heat gains by water in the boiler}, Q(\text{boiler}) = m_1(h_{22} - h_{21}) = 47,437 \text{ kW}$$

$$\text{heat gains by water in the super heater}, Q(\text{sup}) = m_1(h_{23} - h_{22}) = 10,062 \text{ kW}$$

$$\text{heat gains by water in the reheater}, \ Q(\text{reh}) = 14,743\text{kW}$$

Table 4
Turbine state points in the reference plant

Turbine Section	P_{inlet} (bar)	P_{outlet} (bar)	h_{inlet} (kJ/kg)	h_{outlet} (kJ/kg)
HP-1	100	33.61	3,005	2,807
HP-2	33.61	18.58	2,807	2,710
LP-1	17.10	7.98	3,190	3,016
LP-2	7.98	2.73	3,016	2,798
LP-3	2.73	0.96	2,798	2,624
LP-4	0.96	0.29	2,624	2,325
LP-5	0.29	0.08	2,325	2,348

$$\text{Total thermal energy gained by water } Q_w = Q(\text{pre}) + Q(\text{boiler}) + Q(\text{sup}) + Q(\text{reh}) = 86,390 \text{ kW}$$

Heat supplied by the solar collectors, Q_s, by assuming 5% thermal energy loss is:

$$Q_s = 90,937 \text{ kW}$$

This means that the share of preheater, boiler, super heater Q, and reheater are 16.37, 54.91, 11.65, and 17.1%, respectively. The plant efficiency $\eta = W_n/Q_s = 30,117/90,937 = 0.331$.

Some data taken from operating plant are given in the Tables 4 and 5, [3].

4. First modification: fuel-assisted SPPs

The limitation of maximum steam temperature, 370°C, is imposed by HTF flowing in the PTC, is the main reason of the above cycle low efficiency. This steam temperature can be raised in a super heater operated by NG to raise it to that of the conventional steam power cycles, say 535°C. The hot gases generated by the NG fuel, (or rejected from gas turbines) supplied to this super heater would leave at temperature higher than that of the incoming steam, say saturated at 100 bar (or 311°C). It is wasteful to discharge these gases, say at 360°C, to the environment. The gases heat content can be used to heat the feedwater returning from the condenser to the solar steam generator. This saves the extracted steam from being supplied to FWHs, and let it expanding to the condenser to give more work. So the first modification would increase the power output and the efficiency of the SPP by using NG-assisted fuel. This can be done, Fig. 8, by:

(1) Superheating the saturated steam leaving the PTC to 535°C, the maximum allowable temperature used in conventional steam power plants.

Table 5
Reference inlet and outlet stream conditions for closed FWH

Heater #	$P_{\text{in\_ stream}}$ (bar)	$P_{\text{in\_water}}$ (bar)	$P_{\text{out\_stream}}$ (bar)	$P_{\text{out\_water}}$ (bar)	$h_{\text{extraction}}$ (kJ/kg)	$h_{\text{in\_water}}$ (kJ/kg)
6	33.61	112.0	20.5	103.56	2,807	873.2
5	18.58	125	9.86	112	2,709.6	722.5
3	2.73	8.7	1.21	7.94	2,798	398.9
2	0.96	10	0.38	8.7	2,624.4	271.7
1	0.28	14.76	0.14	10	2,528.1	174.9

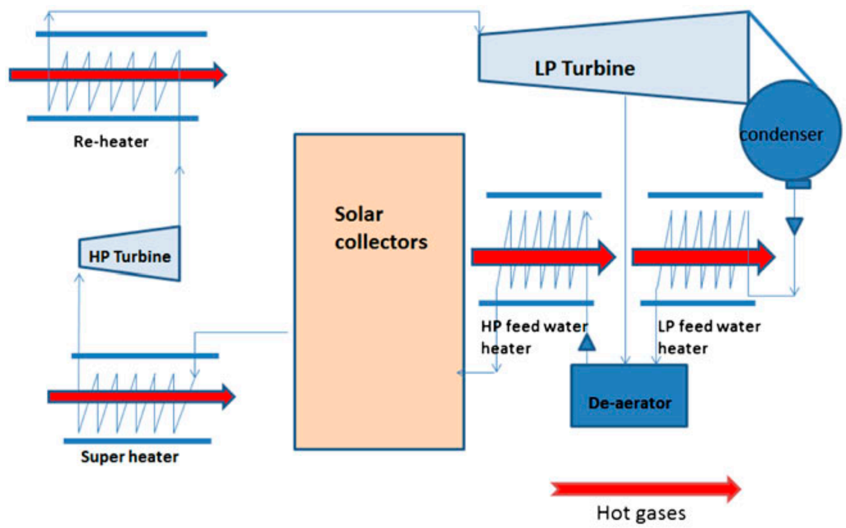


Fig. 8. Schematic diagram of the modified steam cycle using assisted fuel producing hot gases.

(2) Supplying heat to the reheater to raise the steam in the hot reheat to 535°C.

(3) Heating the feedwater in the LP feedwater heaters returning from the condenser to the de-aerator; and in the HP feedwater heaters from the de-aerator to the preheater (economizer).

Moreover, the strategy of using assisted fuel with CSP solar plant or incorporating thermal storage system is necessary to make the SPP dispatchability.

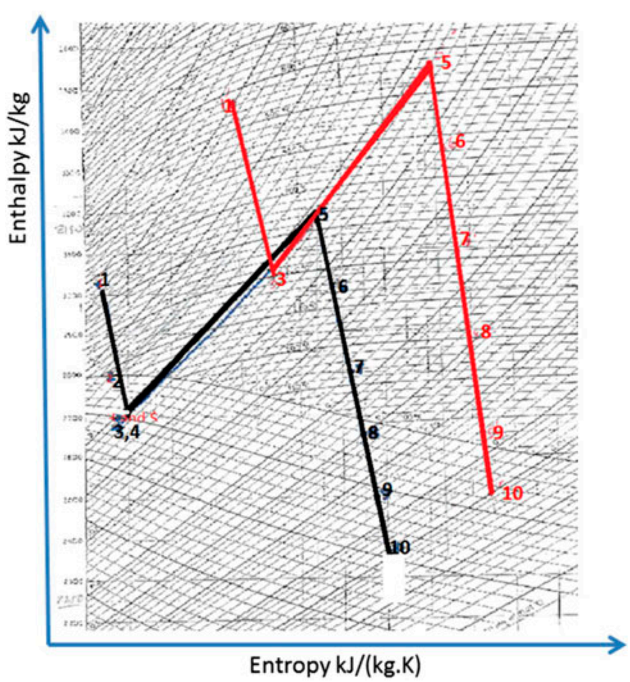


Fig. 9. Plotting reference and modified cycle with fuel assisted hot gases on enthalpy–entropy chart.

Rising the throttling and reheater outlet temperature to 535°C increases significantly the turbine output and improves the cycle efficiency. Also avoiding extracting of steam from the turbine to all feed heaters (except the de-aerator) increases the turbine power output. Plotting of both reference (black lines) and modified cycles (red lines) on the enthalpy–entropy (h–s) chart is given in Fig 9. Notice that the vertical distances between points 1 and 3, and between 5 and 10 represent the work outputs from the HP and LP, respectively. Schematic temperature relations the hot gases and heated water in the sections of the heat HX operated by hot gases are shown in Fig. 10, where upper line is for hot gases and the lower line is for water in each section.

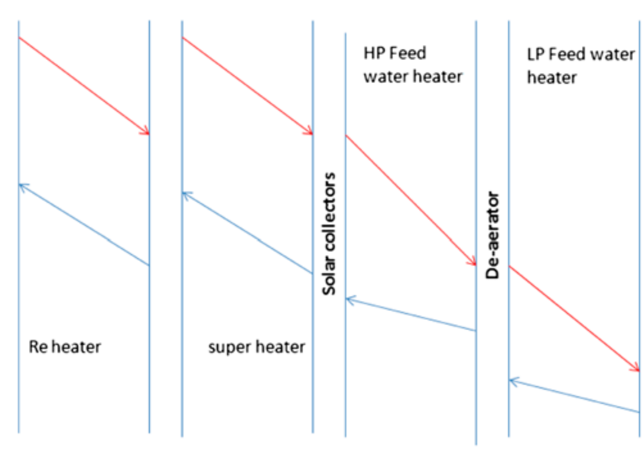


Fig. 10. Schematic temperatures of hot gases (upper line) and water (lower line) in HX operated by hot gases.

The work output for this case can be calculated as:

$$W_c(\text{HP}) = 15,840\,\text{kW}, W_c(\text{LP}) = 40,270.5\,\text{kW}, W_c(\text{total}) = 5,6110.5\,\text{kW}$$

gross work, and $W_n = 50,499.5\,\text{kW}$

The steam mass flow rate discharged from the turbine is 33.586 kg/s.

So, by using NG fuel assisted to raise the steam inlet temperature to the HP and LP turbines to 535°C and to heat the feedwater in the HP and LP feedwater heaters increase the cycle output from 30,117 kW to 50,499.5 kW (68%) increase. The mass flow rates, enthalpy, pressure, and temperature at different points of the cycle are given as:

Condition	Pressure (bar)	Temperature (°C) (assumed)	Enthalpy (kJ/kg)	Mass flow rate (kg/s)
1	100	535	3,470	36
4	18.58	300	3,030	36
5	17.1	535	3,535	36
6	7.98	435	3,340	1.34
10	0.1	45	2,500	24.5375

The thermal energy required for heating the feedwater leaving the condenser at 42°C, 33.59 kg/s, and 174.9 kJ/kg enthalpy to the de-aerator, say at 127°C (532.65 kJ/kg enthalpy) in the LP feed heaters is

$$Q(\text{LP feed heaters}) = 12,732\text{kW}.$$

A check to have the leaving hot gases from the LP feed heater is higher than the dew point is necessary.

Similarly, the thermal load Q(HP feed heaters) for heating the feedwater in the HP feed heaters from the de-aerator at 170°C, 36 kg/s, and 720.87 kJ/kg to the preheater of the solar energy heat exchanger at 1014.8 kJ/kg enthalpy is:

$$Q(\text{HP feed heater by NG}) = 10,581.5\text{kW}$$

The thermal load of the PTCs, which heat the feedwater to its saturated liquid condition and transfer it to saturated vapor is:

$$Q_s(\text{solar}) = 36(2,725.5 - 1,014.8) = 61,585\text{kW}$$

This means that the heat gained by solar collectors was reduced from 86,390 to 61,585 kW, (28.7% decrease); while the power out increased from 30 to 50.5 MW (68% increase)

The thermal load of the super heater by NG = 36 (3,470−2,725.5) = 26,802 kW

The thermal load of the preheater = 18,180 kW

So, the heat added by NG = 68,656 kW

The ratio of heat supplied by the NG to total heat supplied is 52.7%.

Total heat supplied by both solar and NG = 130,241 kW

Considering 5% loss in heat, the actual heat input 137,096 kW.

Again, the rating of this plant can be done by the efficiency η

The efficiency $\eta = W_n/(Q_s + \text{NGheat}) = 0.368$, compared to 0.331 of the previous case.

In this arrangement, the heat gained by solar energy was used only to preheat and boil the water leaving the HP FWH to saturated vapor, or Q(solar) = 61,585 kW, (compared to 86,390 kW heat gained from the solar collector in the first case). This means that the solar collector's area was decreased by 28.7%, while the work output is increased to 50.5 MW. This means that the collector's area per MW is decreased from about 6,000 m^2/MW to 2,541 m^2/WM by using the fuel assisted, a reduction of 57.6%.

5. Second modification: combining desalting plant (DP) with solar power plant

In the reference and first modified cycle EP is produced by electric generator driven by the steam turbine. DW is produced in Qatar to satisfy almost all (99%) of potable water needs. Thermally operated DPs are used, and need steam at relatively LP as heat input to run these DPs. It is wasteful and expensive to generate LP steam to drive the DPs. Usually steam is generated at high temperature and pressure, expanded in steam turbine to the pressure required by the DPs, and then extracted (or fully discharged) to the DPs. This means that steam is used to generate EP before its supply to DPs to generate DW in what is called CPDP, widely used when EP and DW are required. The same can be used by the SPP such as the reference plant given before after some modifications.

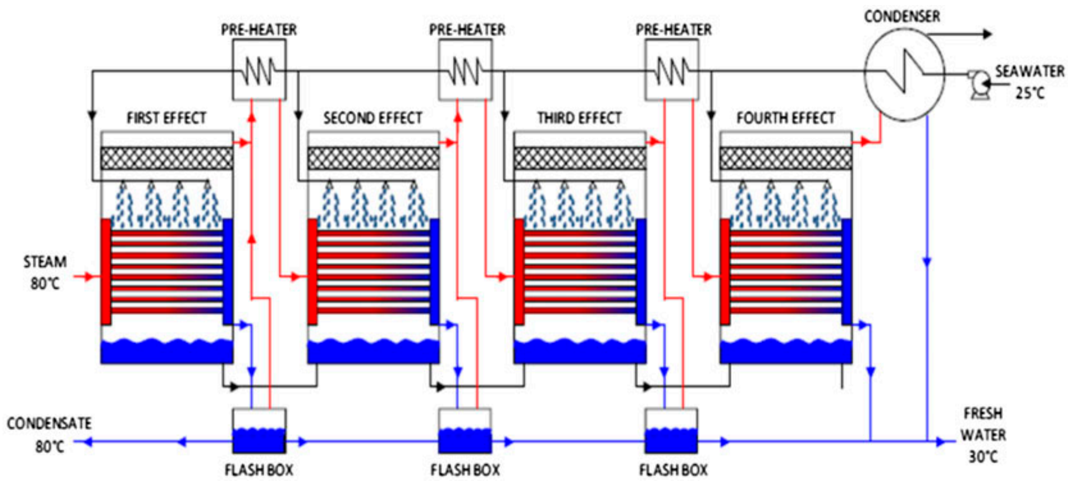


Fig. 11. Schematic diagram of LT-MED plant [7].

5.1. Potential types of thermal DPs

Although DP using seawater reverse osmosis membranes is the most energy efficient and has the lowest desalting water production cost, the GCC is still interested in the thermally operated DP such as multi-stage flash (MSF), thermal vapor compression (TVC), and multi-effect distillation (MED). The MSF is the most used method and it requires thermal energy input of 250–300 MJ/m^3 and about 4 kWh/m^3 pumping energy. Its steam supply is in the range of 2–3 bar to suit the top brine temperature (TBT) of 110–115°C. The TVC has TBT of almost 70C; but it requires much higher steam pressure (3–10 bar) as this steam operates its thermal compressor, and not simply heats seawater as in MSF or MED cases. Its consumed thermal energy may be less than that of the MSF system, but the availability (exergy) of the required steam is higher and more expensive than that of the MSF. It needs pumping energy of 1.5–2 kWh/m^3 to move its streams. The conventional low temperature (LT) MED, Fig. 11, is the most energy efficient thermal operated desalting system. It consumes almost the same thermal energy as the MSF, but at lower pressure and temperature, and thus availability (exergy) than that is required for MSF or TVC. So, the MED is the most economical desalination by distillation. So, it is suggested here to combine LT-MED to the solar plant which is similar to the reference plant.

The second modification is to transfer the reference SPP cycle to CPDP producing both EP and DW. Steam can be bled from any extraction point in the turbine where the steam condition suits that is required by the DP. To get the maximum DW to EP ratio, the turbine type can be changed from condensing steam turbine to back pressure steam turbine (BPST), where all the steam expanded in the turbine up to certain point (pressure) is discharged to the DP. Although the steam discharged at point 9 (where is the saturation temperature is around 70°C) in Fig. 5 satisfies the needed condition for the LT-MED system at full load, the saturation temperature would be less than 70°C when the turbine is operating at part load. As the turbine load decreases, the pressure at the steam inlet to the turbine is decreased by throttling, while the pressure at the condenser end is kept the same. This decreases the pressure along the whole turbine, see Fig. 12.

So, it is suggested to make the discharge pressure at point 8 (where the steam is at 0.96 bar, and 99°C saturation temperature) to overcome the problem of

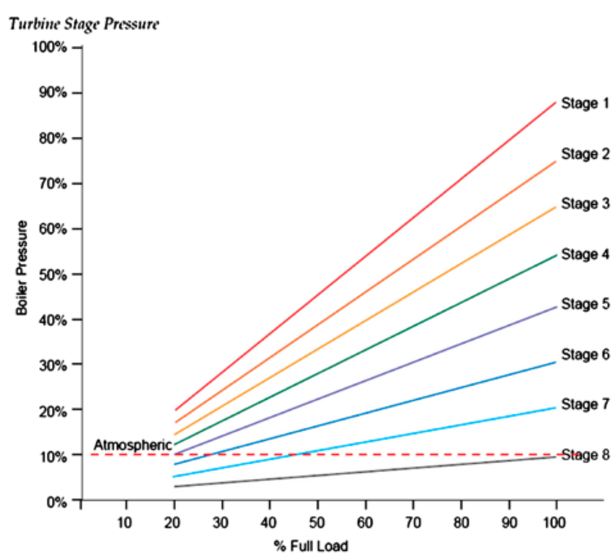


Fig. 12. Variation of pressure along the extraction point of the turbine [8].

decreasing pressure at part load. However, at turbine full load, the steam at point 8 should be throttled to the pressure required by the DP. It is also pointed here that there is a minimum limit to the turbine to supply steam suitable to the DP, when the pressure is at or lower that of 70°C saturation temperature.

This arrangement, shown in Fig. 13, deletes the last LP part of the turbine from point 8 to the end condenser inlet (at point 10); and this the most expensive and inefficient part of the turbine. The condenser and the two last LP FWH (#1 and 2) are also deleted. The work that would be obtained from expanding the steam from 8 to 10 would be lost; and this is equivalent to the thermal energy supplied to the DP.

Accordingly for the same steam flow rate at the turbine inlet (36 kg/s), the work loss between point 8 and end condenser due to discharging steam to the DP and not to condenser is $W_{dc} := W_{9-10} + W_{8-9} = 7,379.4\text{kW}$

This is the cycle loss work, but the net work loss is $W_{dn} = 6,641.5\text{kW}$

The discharged flow rate is 27.06 kg/s.

The LT-MED is rated by the gain ratio (GR), which is defined by the ratio of the distillate product (D) to heating steam supply (S). The GR depends on the number of effect (n). The GR is usually less but close to n. For saturation temperature of 70°C for heating steam, and 38.5°C in the LT-MED end condenser, a temperature difference of 31.5°C is enough to accommodate nine effects with 3.5°C temperature difference across each effect. This can give $\text{GR} = D/S = 8$, or $D = 27.06 \times 8 = 216.45\text{kg/s}$

So, the specific equivalent work (W_{dn}/D) to thermal energy supplied to the DP is:

$$W_{dn}/D = 30.7\text{kJ/kg}\ (8.52\text{kWh/m}^3)$$

The equivalent work for the LT-MED including pumping energy (2 kWh/m^3) is 10.52 kWh/m^3

The steam supplied to the DP is discharged at by the steam turbine at 2,650 kJ/kg enthalpy entering the DP, and leaves as saturated liquid at 70°C (293.07 kJ/kg enthalpy), so the heat gained by the DP is $Q_d = 27.06(2,650 - 293.07) = 63,778.5\text{kW}$, and the specific heat in kJ/kg of desalted water (D) is $Q_d/D = 294.6\text{kJ/kg}$.

The BPST work output consists of the HP turbine work output calculated before as $W_c(\text{HP}) = 10,357$ kW; and the LP turbine work output to be calculated as:

$$\begin{aligned} W_c(\text{LP}) &= 15,596.7\ \text{kW},\ W_c(\text{total}) \\ &= 25,953.7\ \text{kW gross work, and the netwok}\ W_n \\ &= 23,358\ \text{kW} \end{aligned}$$

The rating of the CPDP plant can be done by different methods, namely efficiency η, utilization factor (UF), and modified efficiency η_m.

The efficiency is defined as the network output divided by the solar energy gained:

$$\eta = W_n/Q_s = 23,358.3/90,937 = 0.2568$$

This underestimates the performance of the plant as it does not take in consideration the heat supplied to the DP.

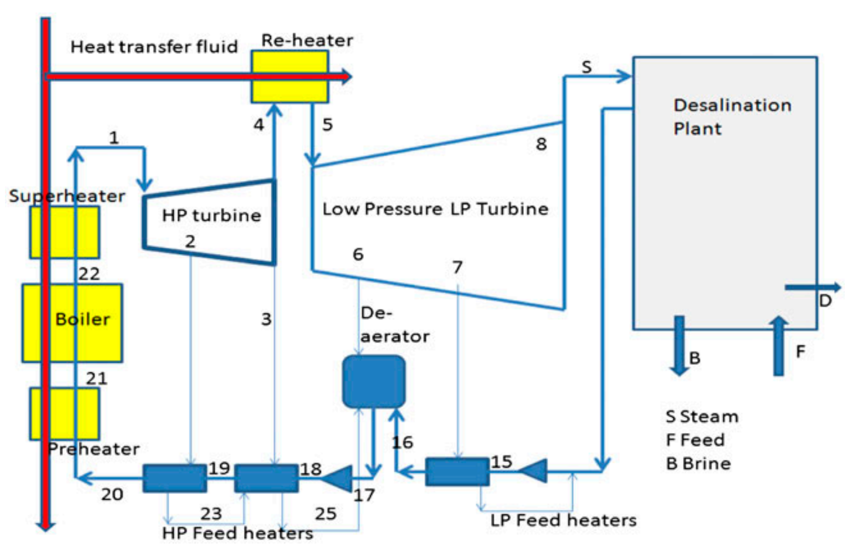


Fig. 13. Schematic diagram of solar CPDP using LT-MED plant.

The UF is defined as total plant output, W_n and the heat supplied to the DP, i.e. $(W_n + Q_d)/Q_s = (23,358.3 + 63,778)/90,937 = 0.95$.

This definition is basically unacceptable from thermodynamics view point as it adds high-quality energy of work W_n to the low available heat Q_d, and thus gives false overestimated performance. A modified efficiency called effectiveness is defined as $(W_n + W_{dn})/Q_s = (23,358.3 + 6,641.5)/90,937 = 0.33$.

6. Third modification

The third modification, Fig. 14, is to transfer the second modification from solar CPDP producing EP and DW to that using NG-assisted fuel as has been done in first modification, but with EP and DW output. The use of NG-assisted fuel increases the EP and DP output. The steps used in the first modification to the reference cycle is repeated here, i.e. superheat the steam supply to the HP turbine and reheat it to the LP steam turbines to 535°C, and heat the feedwater to the steam generator.

The work output for this case, Fig. 8, can be calculated as

$$W_c(\mathrm{HP}) = 15,840\ \mathrm{kW},\ W_c(\mathrm{LP}) = 30,922\ \mathrm{kW},\ W_c(\mathrm{total}) = 46,762\ \mathrm{kW\ gross\ work, and}\ W_n = 42,086\mathrm{kW}$$

The steam mass flow rate discharged from the turbine is 34.64 kg/s. For GR = 8, the DP output D is 277.12 kg/s.

$$Q_d = 34.64(2,650 - 293.07) = 81,644\ \mathrm{kW}$$

So, raising the throttling condition to 353°C, and using NG fuel (or hot gases rejected from gas turbines) to heat the feedwater in the HP and LP feedwater heaters, as well as superheating the steam leaving the boiler driven by solar heat and reheat the steam between the HP and LP turbines increase the cycle output from 23,358.3 to 42,086 kW (80%) increase. It also increases the DP output from 216.45 to 277.12 kg/s (28%).

The thermal energy required for heating the feedwater leaving the first effect of the LT-MED at 70°C, to the de-aerator, say at 147°C (617 kJ/kg enthalpy), and the feedwater from the de-aerator at 7.89 bar and 170°C temperature (720 kJ/kg) can be calculated as:

Q(LP feed heater by NG) = 13,092 kW

The HP feed heaters (now operated by NG) raise the feedwater temperature from saturated temperature at the de-aerator at 7.98 bar (about 720 kJ/kg enthalpy) to the inlet reheater of 1,048 kJ/kg can be calculated as follows. By noticing that the cold HTF leaving the solar collectors at 292°C, and the saturation temperature at 100 bar is 311°C, and the feedwater inlet to the preheater of the solar collectors is assumed at 250°C, then:

Q(HP feed heater by NG) = 10,613 kW

The thermal load of the solar collectors, which preheat and boil the feedwater is:

Q(solar) = 61,585 kW

The thermal load of the super heater by NG = 26,802 kW

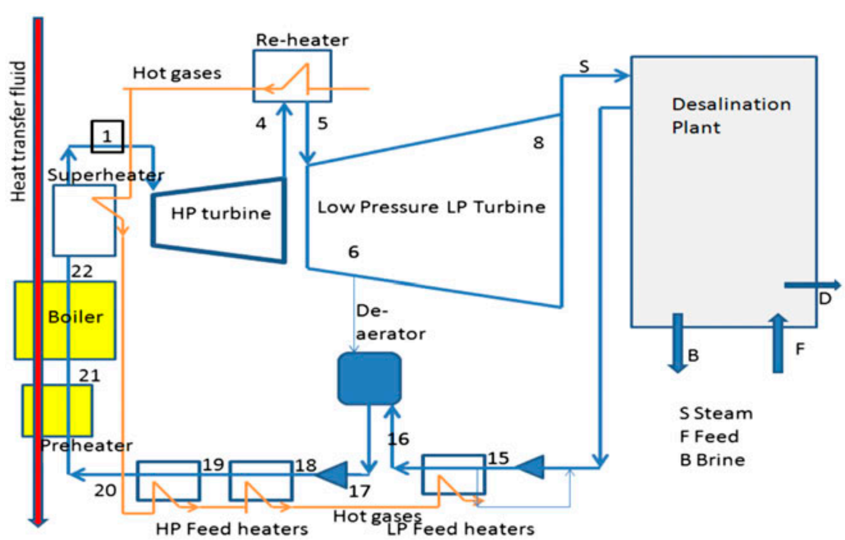


Fig. 14. Schematic diagram of solar fuel-assisted CPDP using LT-MED plant.

The thermal load of the preheater = 18,180 kW

So, the heat added by NG = 58,687 kW
Total heat supplied by both solar and NG = 120,272 kW

Considering 5% loss in heat, the actual heat input is 126,602 kW

Again, the rating of this plant can be done by the methods given in the second modification, namely efficiency η, UF and modified efficiency η_m.

The efficiency $\eta = W_n/(Q_s + \text{NGheat}) = 0.35$, compared to 0.2568 of the previous case

The UF is defined as total plant output, W_n and the heat supplied to the DP, i.e. $(W_n + Q_d)/Q_s = (42,086 + 81,644)/126,602 = 0.95$

Again this gives false overestimated performance.
The modified efficiency is defined as

$(W_n + W_{dn})/Q_t = (42,086 + 8,503)/126,602 = 0.399.$

7. Water consumption

The SPP using PTC requires cooling to condensate the vapor leaving the condenser in case the plant is producing EP only. Selection of cooling technology depends on economics, water availability, and policy. If available, wet cooling is often preferred and provides the lowest cost; however, dry cooling can be used to reduce water consumption, especially in arid areas. A typical trough or power tower plant that employs wet cooling can consume 2.8–3.8 m^3 to produce 1 MWh of solar electricity [2].

7.1. Economic analysis

The high cost of EP unit production is the main obstacle for widespread of large SPP. The SPP generated EP cost (0.19–0.25$/kWh) is much higher than that of conventional PP (0.037–0.05$/kWh) [9].

However, large-scale implementation and technological advancements in thermal SPP are expected to decrease the cost continuously, and may become competitive with continuous increase of fossil fuel prices and associated social costs of carbon emissions.

7.2. Levelized energy cost (LEC)

Any new PP type choice is usually based on the levelized electricity cost (LEC) over the plant life span. The LEC is the real annual cost converted to the equivalent present value of money, [10]. It is an economic assessment of the cost of the electricity-generating system including all the costs over its lifetime: initial investment, operations and maintenance, cost of fuel (if any), and cost of capital. This annualized cost value allows for the comparison of one technology against the other, while differing annual costs are not easily compared. The LEC is based on SPP operating for 30 years. The intent of this analysis was to provide a technical LCE based on the data assumed that would allow a comparison of the performance and costs of the various designs.

The LEC is also defined as the minimum price at which energy must be sold for an energy project to break-even. The LEC is defined in a single formula as, [9].

$$\text{LEC} = \frac{\sum_{t=1}^{n} \frac{I_t + M_t + F_t}{(1+r)^t}}{\sum_{t=1}^{n} \frac{E_t}{(1+t)^t}}$$

where LEC = average lifetime levelized electricity generation cost, I_t = investment expenditures in the year t, M_t = operations and maintenance expenditures in the year t, F_t = fuel expenditures in the year t, E_t = electricity generation in the year t, r = inflation rate, and n = life of the system in years.

The LEC of PP strongly depends on the plant type, CF, and the size of the plant. As the size of the plant increases, its cost/kW decreases.

Typical LEC is usually calculated over lifetime years of the PP and is given in the units of currency per kWh, for example $/kWh or $/MWh.

The LEC of generated EP from two solar plants are considered here, and the characteristics of these plants are briefly discussed. Dispatch ability is a very important for SPP plants to allow delivery of firm power according to demand.

For example, high temperature thermal energy stored during the off-peak periods can be utilized during peak hours or in the evening to generate electricity. It can also be configured with auxiliary gas-fired equipment to supply thermal energy to achieve full power and remove intermittency from operation with insufficient sunlight.

The LEC of generated EP from two plants are considered here.

(1) The first plant is similar to the reference plant, but has solar TES of 7.5 h, and 50 MW net power output.
(2) The second plant is the same steam flow rate to the HP turbine as the first plant, but the solar

collectors are used only to preheat the feedwater and boil it as in the first modification. NG fuel assisted is used to superheat the steam leaving the collectors, reheats the steam leaving the HP turbine, and heats the feedwater to the collectors as in first modification. It has similar TES as the first plant. The power output is increased 68%, while the solar collectors' area is decreased 28.7% compared to the first plant. The NG fuel assisted raises the steam temperature at HP and LP inlets to 535°C, and heat feedwater as in the first modification to the reference plant.

Economic analysis conducted here is for the first solar plant using parabolic trough of 50 MWe net power output with 7.5 h TES to assure the plant capability to cover the peak load period and to raise the capacity factor (CF) to 43%. Many commercial units of 50 MW with TES are already under operation.

The first plant is similar to one of the Andasol's SPP using 510,120 m^2 solar field having 209,644 parabolic mirrors, 22,464 receivers (absorption) pipes of 4 m length each, [9]. The land area is four times the solar collectors' area. Other plant data are: annual DNI is 2,136 kWh/m^2.y (close to that of Qatar), peak solar field efficiency is 70%, and approximate annual average of 50%, and heat storage capacity is 28,500 tons of salt for 7.5 peak load hours. The peak efficiency of the entire power plant is 28% peak efficiency and 15% annual average, and estimated life span is 30 years.

Data in the literature on the cost of different items of the SPP are very limited, and include Refs. 10, 12, and 13. Table 6 gives some of the information taken from these references. The cost items for the two plants are chosen as follows.

Average values for solar field cost in dollars per m^2 ($\$/m^2$) are: 166 for support structure, 56.5 for receivers, 61.5 for mirrors, and 155 for balance (auxiliaries) cost, a total of $\$439/m^2$. Notice that the support structure cost is almost three times of receivers or the mirrors. The solar balance of solar field consists of the remaining items, components and structures that comprise a complete solar field that are not included amongst the steel support structure, receivers and mirrors. For instance: solar tracking system, HTF system, interconnection piping, electronics, and others.

This gives collector solar field cost as $223.943 million (*M*).

There is a big discrepancy between the cost of power block and its balance; one source reported as $2,500/kW, and other reported as $1,183/kW. The value of $2,000/kW is chosen here and this gives the power block cost as $100 M. The average solar storage cost is $875/kW, and this gives the plant TES cost as $43.75 M. Other costs include land preparation of $20/$m^2$, and for 2,040,480 m^2 land (about four times the solar field area), the land preparation would be $40.81 M. This gives a total investment cost of $408.5 M; or $8,170/kW.

Table 6
Data on itemized cost of SPP [9,11,12]

Plant	Sargent and laundy					Fitchner
Trough	Unit	2003 SEGS VI Hybrid	2003 Trough 50 Storage	2008 Trough 100 No storage	2008 Trough 100 Storage	2008 Trough 100 Hybrid
Collector area	m^2	188,000	49,600	767,000	1,110,000	580,000
Capacity	MWe	30	50	100	100	100
Capacity factor	%	22%	47%	33%	51%	25%
Annual capacity output	GWh/y	58	206	290	451	223
Support structure cost	$\$/m^2$	67	67	171	172	160
Receivers cost	$\$/m^2$	43	43	53	53	60
Mirrors cost	$\$/m^2$	43	40	63	63	60
Solar balance cost for parabolic trough plant	$\$/m^2$	234	250	141	141	150
Power block cost and balance	$/kWe	527	306	1,183	1,183	2,500
Thermal storage cost	$/kWe	No storage	985	No storage	765	No storage
Total investment cost	$M	92	254	447	671	559
	$/kWe	3,052	5,073	4,471	6,708	5,594
Annual net electricity output	GWh	58	206	290	451	223
LEC	$/kWhe	0.181	0.143	0.168	0.157	0.239

For 43% CF, the annual generated EP is 188.34 GWh.

The operation and maintenance (O & M) was reported by Ref. [4] as 2.9c/kWh, and thus total annual O & M for this plant is $5.462 M.

For $408.5026 M capital cost (principal), 8% interest rate (IR), and 30 years (y) loan, the accumulated interest is $490.203 M. The total interest and principal of $898.7056 M is to be paid in 30 y with annual capital cost is $29.956 M. When the IR is decreased to 6%, the interest and principal is $776.155 M with $25.872 M annual capital cost.

7.3. Water cost

A typical trough plant using wet cooling condenser can consume 2.8–3.8 m^3 of water to produce one MWh of solar electricity, [13]. So, the annual consumed water by the first plant is 659,190 m^3 based on 3.5 m^3/MWh. By knowing that desalted water cost is about $3/$m^3$, the water cost is $1.98 M. So, the total annual cost is $37.342 M for 8% IR, and 33.230 M for 6% IR. This gives the LEC as $0.199/kWh for 8% IR, and $0.177/kWh for 6% IR.

The above example shows that the high capital cost per MW of $8,155/MW for the SPP, compared to $2000 per MW for the power block (or steam power plant), is the main item hindering SPP application. Another factor is the required land area (about 40,000 m^2/MW when TES is used). The TES is needed for plant dispatch ability. The main item of the capital cost is the solar collectors representing 54.8% of the capital cost.

The second plant has the same steam flow rate to the HP turbine as the first plant but NG fuel assisted is added. Similar to the previous analysis of the reference and the first modified plants, where NG assisted and throttling condition was raised, the heat gained by solar collectors was reduced from 28.7%, while the work output was raised 68%. So, the area of solar collectors becomes 363,716 m^2, while the power output becomes 84 MW. In this case, the solar collector's area per MW becomes 4321 m^2/MW. The cost of the solar field is $159.6714 M. The power block and its balance would be equal to $168 M ($2000/kW as first plant). Since the percentage of solar energy in the second to the first case is 71.3%, the TES cost would be $31.194 M. The land preparation cost is $29.1 M. This gives a total investment cost of $387.965 M; or $4,619/kW. For $387.965 M capital cost (principal), 8% IR and 30 years (y) loan, the accumulated interest is $465.558 M. The total interest and principal of $853.532 M is to be paid in 30 y with annual capital cost is $28.450 M. When the IR is decreased to 6%, the

Table 7
Itemized cost of the economical considered first and second plants

Items	First plant (solar only) 50 MW	Second plant (fuel-assisted solar) 84 MW
Solar collectors area, m^2	510,120	363,716
Land required area, m^2	2,040,480	1,454,862
Solar field cost, $M	223.943	159.6714
Power block cost, $M	100	168.000
Thermal storage cost, $M	43.75	31.194
Land preparation cost	40.8096	29.1
Total capital cost, $M	408.5026	387.965
Capital cost/kW	8,170	4,619
Interest over 30 y, 8% IR	490.203	465.558
Interest over 30 y, 6% IR	367.652	349.169
Interest over 30 y + Principal, 8% IR	898.7056	853.523
Interest over 30 y + Principal, 6% IR	776.1546	737.1335
Annual fixed payment, $M, 8% IR	29.957	28.450
Annual fixed payment, $M, 6% IR	25.872	24.571
Electric power generation GWh	188.34	316.411
Operation and maintenance 0.29c/kWh in $M	5.46186	9.176
Water cost, $3/m3 and 3.5 m3/MWh in $M	1.9776	3.322
Annual fuel cost, $M, $4/GJ	0	6.525
Annual fuel cost, $M, $7.5/GJ		12.234
Total annual cost, 8% IR	37.396	47.473
Total annual cost, 6% IR	33.311	43.594
LEC, $/kWh for 8% IR	0.199	0.150
LEC, $/kWh for 6% IR	0.177	0.138

interest and principal is 737.1335 M with \$24.571 M annual capital cost.

The annual power output is 316.411 GWh (compared to 188.34 GWh of first case). The O and M cost (for \$0.029/kWh) is \$9.176 M, and water cost of \$3.322 M.

Another factor is the cost of fuel here. The share of NG fuel heat to the total heat supplied (by solar and NG) is 52.7%, or 123.4 MWt. For net efficiency of 0.368, the heat added by the NG fuel is 1.631×10^6 GJ. The cost of NG is in the range of \$4/GJ [14], then the annual fuel cost is $4 \times 1.631 \times 10^6 = \$6.525$ M.

When the NG fuel cost is calculated, as \$7.5/GJ, the annual consumed NG cost is \$12.234 M.

So, the total annual cost is \$47.473 M when the IR is 8% and \$43.594 M when the IR is 6%. This gives the LEC as \$0.15/kWh when the IR is 8% and \$0.138/kWh when the IR is 6%.

The itemized cost of the considered first and second plants are given in Table 7.

Simple economics facts may be mentioned here. The cost of one barrel (bbl) of oil now is about \$100/bbl, which contains 6 GJ. If this is used to produce EP in a power plant of 40% efficiency, it produces 666.7 kWh. This gives the fuel cost per kW when crude oil is used as \$0.15/kWh, and this would be raised to \$0.222/kWh if the oil cost reaches \$150/bbl. The corresponding kWh cost, when NG of \$4/GJ is used, is \$0.036/kWh. So, the use of oil to partially operate PP (practice used in Kuwait and Saudi Arabia) is very expensive; besides it is negatively affecting the environment by CO_2 and other polluting gases emission.

All the GCC, except Qatar, consumes all their NG productions, and have to import NG to run their PP. The cost of NG is cheap now, but this cannot be guaranteed on the long run, and its secure supply is not guaranteed. It is noticed here that while Iran and Saudi Arabia have the second and fourth NG resources, respectively, worldwide, all their NG production are consumed locally. So, in PP using crude oil is very expensive, and using NG is not securely guaranteed and its cost can be rise.

NG is mainly a local commodity as it is difficult to transport, meaning that gas production has been stuck in the producing country getting low local prices, but not outside where transported liquid natural gas (LNG) can become expensive. Starting in 2015, the US NG can be transported away and sold into the global market via LNG exportation. Gas costs over \$10/GJ in Europe and over \$15 in Asia [14]. This concludes that more emphasis should be given the use of renewable energy such as solar and wind energy.

8. Conclusion

The use of crude oil to produce EP is very expensive, and cheap NG is not secured in all GCC except Qatar. So, the use of alternative prime energy (to oil and NG) such as solar energy should be considered. The use of concentrated solar power to produce EP and DW is studied. Besides using NG as assisted fuel to improve dispatch ability of the SPP, the feasibility of using NG to raise the steam temperatures to the HP and LP turbines is considered. This increases the SPP power output and cycle efficiency, while significantly decreases the required areas of the PTC cost, as well as capital cost per installed kW. A reference SPP was presented and analyzed. Then three modifications for this plant are given. The first is using NG to raise the steam temperatures to the HP and LP turbines. The second is to transfer the SPP producing EP only to solar CPDP producing both EP and DW. The third is to use NG to raise the steam temperature to the turbines in the CPDP. An economic analysis to calculate the cost of the produce the EP is also presented.

When NG was used to raise the steam temperature at the HP and LP inlet to 535°C and only EP is generated, the solar collector's area was decreased by 28.7%; while the work output is increased from 30 to 50.5 MW. In CPDP, raising the throttling condition to 353°C by using NG increases the cycle output from 23,358.3 kW to 42,086 kW; 80% increase. It also increases the DP output from 216.45 kg/s to 277.12 kg/s (28%).

List of abbreviations

BPST	—	back pressure steam turbine
CPDP	—	cogeneration power desalting plants
CSP	—	concentrated solar power
CST	—	condensing steam turbine
CF	—	capacity factor
D	—	distillate, kg/s
DP	—	desalting plant
DW	—	desalted seawater
DNI	—	direct normal irradiance
E	—	East
EP	—	electric power
ECST	—	extraction-condensing steam turbine
FERC	—	Federal Energy Regulatory Commission
FWH	—	feedwater heaters
GHG	—	greenhouse gases
GR (D/S)	—	gain ratio of distillate output D divided by heating steam supply S

GCC — Gulf cooperation countries
H — specific enthalpy, thermodynamics property, kJ/kg
HCE — heat collection element
HP — high pressure
HTF — heat transfer fluid
HX — heat exchangers
IR — interest rate
LEC — average lifetime levelized electricity generation cost
LP — low pressure
LT-MED — low temperature multi-effect distillation desalting system
LNG — liquid natural gas
MSF — multi-stage flash desalination system
MED — multi-effect distillation
NG — natural gas
N — North
PP — power plants
PTC — parabolic trough collectors
SCA — solar collector assembly
SEGS — solar electric generation system, solar power plants in California, US
SWRO — seawater reverse osmosis desalination system
SPP — solar power plants
TES — thermal energy storage
TVC — thermal vapor compression desalination system
TBT — top brine temperature
UF — utilization factor of CPDP, defined by $(W_n + Q_d)/Q_s$
UAE — United Arab Emirates

List of symbols

As/MW — solar field area per mega Watt
E_t — electricity generation in the year t
F_t — fuel expenditures in the year t
I_t — investment expenditures in the year t
M_t — operations and maintenance expenditures in the year t
M — steam mass flow rate through turbine in kg/s
n — life of the system in years
Q — thermal energy (heat) flow rate, kJ/s or kW
r — inflation rate
S — specific entropy, thermodynamics property, kJ/(kg K)
S — South, and steam
W — West and work
W_c — work calculated from cycle
W_n — net work
W_{dn} — work loss due to steam extraction to desalting plant
η — efficiency
η_m — modified efficiency

References

[1] A.M. Patnode, Simulation and Performance Evaluation of Parabolic Trough Solar Power Plants, University of Wisconsin, Madison, WI, 2006.
[2] Sun Shot Vision Study February 2012. Available from: http://www1.eere.energy.gov/solar/pdfs/47927.pdf
[3] Solar Power Plants Worldwide. Available from: http://www.nrel.gov/csp/solarpaces/operational.cfm
[4] Parabolic solar power network, National Renewable Energy Laboratory (NREL). Available from: http://www.nrel.gov/csp/troughnet/
[5] Solar Millennium AG, The Parabolic Trough Power Plants Andasol 1 to 3, The largest Solar Power Plants in the World–Technology Ppremiere in Europe. Available from: http://www.solarmillennium.de/upload/Download/Technologie/eng/Andasol1-3engl.pdf (accessed March 16, 2012).
[6] T.A. Stuetzle, Automatic Control of the 30 MWe SEGS VI Parabolic Trough Plant, University of Wisconsin, Madison, WI, 2002.
[7] R. Barraza, Solar Desalination Systems Research and Modeling Process, M.Sc. thesis, University of Wisconsin, Madison, WI, 2011.
[8] M.A. Darwish, M.E. Eleshaky, N.M. Al-Najem, B.S. Alazmi, Alternative primary energy for power desalting plants in Kuwait: The nuclear option II—The steam cycle and its combination with desalting units, Desalin. Water Treat. 1 (2009) 42–57.
[9] Wen Zhang, Concentrating Solar Power—State of the Art, Cost Analysis and Pre-Feasibility Study for the Implementation in China, Diplomarbeit, Institut für Energiewirtschaft und Rationelle Energieanwendung, Stuttgart, 2009.
[10] M.A. Darwish, Prospect of using alternative energy for power and desalted water productions in Kuwait, Desalin. Water Treat. 36 (2011) 219–238.
[11] MENA Regional Water Outlook, Part II, Desalination Using Renewable Energy, FINAL REPORT, *March 2011*, 6543P07/FICHT-7109954-v2. Available from: http://wrri.nmsu.edu/conf/conf11/mna_rdrens.pdf
[12] Sargent, Lundy LLC Consulting Group, Assessment of Parabolic Trough, Power Tower, and Dish Solar Technology, Cost and Performance Forecasts, National Renewable Energy Laboratory (NREL), Golden, CO, 2008.
[13] C.S. Turchi, M.J. Wagner, C.F. Kutscher, Water Use in Parabolic Trough Power Plants: Summary Results from WorleyParsons' Analyses, Technical Report NREL/TP-5500-49468, National Renewable Energy Laboratory (NREL), Golden, CO, December 2010. Available from: http://www.nrel.gov/docs/fy11osti/49468.pdf
[14] Natural Gas Prices—Historical and Forecast. Available from: http://www.energyshop.com/es/homes/gas/gaspriceforecast.cfm