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THERMAL ANALYSIS OF HYBRID SOLAR POWER PLANT

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ABSTRACT
Solar thermal power plants are commercially proven in more than a decade of successful operation. As steam power plant, solar thermal power plant can be combined with fossil energy sources. It is possible to retrofit existing conventional steam power plant with solar collectors (e.g., flat plate solar collectors) as an additional solar steam generator. It causes the higher the temperature of steam and so the higher the efficiency of the cycle. In this paper based on 1st, and 2nd of thermodynamic and exergy concept, several parts of power plant studied. Plant highest thermal cycle efficiency is determined for mean day of each month during a year and new operating conditions are proposed.

INTRODUCTION
In a world having finite natural resources and increasing energy demand by developing countries, it becomes increasingly important to understand the mechanisms that degrade energy and resources, and develop systematic approaches for improving the design of energy systems and reducing the impact on the environment [1]. During 1984-1990, a number of nine solar power plants were built in the southern California. Each one consists of two cycle, an oil cycle and steam cycle like rankine cycle. For oil cycle used parabolic trough solar collectors to heat oil as heat transfer fluid, that used for generate steam for rankine cycle power plant. However, for a variety of economic reasons, no new domestic or international parabolic trough power plants have been constructed since that time. To improve system cost, integrated solar combined cycle system (ISCCS) proposed. An ISCCS combined a solar field and a combined cycle.

NOMENCLATURE

A		area, m2
AC		air compressor
GT		gas turbine
e		specific exergy, kJ/kg cover
$\dot{E}_x$		exergy rate, kW
h		specific enthalpy, kJ/kg ρ
IR	[MW]	exergy destruction,
$\dot{m}$	[kg/sec]	mass flow rate,
P	[bar]	pressure,
S	[kJ/kgK]	specific entropy,
T	[K]	temperature,
$\dot{Q}$	[MW]	heat transfer rate,
$\dot{H}$	[MW]	enthalpy rate,
$\dot{W}$	[MW]	work rate, power,

Subscripts

f	exergetic factor a ambient
ISCCS	integrated solar combined cycle system c collector
HRSG	heat recovery steam generator
IP	low pressure
HP	high pressure
SH	super heater
EVA	evaporator
ECO	economizer

Special characters

α	absorptivity of absorber
τ	transmissivity of
γ	intercept factor
r	reflectivity/density(m3/kg)
η	first law efficiency
K(θ)	incidence angle modifier
δ	fuel depletion rate, %
ψ	flow exergy, kJ/kg
ξ	productivity lack, %
ε	exergy efficiency

Subscripts

s	steam/absorber/solar
u	length, m
N	number of collectors
D	destruction
b	beam
0	dead state
W_o	aperture, m
F	fuel
I	solar intensity,
N	normal
o	optical/oil/outlet
L	loss
r	receiver
Tot	total
i	inlet stream

In this system quantitative of steam generated by solar field and surplus of steam for high pressure and intermediate pressure turbine generate with hrsg. Indeed, solar field Couse to increase steam generation that increase rankine output or decrease fuel consumption that decrease costs.

SYSTEM DESCRIPTION

Iran is located in the northern moderate dry region on the earth's mean latitude close to the equator. On the average, Iran has a dry climate with low rainfall. Taking the average annual temperature into consideration, Iran is one of the countries which have a substantial amount of good solar irradiation [13]. The region of Qom with 31^0 North latitude is an excellent site for the construction and operation of a solar field as a renewable energy source. Qom combined cycle power plant constrauted on 4 gas units and tow staeam units. Each gas unit have a 100 MW turbine and each steam unit have a 100 MW steam turbine. In this paper we want develop this site with solar field to save our fuel consumption, because Qom located in the central position of Iran and in this area we have limitation of consume fuel due very large industrial in central and large cities. Fig. 1 shows the plan of integrated solar combined cycle in Qom.

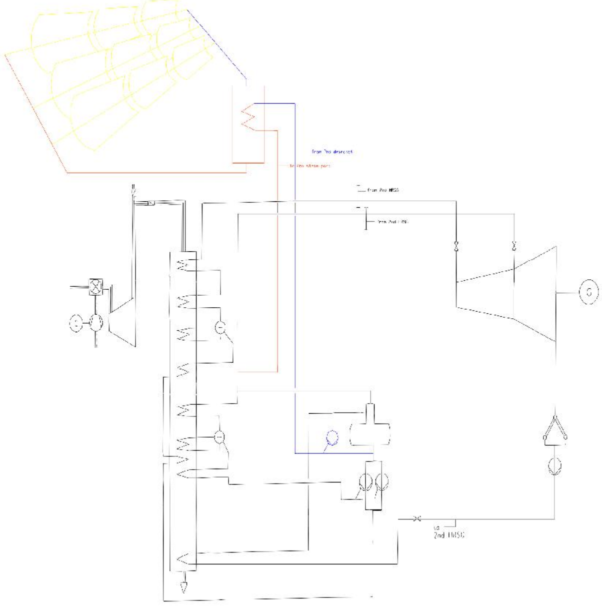

Figure 1 integrated solar combined cycle in Qom

- four gas turbine units with natural gas fuel (Table 1).
- Two pressure heat recovery steam generator. The high and low pressure steam conditions were as follows: 80 bar and 485.4 °C and 5.4 bar and 194.7 °C.
- A no reheat two pressures steam turbine.

For the purpose of analysis the following assumptions are made:

- The Integrated Solar Combined Cycle System operates at a steady state.
- Ideal-gas mixture principles apply for the air and the combustion products.
- The combustion is ideal and heat losses have been considered from the collectors, solar heat exchangers and stacks. All other components operate without heat loss.
- The exit temperature is above the dew point temperature of the combustion product.

We used the mineral oil vp1 that can reach 393 °C and it is a convenient temperature for trough collectors.

Analysis

Mass, energy and exergy balance equations.

In general, the mass balance equation can be expressed in the form:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out}$$

Where $\dot{m}$ is the mass flow rate, and the subscript in stands for inlet and out for outlet. The energy balance is

$$\dot{Q} + \sum \dot{m}_{in} h_{in} = \dot{W} + \sum \dot{m}_{out} h_{out}$$

Where $\dot{Q}$ is the rate of net heat input, $\dot{W}$ is the rate net work output, and h is the enthalpy per unit mass.

Unlike energy, exergy is not subject to a conservation law (except for reversible processes). Rather exergy is consumed or destroyed, due to irreversibilities in any real process. The general exergy balance can be written as follows:

$$\sum \dot{Ex}_{in} - \sum \dot{Ex}_{out} = \sum \dot{Ex}_D$$

Or

$$\dot{Ex}_{heat} - \dot{Ex}_{work} + \dot{Ex}_{mass,in} - \dot{Ex}_{mass,out} = \dot{Ex}_D$$

And,

$$\dot{Ex}_{heat} = \sum \left(1 - \frac{T_0}{T_k}\right)\dot{Q}_k$$

$$\dot{Ex}_{work} = \dot{W}$$

$$\dot{Ex}_{mass,in} = \sum \dot{m}_{in} \Psi_{in}$$

$$\dot{Ex}_{mass,out} = \sum \dot{m}_{out} \Psi_{out}$$

Where k $\dot{Q}_k$ is the heat transfer rate through the boundary at temperature T_k at location k and $\dot{W}$ is the work rate.

The flow (specific) exergy is calculated as follows:

$$\Psi = (h - h_0) - T_0(s - s_0)$$

Where h is enthalpy, s is entropy, and the subscript zero indicates properties at the restricted dead state of P_0 and T_0.

An exergy rate balance for the system reads:

$$\dot{Ex}_F = \dot{Ex}_p + \dot{Ex}_L + \dot{Ex}_D$$

Where $\dot{Ex}_D$ and $\dot{Ex}_L$ denotes the rates of exergy destruction and exergy loss, respectively.

Defining inlet and outlet exergy for each component, the destruction of exergy would be gained for each component and exergy efficiency will be find out from[6]:

$$\eta = \frac{net\ exergy\ rate\ of\ product}{net\ exergy\ rate\ of\ input}$$

or exegetic efficiency ε is the ratio between product and fuel:

$$\varepsilon = \frac{E\dot{x}_P}{E\dot{x}_F} = 1 - \frac{E\dot{x}_L + E\dot{x}_D}{E\dot{x}_F}$$

Relative irreversibility defined as:

$$\chi_i = \frac{(E\dot{x}_D)_i}{(E\dot{x}_D)_{Tot}}$$

Note that exergy is always evaluated with respect to a reference environment (i.e., dead state). When a system is in equilibrium with the environment, the state of the system is called the dead state due to the fact that the exergy is zero.

Relations for analysis of collectors field:
Energy analysis
Collector subsystem:
Energy received by the collectors system is [8]:

$$Q_i = A.I.N = (I_b.R_b).W..L.N$$

Where A=$W..L$ and N are the number of collators.
Energy absorbed by the collectors is:

$$Q_s = \{K(\theta)[\rho(\tau\alpha)_n\gamma]\}Q_i$$

Now we defined optical efficiency for parabolic trough collectors is:

$$\eta_{\text{۰}} = K(\theta)[\rho(\tau\alpha)_n\gamma]$$

First law efficiency for collector subsystem:

$$\eta = \frac{Q_s}{Q_i}$$

Receiver subsystem:
Useful energy delivered to the fluid in the receiver (Q_u):

$$Q_u = N.\dot{m}_f.C_{pf}(T_{fo} - T_{fi})$$

$$\text{Energy loss} = Q_s - Q_u$$

$$\text{\% energy loss} = [(Q_s - Q_u)/\,Q_s].\text{١٠٠}$$

First law efficiency for receiver subsystem:

$$\eta = \frac{Q_u}{Q_s}$$

Overall efficiency of the collector-receiver subsystem:

$$\eta = \frac{Q_u}{Q_i}$$

Exergy analysis
Collector subsystem:
Exergy received by the collector subsystem is:

$$E\dot{x}_i = (\text{١} - \frac{T_a}{T_s})\dot{Q}_i$$

Exergy absorbed by the collector absorber $E\dot{x}_c$:

$$E\dot{x}_c = (\text{١} - \frac{T_a}{T_r})\dot{Q}_s$$

Exergy loss = irreversibility (IP)= $E\dot{x}_i - E\dot{x}_c$

% exergy loss $=(\frac{IP}{E\dot{x}_c}).100$

Second law efficiency:

$$\varepsilon = \frac{E\dot{x}_c}{E\dot{x}_i}$$

Receiver subsystem:

Exergy absorbed by the collector $E\dot{x}_c$:

$$E\dot{x}_c = (\text{١} - \frac{T_a}{T_s})\dot{Q}_s$$

% exergy loss $=(\frac{IP}{E\dot{x}_c}).100$

the exergy given to the working fluid or the usedful exergy is obtained from:

$$E\dot{x}_u = N.\dot{m}_f[(h_{fo} - h_{fi}) - T_a(s_{fo} - s_{fi})]$$

Second law efficiency:

$$\varepsilon = \frac{E\dot{x}_u}{E\dot{x}_c}$$

Hence, the overall second law efficiency of the collector-receiver circuit is:

$$\varepsilon = \frac{E\dot{x}_u}{E\dot{x}_i}$$

heat engine subsystem:
air compressor:
energy analysis

$$\dot{W}_{AC} = \dot{m}_{air}(h_o - h_i)$$

Exergy analysis

$$\dot{I}_{AC} = \dot{m}_{air}(\varepsilon_o - \varepsilon_i) + \dot{W}_{AC}$$

Second law efficiency:

$$\eta_{AC} = \frac{\dot{m}_{air}(\varepsilon_o - \varepsilon_i)}{\dot{W}_{AC}}$$

Combustion chamber
Exergy analysis:

$$\dot{I}_{comb} = \dot{m}_a\varepsilon_a + \dot{m}_f\varepsilon_f - \dot{m}_p\varepsilon_p$$

$$\eta_{comb} = \text{١} - \frac{\dot{I}_{comb}}{\dot{m}_a\varepsilon_a + \dot{m}_f\varepsilon_f}$$

Gas turbine:
Energy, exergy and second law efficiency respectively:

$$\dot{W}_{GT} = \dot{m}_p(h_i - h_o)$$

$$GT = \dot{m}_p(\varepsilon_o - \varepsilon_i) - \dot{W}_{GT}$$

$$\eta_{GT} = \frac{\dot{W}_{GT}}{\dot{m}_p(\varepsilon_i - \varepsilon_o)}$$

we have same equation for steam turbine.
Heat recovery steam generated (HRSG):

$$\dot{I}_{HRSG} = \dot{m}_g(\varepsilon_{g,in} - \varepsilon_{g,out}) + \dot{m}_w(\varepsilon_{w,in} - \varepsilon_{w,out})$$

$$\eta_{HRSG} = \frac{\dot{m}_w(\varepsilon_{w,out} - \varepsilon_{w,in})}{\dot{m}_g(\varepsilon_{g,in} - \varepsilon_{g,out})}$$

In any HRSG component we can used same relative for irreversibility and second law efficiency.
Finally for total power plant we can write:

$$\dot{W}_{net} = \dot{W}_{GT} + \dot{W}_{AC} + \dot{W}_{ST} + \dot{W}_{pump}$$

First law and second law efficiency for ISCCS are respectively:

$$\eta_{total} = \frac{\dot{W}_{net}}{\dot{m}_f\,LHV}$$

$$\eta_{total} = \frac{\dot{W}_{net}}{\dot{m}_f\,\varepsilon_f}$$

That LHV and ε_f must be calculate for natural gas .
For ε_f in table 1 we have:

$$\overline{\varrho^{\cdot}}_{ch,NG} = \sum \mathrm{Y_i}\, \overline{\varrho^{\cdot}}_{ch,i}$$

$$M = \sum \mathrm{Y_i}\, M_i = 19.097\, \frac{Kg}{Kmol}$$

Fuel exergy obtained with above relative is in standard condition (p=0.1 Mpa), thus for correct it use following equation:

$$\varrho_{ch,NG} = \frac{\varrho_{ch,NG}}{M_{NG}} = \frac{\overline{\varrho^{\cdot}}_{ch,NG} + \bar{R}T.\ \ln(\frac{p}{p_s})}{M_{NG}}$$

$$=50433 \quad Kj/Kg$$

	$(\mathrm{Y_i})$	$\mathrm{M_i}$ (Kg/Kmol)	$\mathrm{Y_i}$ x $\mathrm{M_i}$	$e^{\cdot}_{ch}$	$\mathrm{Y_i}$ x $e^{\cdot}_{ch}$
CH_4	0.848	16.042	13.588	831.65	705.24
C_2H_6	0.0077	30.068	2.998	1495.84	150.63
C_3H_8	0.0362	44.094	1.596	2154	77.97
n-C_4H_{10}	0.0055	58.120	0.378	2805.8	15.43
Iso-C_4H_{10}	0.0026	58.120	0.209	2351.7	6.02
n-C_5H_{10}	0.0007	72.146	0.051	3463.3	2.42
Iso-C_5H_{10}	0.0008	72.146	0.058	3086	2.42
N_2	0.0014	28.02	0.039	0	0
CO_2	0.0041	44.01	0.18	0	0
SUM	1	-	19.097	-	960.18

In the same method for LHV we have:
LHV=11605 Kcal/Kg
=48555 Kj/Kg

NUMERICAL RESULTS AND DISCUSSION

In this study, numerical results are based on site design condition with ambient temperature of 45 °C and a relative humidity of 42.5 percent and wind speed 3 m/s. The analysis was carried out on 21 Jun in Qom at 12:00 noon (LAT). At this hour, solar radiation intensity at the plant site is about 700 W/m2.

The solar field considered in this site is comprised of 56 loops and for each loop, 8 collectors from type of LS-3 [5] which are single axis tracking and aligned on a north–south line, thus tracking the sun from east to west. Various design parameters of these collectors are given in Table 2.

Aperture Area per SCA (m2)	545
Mirror Segments	224
Aperture (m)	5.76
Average Focal Distance (m)	0.94
HCE Absorptivity	0.96
HCE Emittance	0.17
HCE Transmittance	0.96
Mirror Reflectivity	0.94
Length	99
Concentration Ratio	82
Annual Thermal Efficiency (%)	53
Optical Efficiency (%)	0.8

Table 2, LS-3 collectors specifications

Therminol VP-1 is used as heat transfer fluid. Therminol VP-1 is a eutectic mixture of 73.5% diphenyl oxide and 26.5% biphenyl. It is usable as a liquid or as a boiling-condensing heat transfer medium up to 750°F (400°C). It is miscible and interchangeable (for top-up or design purposes) with other similarly constituted diphenyl-oxide/biphenyl fluids.

Physical Property Formulae of Liquid:
Density (kg/m3) = - 0.90797 * T(°C) + 0.00078116 * T2 (°C) - 2.367 * 10-6 * T3(°C) + 1083.25
Heat capacity (kJ/kg.K) = + 0.002414 * T(°C) + 5.9591 * 10-6 * T2(°C) - 2.9879 * 10-8 * T3(°C) + 4.4172 * 10-11 * T4 (°C) + 1.498
Thermal Conductivity (W/m.K) = - 8.19477 * 10-5 * T(°C) - 1.92257 * 10-7 * T2(°C) + 2.5034 * 10-11 * T3 (°C) - 7.2974 * 10-15 * T4(°C) + 0.137743

Total solar radiation incident on the collector subsystem is
Q_i= 85.7175 MW, total energy absorbed by the absorbers is Q_s= 67.717 MW. Assuming the collector fluid entering temperature at 291 °C and the mass flow rate of fluid in the collectors to be 16.5 kg/sec, the outlet collector fluid temperature is found to be 393 °C.
Table 3 shows the results of energy analysis Table 4 shows the results of exergy analysis for the solar field only.

subsystem	Energy received (Mw)	Energy delivered (MW)	Energy loss (MW)	Energy loss (%)	First law efficiency (%)
Collector	85.72	67.72	18	0.21	0.79
Receiver	67.72	37.25	30.47	0.45	0.55
Collector-receiver	85.72	37.25	48.47	0.57	0.44

Table 3 results of energy analysis

subsystem	Exergy received (Mw)	Exergy delivered (MW)	Exergy loss (MW)	Exergy loss (%)	second law efficiency (%)
Collector	81.05	32.15	48.9	0.60	0.38
Receiver	32.18	19.16	13.02	0.41	0.62
Collector-receiver	81.05	19.6	61.45	0.76	0.24

Table 4, results of exergy analysis

A comparison of the exergy and energy analysis is shown in Table 5.

subsystem	Energy loss (MW)	Exergy loss (%)	First law efficiency (%)	second law efficiency (%)
Collector	48.9	0.60	0.79	0.38
Receiver	13.02	0.41	0.55	0.62
Collector-receiver	61.45	0.76	0.44	0.24

Table 5. comparison of the exergy and energy analysis

Table 6. show Exergy rates and other properties at various locations of ISCCS.

	h[i]	p[i]	s[i]	t[i]	v[i]	x[i]	e[i]
		[bar]	[kj/kg]	[c]	[m^3/kg]		
1	325.1	0.43	1.047	77.66	0.001027	0	6.50
2	325.1	15.69	1.043	77.37	0.001027		7.92
3	530.2	15.69	1.59	126			38.73
4	581.6	5	1.72	138.2	0.001078		48.87
5	581.6	5.82	1.72	138.2	0.001078		48.94
6	2839	5.82	6.95	194.7	0.3582		640.4
7	2839	5.82	6.95	194.7			640.4
8	2833	5.35	6.981	190.8			626.7
9	2433	0.43	1.04	77.66		0.911	6.50
10	581.7	79.22	1.70	137.1	0.001072		55.0
11	3363	79.22	6.68	485.4	0.04115		1252
12	3358	76.26	6.69	481.9			1244
13	318.6	1	6.92	45			14.78
14	713.1	11.36	7.04	426			373.3
15	1547	11.36	7.85	1152			948.9
16	843.6	0.92	7.93	546			219.2
17	406.4	0.92	7.19	131.9			17.35
18	581.7	5.82	1.72	138.2			48.95
19	2839	5.82	6.95	194.7			640.4

Table 6. Exergy rates and other properties at various locations of ISCCS.

Finally calculate the total first and second efficiency:

η_{isccs}=31.1

η_{isccs}=30.41

CONCLUSIONS

In this study, a comprehensive energy and exergy analysis of Integrated Solar Combined Cycle System in Yazd, Iran is conducted using design plant data, and a performance assessment of this integrated solar combined cycle system is made through energy and exergy efficiencies, exergetic improvement potential, as well as some other thermodynamic parameters. The exergy destructions n the overall ISCCS are quantified and illustrated using an exergy flow diagram along with an energy flow diagram. Concluding remarks that can be made are:

Exergy analysis would be a potential tool in determining locations, types and true magnitudes of wastes and losses, furthering the objective of more efficient energy resource utilization, and revealing whether or not and how much it is possible to design a more efficient solar thermal power plant by reducing the inefficiencies in the systems and their components

- Exergy destruction throughout the plant is quantified include: losses in combustor, collector, stack and heat exchangers, and pump & turbines are determined which accounts for 29.62, 9, 7.78 and 8% of the total exergy input to the plant, respectively.
- The values of energy and exergy efficiencies for the ISCCS are found to be 31.1% and 30.41%, respectively.
- It can be seen that the solar collectors is the part where the exergy efficiency is 24%, making it the least efficient
- Because of combustors have maximum exergy loss in system, therefore their inefficiency can be reduced by preheating
- the combustion air and reducing the air-fuel ratio, consequently the performance of the combustors and total system can be improved.
- It is found that solar heat exchangers have maximum potential for improvement the system efficiency between heat exchangers.
- Results are expected to be beneficial to the researchers and engineers working in the area of solar thermal systems to improve the overall performance and increasing solar energy contribution for electricity generation.

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