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Study on mathematical model of solar thermal parabolic

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Abstract
The conceptual design of the concentrating solar power plant (CSPP) includes selection and sizing of solar field, power generation cycle, types of working fluids, sizing of the power block, etc. The conceptual design is based on a mathematical modeling of the CSPP. In the present studies, solar thermal system, a turbine is integrated with 1.2 kW generators, and steam is produced by flow loop energized by solar parabolic trough concentrators. One of the most significant factors for the successful conceptual design of a CSPP is a precise determination of the transient solar radiation over the day and the season. Accordingly, in-situ measurements of the direct normal irradiance (DNI) has been carried out and used as input to the mathematical model.

Keywords: Study, mathematical, solar thermal, parabolic, CSPP, DNI

Introduction

Thermal performance of solar thermal power plant trough collector inside a Solar Energy Generating Systems plant was examined by Odeh *et al.* [1]. A design was presented by Hirsch along with Eck [2] of a phase separation system for DSG PTC field. A comparison was made between analysis of PTCs molten salt as heat transfer fluids, steam/water, and making use of oil. Giostri *et al.* [3] made comparison between different technologies of a solar field with regards to energy production annually as well as performance at design conditions.

In recent times, Desai *et al.* [4] presented a technique in order to find the most favorable design of radiation for concentrating solar powerplant exclusive of thermal storage as well as hybridization. The aim of present study is to examine the conceptual design and make prediction of performance of the solar thermal power plant by making use of DSG inside a parabolic trough solar field.

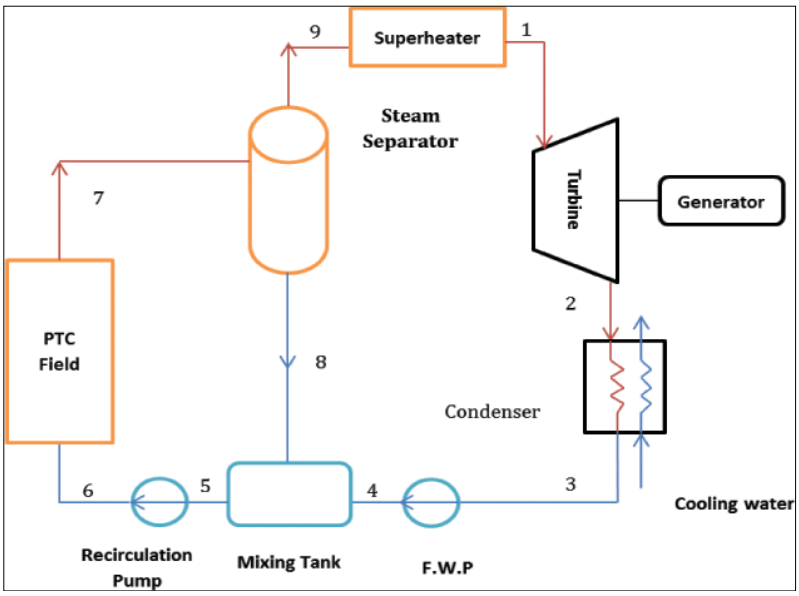


Fig 1: PTC based Solar

Formulation

The size of the solar power generation plant using a parabolic trough collector field is selected to be 1.2 kW. As shown in Figure-1 illustrates the proposed solar power plant established. Following are the specifications of the proposed system:

- The capacity of the plant is 1.2 kW.
- The composition of Collector receiver tubes includes a selective outer layer and cover glass made up of evacuated space amid to minimize the heat loss.
- The solar field is made from parabolic trough collectors which have its alignment in the direction of the axis in the north to south and having a tracking system in the direction of the axis in the east to west direction in order to the trace Incident radiation in day time.
- The steam turbine is kind of condensing with pressure of 0.1 bar.
- Solar field consists of 5 modules. Every collector has three pipes each having the size 2.135 meters in length.
- Circulation water pumps with 6 bar pressure and min having a flow rate of 0.009 kg/s.

Table-1 shows specifications along with dimensions of the collector. To calculate the heat gain for the steam generation with a total output power 1.2Kw. The steam pressure at supply is at 5.2 bars and 200 ˚C and the condenser pressure is 0.1 bar. Figure 1 displays collectors with a turbine for electricity generation and a Rankine cycle of steam generator,

Table 1: Specifications of solar collector.

Model	Symbol	High temperature solar collector pipe
Type		JJR-T-2140 collector tube
Length	-	2135 mm
Working temperature(Highest)	-	> 200˚C(350˚C)
Diameter of absorption tube	-	40 mm
Diameter of glass tube	-	100 mm
Absorptivity	α	≥ 0.94
Emittance	ε_{Cv}	≤0.10at temperature 200˚C
Transmittance of glass	τ	≥ 0.91.4
Reflectance of mirror	ρ_c	0.7
Intercept factor	γ	0.91
Incidence angle modifier	K_γ	1
Optical efficiency	η_o	0.55
Operating pressure (200˚C)		≤ 40 bar
Design life		> 15(Year)
Vacuum degree		3.0×10^{-4}Pa

Mathematical Modeling

The mathematical model of the system is proposed in this particular system. The analysis of energy of parabolic through solar collector in this segment is laying its foundation on the equations stated in the references [5, 6] The first thing proposed is model of solar system. After that the equations for performance evaluation for whole system is proposed. An assumption is made that the status of system is fixed. The variations in pressure are ignored apart from pressure in the turbines as well as pumps. The definition of vital rate of energy from the collector is given as:

$$Q_{gain} = m_{CL}.(h_7 - h_6) = \eta.I.A_p \quad (1)$$

$$\eta = \eta_0 - U_L.\left(\frac{\Delta T}{I}\right) \quad (2)$$

$$\Delta T = T_m - T_a \quad (3)$$

Here m(CL) denotes the rate of mass flow rate of steam passing from Collector unit is kg/s, I= direct normal irradiance. cosØ is direct normal incidence radiation on the reflector (W/m^2), AP denotes aperture area of the collector unit is m^2, Tm = (T_6 + T_7)/2, is the average temperature unit: ˚C, Ta is ambient temperature unit: ˚C, and UL is loss co-efficient based on aperture area unit is W/m^2.K. If the work input Feed Water Pump (F.W.P) and Recirculation Water Pump (RC.W.P) and heat losses through piping are ignored then:

$$Q_{gain} =. (h_7 - h_6 =)\ m\ .(h_1 - h_3) \quad (4)$$

From Equations. (1) and (4):

$$m.\ (h_1 - h_3) = \eta.I.Ap \quad (5)$$

It must be kept in mind that the rate of PTC inlet mass flow will normally stay firm throughout the operation, consequently, the difference of dryness factor and the average temperature will differ at the part load. The difference of temperature can be evaluated by ignoring the pump work and supposing the design point outlet dryness fraction of PTC field at point 7. By keeping inlet pressure and enthalpy at point 6 the value of PTC field inlet temperature could be found out.

$$h_6 = (1 - x_7).h_8 + x_7.h_3 \quad (6)$$

It should be known here that hi and Ti represents the temperature along with enthalpy at i-th state point. η_0 is optical efficiency which is defined as:

$$\eta_0 = \rho c\ \gamma\ \tau\ \alpha\ K\gamma \quad (7)$$

ρc, γ, τ, α and Kγ are the reflectance of the mirror, intercept factor, transmittance of the glass cover, absorbance of the receiver, and incidence angle modifier, and their values are mentioned in Table-1. The definition of aperture area is given as:

$$A_p = (w - D_{co})L \quad (8)$$

Where Dco, L, and ware collector module receiver cover outer diameter, length along with width. The definition of solar collector heat loss coefficient between receiver and ambient is given as:

$$U_L = \left[\frac{A_r}{(h_{c,ca} + h_{r,ca})} + \frac{1}{h_{r,cr}}\right]^{-1} \quad (9)$$

The definition of radiation heat coefficient between the cover and ambient is given as:

$$h_{r,} = \varepsilon_{Cv}\ \sigma\ (T_c + T_a)(T_c^2 + T_a^2) \quad (10)$$

Here, denotes emittance of the cover and Stefane-Boltzmann is represented by σ. The definition of the radiation heat coefficient between the receiver and the cover is given as:

$$h_{r,} = \frac{\sigma(T_C + T_{r,av})(T_c^{\ 2} + T_a^{\ 2})}{\frac{1-\varepsilon_r}{\varepsilon_r} + \frac{A_r}{A_C}\left(\frac{1}{\varepsilon_{Cv}} - 1\right) + \frac{1}{F_{12}}} \quad (11)$$

Here ε_r denotes emittance of the receiver and the subscript 'av' is represents average. The definition of convection heat loss coefficient between the cover and ambient is given as:

$$h_{c,} = \left(\frac{Nuk_{air}}{D_{c.o}}\right) \quad (12)$$

Where Nudenotes Nusselt number and K_{air} represents thermal conductivity of the air. The subscript r is the receiver. Calculation of temperature of the cover can be carried out by this equation:

$$Tc = \frac{h_{r,Cr}T_{r,a} + (h_{C,Ca} + h_{r,Ca})To\frac{A_C}{A_r}}{h_{r,Cr} + (h_{C,Ca} + h_{r,Ca})\frac{A_C}{A_r}} \quad (13)$$

The definition of the amount of the solar radiation that shines upon the collector, considered as heat into the system is given as:

$$Q_{solar} = m_{cl}.(h_7 - h_6) + m.(h_1 - h_9) \quad (14)$$

The performance equations of entire system are given next. The definition of power produced by the steam turbine is given as:

$$W_{st} = \dot{m}(h_1 - h_2) \quad (15)$$

Here h denotes enthalpy and subscript st refers to steam. The definition of net power of the steam Rankine cycle is given as:

$$W_{cyc} = \eta_g W_{st} - (W_{F.W.P} + W_{Rc.P.W}) \quad (16)$$

Here $W_{F.W.P}$ denotes power needed for the feed water pump and $W_{Rc.P.W}$ is power for the water recirculation pump; definition of both is given as:

$$W_{F.W.P} = \dot{m}(h_4 - h_3) = \dot{m}\vartheta\ (P_4 - P_3) \quad (17)$$
$$W_{Rc.P.W} = \dot{m}(h_6 - h_5) = \dot{m}\vartheta\ (P_6 - P_5) \quad (18)$$

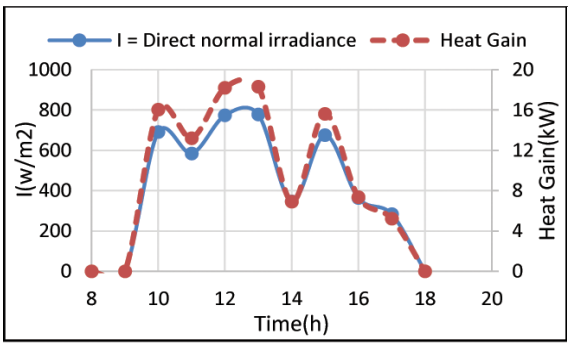
The definition of net electrical efficiency for the steam Rankine cycle system is given as:

$$\eta_{el} = \frac{W_{cyc}}{Q_{solar}} \quad (19)$$

Results and Discussions

The entire data that was measured each 10 minutes and direct normal irradiance (I) was gathered through a device situated on the solar side area of UTP. A normal day in a month is chosen to correspond to month according to (15) criterion of recommended average day of a month. It collects full solar radiation beginning at the sunrise hour and having its ending at sunset hour.

Average of heat gain per hour from PTC is shown in Figure-2 of six days in each selected month. It is noticed that I is comparatively high in November out of all months. On the other hand, the I is the least in September, among these six months. In September as compared to the month of November, the decrease observed in I is 34% resulting in a decreased the heat gain of collector field is 58%. I is because of the cosine effect which occurs in September and cloud covers causing lower collector heat gain.



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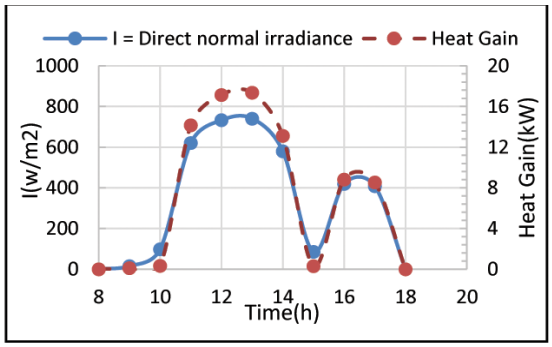

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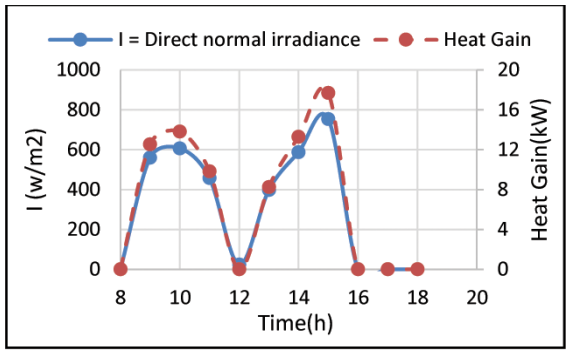

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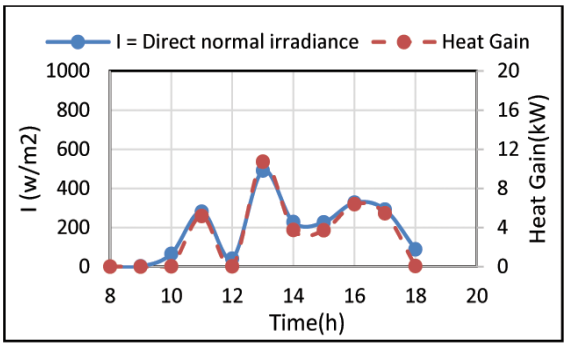

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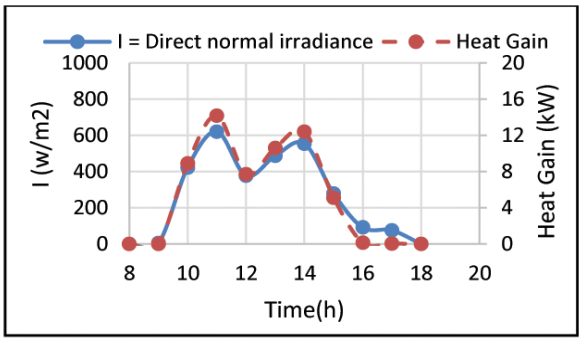

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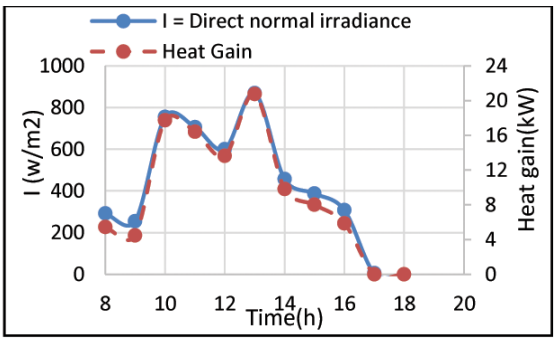

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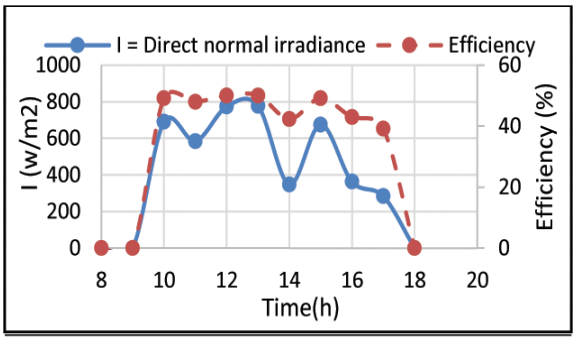
Fig 2: Average of hourly heat gain in PTC collector field

The average of efficiency per hour for PTC of the six days within the selected months is presented in Figure- 3. It could be observed that the PTC efficiency is increasing when I increases and vice versa. On the other hand, the efficiency of the PTC in September compared to November, it was noticed that the decrease in I resulting in a decreased in the efficiency of collectors by 49%.



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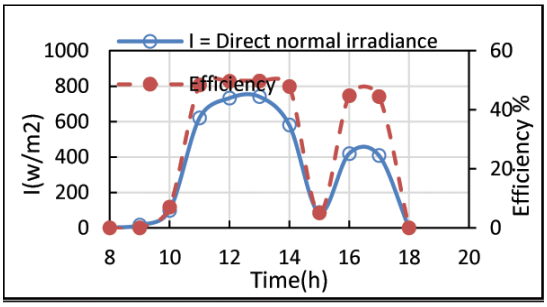

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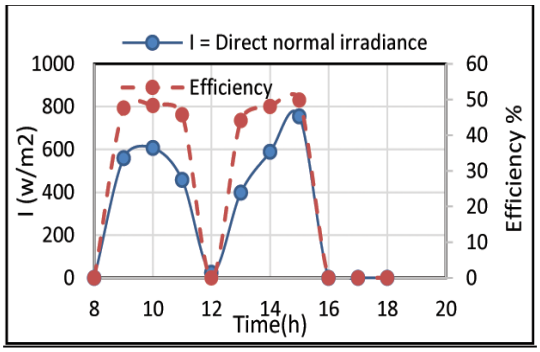

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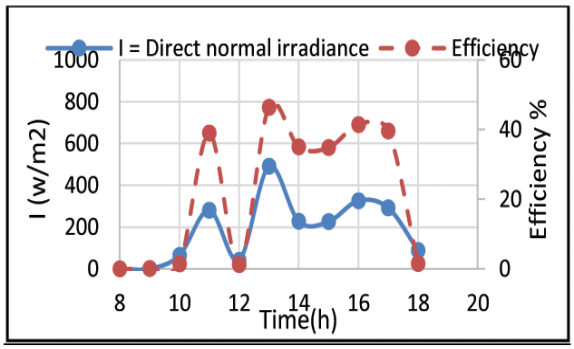

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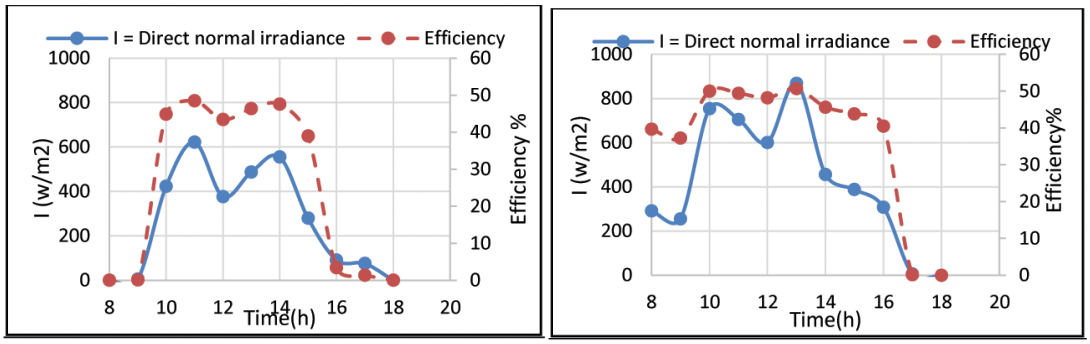

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Fig 3: Efficiency in PTC collector field.

Conclusion

A parabolic based thermal collector solar thermal power plant has been simulated mathematically for conceptual design calculations. The model was converted to a computer program within MATLAB environment, which enable prediction of the required hydrothermal parameters of the system. The control ideology of plant consists the joint effect of the solar field under ambient conditions as well as mentioned solar radiation to account for the constant generation of power from turbine-generator unit in the hours of sunshine.

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