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Analysis of mathematical model for solar thermal parabolic

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Abstract
In this paper, we study the mathematical models of solar thermal collectors and the connected heat storages. Aforementioned models, are studied in a block–oriented simulation environment to show the differences in accuracy and to study the behaviour of such models in a systems.

Keywords: Mathematical model, solar thermal, behaviour

Introduction
Recently, the thermal power plants which comprise of concentrated solar power containing generation of electric power having a range of some kilowatts to 100 mega watts [1]. Design approaches of a plain solar steam generation system which generates steam for industries whose size is ranged from small to medium. He has given essential information likestandard data on data sheets, flash steam pipes along with insulation requirements of steam's quantity, temperature, specifications along with test results from condenser units moreover manufacturer of controls, practical tools as well as pumps. An instance of the design of the solar power system was utilizedin order to generate minimal steam [2].
Thermal performance of solar thermal power plant trough collector inside a Solar Energy Generating Systems plant was examined by Odeh *et al*. [3]. A design was presented by Hirsch along with Eck [4] 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*. [5] made comparison between different technologies of a solar field with regards to energy production annually as well as performance at design conditions. A technique was proposed by Desai *et al*. [6] in order to estimate cost best design of radiation for concentrating solar power plants excluding hybridization as well as storage. In [7] examined the performance on the whole of various vapor cycles along with solar field sizing. Reddy and Kumar. [8] Analyzed how power generation for solar parabolic trough collector is done by making use of various working fluids such as water as well as oil. Arguments were presented by Aldali *et al*. [9] for the use of direct steam generation (DSG) in preference to other forms of generation in particular locations according to the prevailing environmental and economic conditions. Processes of heat transfer along with flow in a trough solar collector of DSG system were analyzed by You *et al*. [10].

Analysis
The size of the solar power generation plant using a parabolic trough collector field is selected to be 1.2 kW. Figure-1

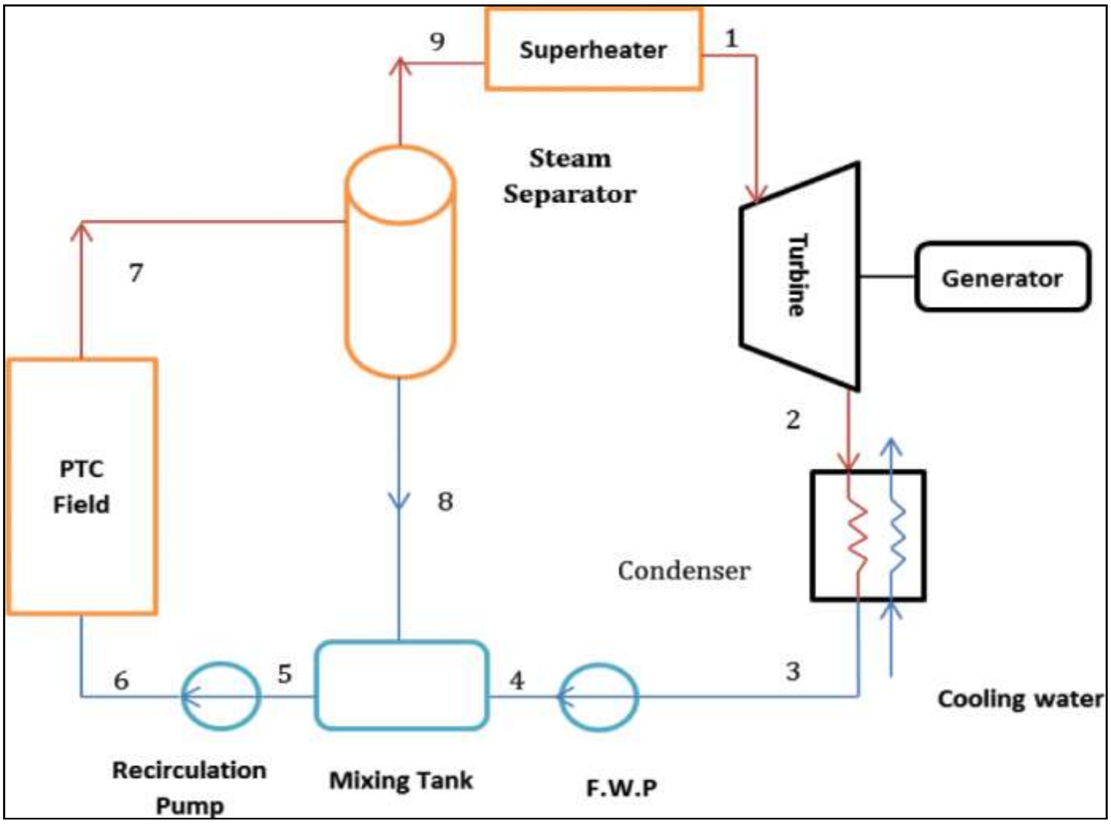

Fig 1: A simplified process flow figure of a PTC based solar thermal power plant

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 2.136 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	-	2136 mm
Working temperature(Highest)	-	> 200˚C(350˚C)
Diameter of absorption tube	-	40 mm
Diameter of glass tube	-	100 mm
Absorptivity	α	≥ 0.93
Emittance	ε_{Cv}	≤0.10 at temperature 200˚C
Transmittance of glass	τ	≥ 0.91.3
Reflectance of mirror	ρc	0.5
Intercept factor	γ	0.93
Incidence angle modifier	K_γ	1
Optical efficiency	η_0	0.55
Operating pressure (200˚C)		≤ 40 bar
Design life		> 15(Year)
Vacuum degree		3.0 × 10-4Pa

Mathematical Modeling

The mathematical model of the system is proposed in this particular system. The analysis of energy of parabolic though solar collector in this segment is laying its foundation on the equations stated in the reference [11]. 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 :

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

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

$$\Delta T = Tm - T_a \tag{3}$$

Here m_(CL) denotes the rate of mass flow rate of steam passing from Collector unit is kg/s, I= direct normal irradiance. cos∅ isdirect normal incidence radiation on the reflector(W/m^2), AP denotes aperture area of the collector unit is m^2, Tm = (T6 + T7)/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:

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

From Equations. (1) and (4):

$$m \,.\, (h_1 - h_3) = \eta \,.\, I \,.\, A_p \tag{5}$$

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.

$$h_6 = (1 - x_7) \,.\, h_8 + x_7 \,.\, h_3 \tag{6}$$

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:

$$\eta_0 = \rho_c \, \gamma \, \tau \, \alpha \, K_\gamma \tag{7}$$

ρ_c, γ, τ, α and K_γ 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:

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

where Dco, L, and w are 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:

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

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

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

Here, ε_{Cv} 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:

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

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:

$$h_{c,} = \frac{Nuk_{air}}{D_{c.o}} \tag{12}$$

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:

$$T_c = \frac{h_{r,cr}T_{r,a} + (h_{c,ca} + h_{r,ca})To\dfrac{A_C}{A_r}}{h_{r,cr} + (h_{c,ca} + h_{r,ca})\dfrac{A_c}{A_r}} \tag{13}$$

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

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

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

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

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

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

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:

$$W_{F.W.P} = \dot{m}(h_4 - h_3) = \dot{m} \, \vartheta \, (P_4 - P_3) \tag{17}$$

$$W_{Rc.P.W} = \dot{m}(h_6 - h_5) = \dot{m} \, \vartheta \, (P_6 - P_5) \tag{18}$$

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

$$\eta_{el} = \frac{W_{cyc}}{Q_{solar}} \tag{19}$$

Result & Discussion

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. The parameters taken as input of PTC field for condenser along with turbine for use of simulation are mentioned in Table-1.

Figure- 2 show the net power per hour for the steam Rankine cycle of the six day within the selected months. In September as compared to the months, the decrease observed the net power of the steam Rainkine cycle. Because of the I decreased which occurs in September and cloud covers causing lower net power of the steam Rainkine cycle.

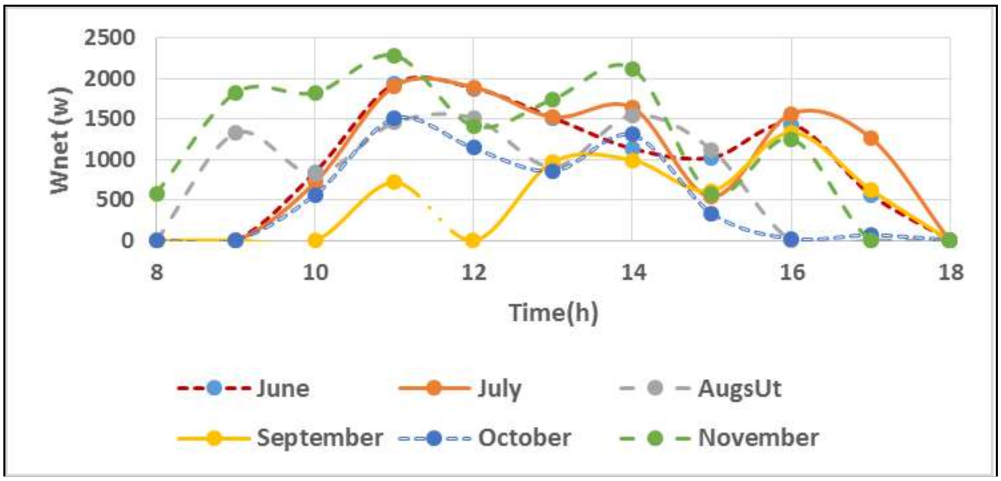

Fig 2: Power net of the steam Rankine cycle.

Conclusion

A parabolic based thermal collector solar thermal power plant has been simulated mathematically for conceptual design calculations.

References

1. Fernandez-Garcıa A, Zarza E, Valenzuela L, Perez M, Parabolic-trough solar collectors and their applications. Renew. Sustain. Energy Rev. 2010;14:1695-1721
2. Thomas A. “Design methodology for agenerationsystemusing concept small solar steam the flash boiler’’. Energy Convers. Mgmt 2016;37(I):1-15.
3. Odeh SD, Morrison GL, Behnia M. “Modelling of parabolic through direct steam generation solar collectors’’. Solar Energy 2018;62(6):395-406.
4. Eck M, Hirsch T. “Dynamics and control of parabolic trough collector loops with direct steam generation’’. Solar Energy 2007;81:268-279.
5. Giostri A, Binotti M, Astolfi M, Silva P, Macchi E, Manzolini G. “Comparison of different solar plants based on parabolic trough technology’’. Solar Energy 2012;86:1208-1221.
6. Desai NB, Kedare SB, Bandyopadhyay S. “Optimization of design radiation for concentrating solar thermal power plants without storage’’ Solar Energy 2014;107:98-112.
7. Al-Sulaiman F. Energy and sizing analyses of parabolic trough solar collector integrated with steam and binary vapor cycles’’ Energy 2013;58:561-570.
8. Reddy KS, Ravi Kumar K. “Solar collector field design and viability analysis of stand-alone parabolic trough power plants for Indian conditions’’ Energy for Sustainable Development 2012;16:456-470.
9. Aldali Y, Davison B, Muneer T, Henderson D. “ Modeling the Behavior of a 50 MW Direct Steam Generation Plant for Southern Libya Based on the Thermodynamic and Thermophysical Properties of Water Substance’’ Journal of Solar Energy Engineering 2012;134:041001-1,
10. You C, W. Zhang and Z. Yin “Modeling of fluid flow and heat transfer in a trough solar collector’’ Applied Thermal Engineering 2013;54:247-254
11. Duffie J, Beckman W. Solar engineering of thermal processes. John Wiley & Sons, Inc 2006.