Development of an innovative code for the design and Performance of Stand-alone Parabolic Trough Solar Thermal Power Plant: Code description and test case

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Abstract: This code presents an innovative code for predicting performances, as well as preliminary plant sizing, for different parabolic trough solar fields operating at nominal conditions. The conceptual design of the Stand-alone Parabolic Trough Solar Thermal Power Plant includes selection and sizing of the system components and the power generation cycle, types of working fluids, sizing of the power block, etc. The conceptual design was based on the mathematical modelling of Stand-alone Parabolic Trough Solar Thermal Power Plant. In the present solar thermal system, a turbine is integrated with 1.2 kW generators, and steam is produced through flow loop energized by solar parabolic trough concentrators. Accordingly, in-situ measurements of the direct normal irradiance (DNI).

The code allows a preliminary design of the solar field lay-out, the sizing of the main components of the plant allows to separately calculate heat loss coefficient based on aperture area (U_L), aperture effective direct normal irradiance (I) , heat gain (Q_{gain}) and the thermal efficiency of Stand-alone Parabolic Trough Solar Thermal Power Plant in commerce as well as combination of components of various commercial systems, in order to exploit different technology solutions: combination of mirrors, receivers and supports. The code is flexible in terms of heat transfer fluid, temperature and pressure range. Regarding the power block, a conventional steam cycle with super-heater.

1- Introduction

This study contains of a review of the configuration of small-scale solar parabolic trough power plant having a capacity of 1.2 kW. The configuration of the plant relies on primary working fluid water in a direct steam generation (DSG) model. The diagram of DSG based solar parabolic trough power plant configuration is displayed in Figs.1, 2. The steam is directly generated in the parabolic trough collector where heat exchangers and cost heat transfer fluid (oil) are removed. The arrangement of parabolic trough collector fields was done in North-South horizontal axis for the purpose of tracking the sun in East to West direction

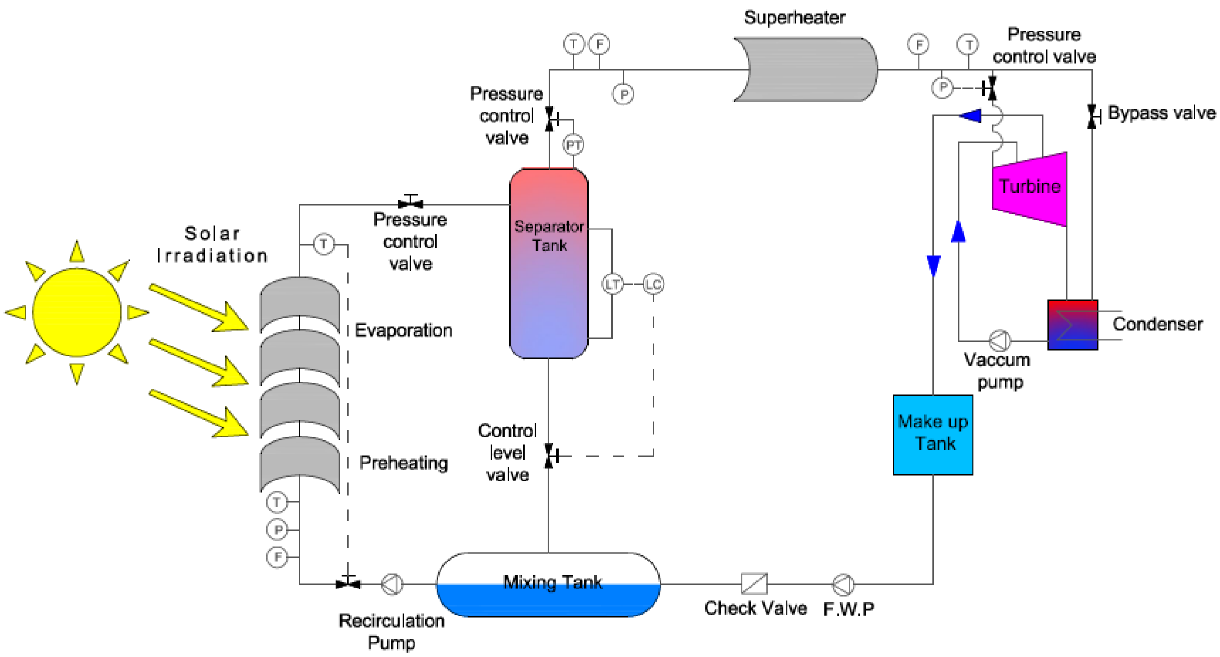


Figure 1: The proposed solar power plant established

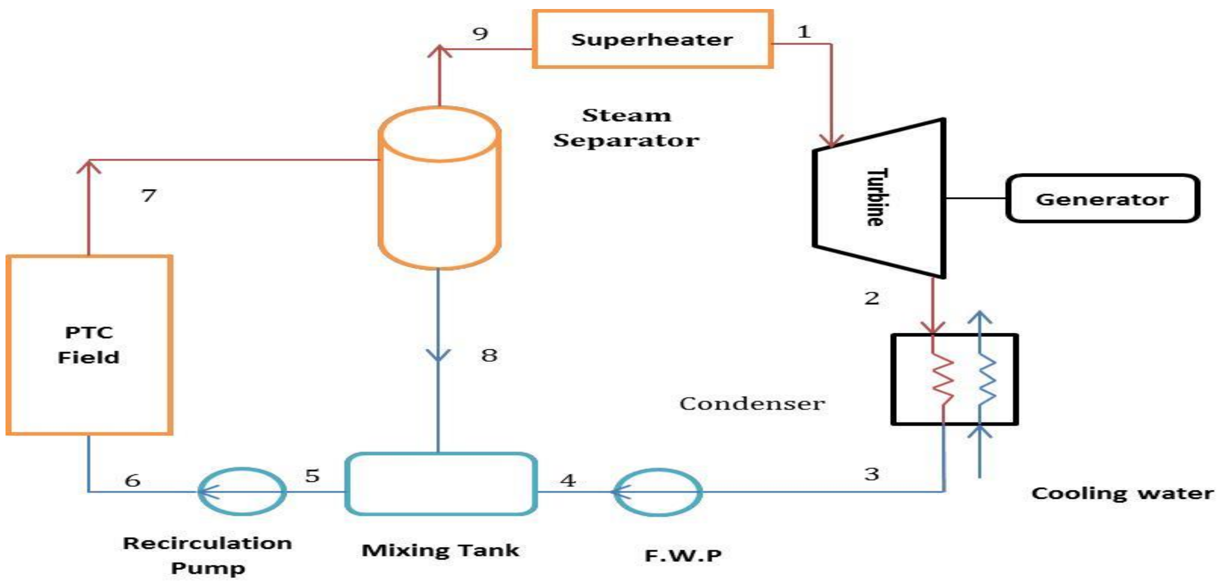


Figure 2: Simplified process flow diagram a PTC based solar thermal power plant [1]

. The solar field comprises of commercially available PTC collector modules connected parallel and series configuration. The five modules comprise of a solar collector assembly, and the typical length of a parabolic trough collector module is 6.4 m. The solar collectors' assembly was arranged in a parallel rows in the parabolic trough collector power plant. The arrangement of parabolic trough collector relies on the collector geometry (rim angle and aperture), operating conditions of the plant. The diagram of solar parabolic trough concentrator is shown in Fig. 3.

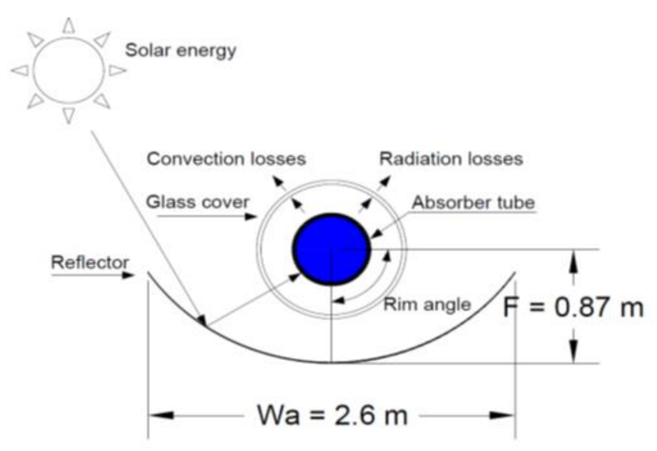


Fig. 3. Cross sectional view of the PTC.

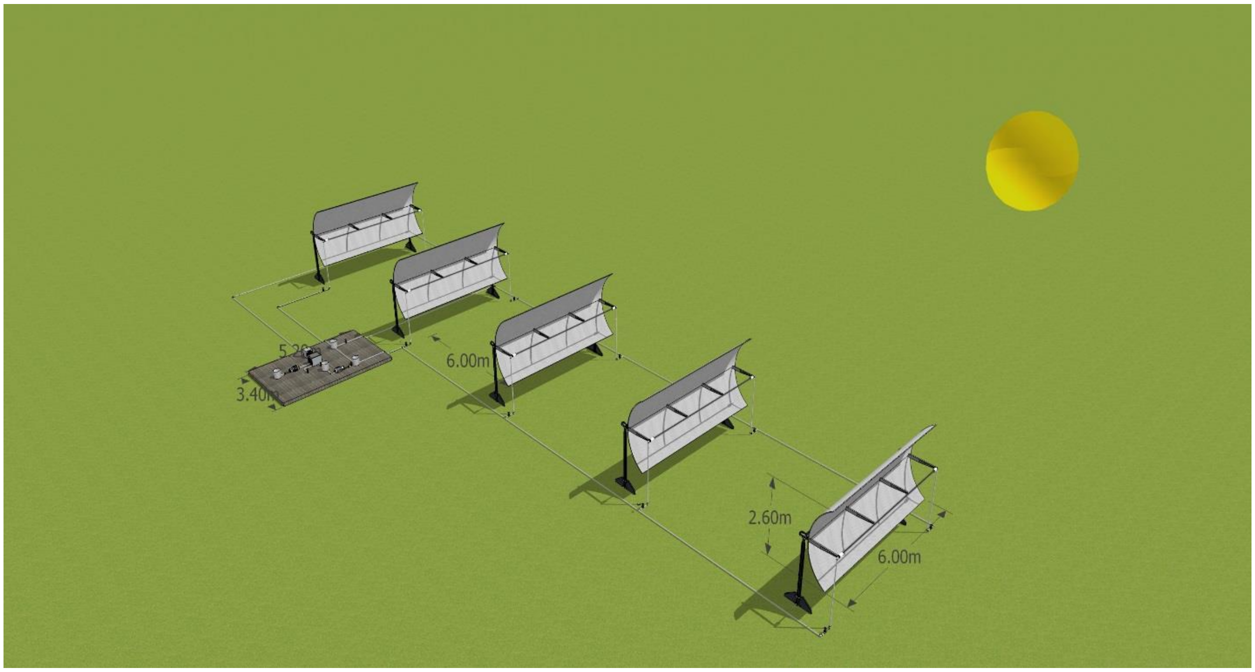
2. Model description

The proposed solar power generation plant, which uses a parabolic trough collector, was targeted to produce 1.2 kW.

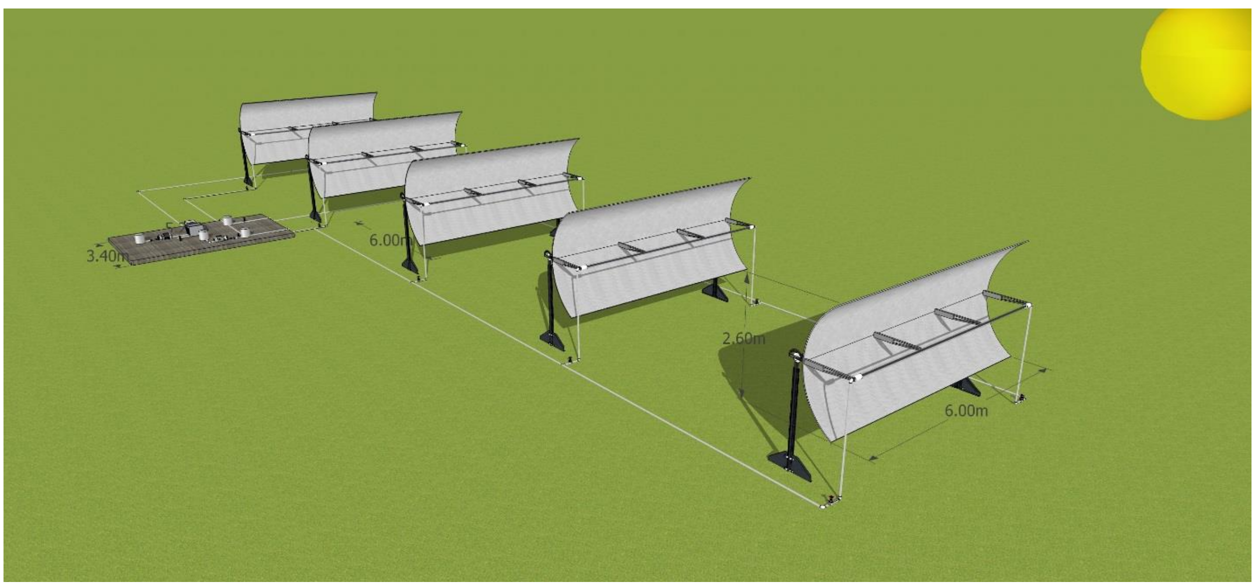
The features of the present system, are:

- Estimation of Thermal performance has been carried out on hourly base.
- The capacity of the plant is 1.2 kW.
- Shading effect of receiver tube is considered negligible inside the reflector.
- The composition of collector receiver tubes includes a selective outer layer and cover glass made up of evacuated space amid in order to minimize the heat loss.
- The effect due to the accumulation of dirt and mud on the surface of reflector and glass tube is ignored.
- 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 a tracking system in the direction of the axis in the east to west direction in order to the trace Incident radiation in the daytime.
- The steam pressure is supplied to the 1.2 kW steam turbine at 5.2 bar and 200˚C, and the condenser pressure is 0.1 bar.
- The solar field comprises of total of five trough concentrators, with specifications shown in Table 1. Every concentrator has three evacuated tubes, connected in series. Each tube is 2.136 meters in length.
- The end losses obtained from the HCE are neglected
- Circulation water pumps with maximum pressure of 6 bars pressure and 0.009 kg/s flow rate.

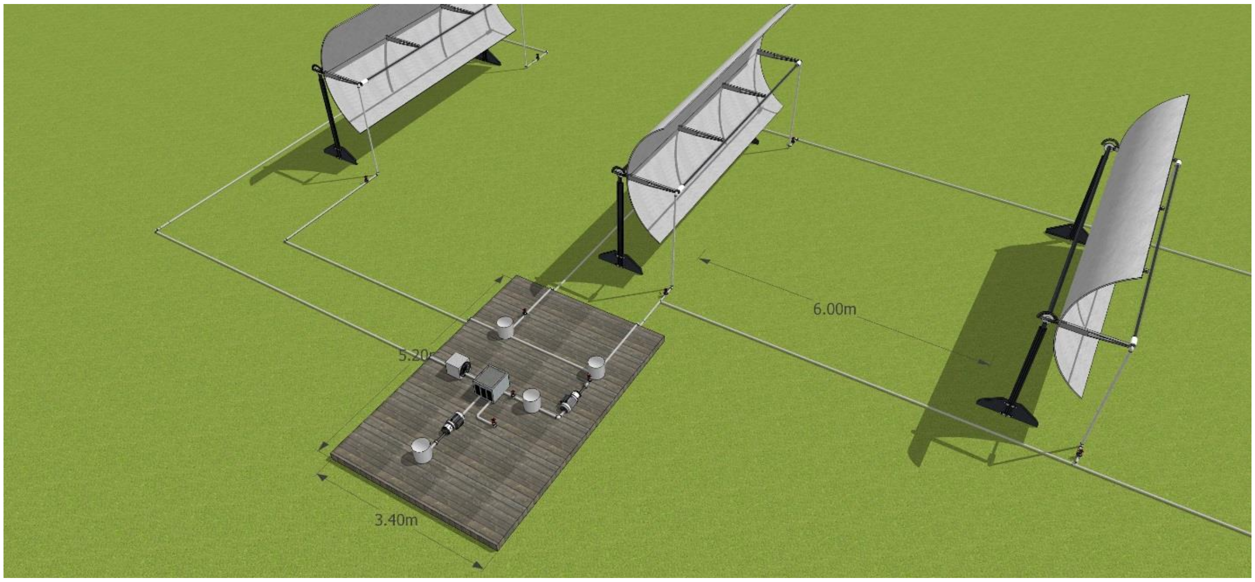
The solar parabolic trough concentrator power plant is shown in Fig. 4.



(a)



(b)



(C)

Fig 4 solar parabolic trough concentrator power plant

3- Mathematical formulation

3.1 Modelling of parabolic trough collector

The solar parabolic trough collector has been modelled to generate 1.2 kW power. The solar parabolic trough power plant contains both power block as well as solar collection/concentrating field. MATLab R2015a was utilized to design the solar parabolic trough field. Moreover, development of the analytical model was done to estimate its efficiency, area of the collector and annual power generated by the parabolic trough power plant.

3.1.1 Thermo-optical performance models

The analysis of energy of parabolic trough solar collector in this segment laid its foundation on the equations mostly stated in the literature. The mathematical procedure was first started with modeling of the solar system. After that, performance evaluation, for the entire system, was modelled. An assumption was made that the status of the system is fixed, i.e., steady state at the time of prediction. The transient behaviour was captured through the change of the solar irradiance and the weather conditions, which were supplied as input boundary conditions to the mathematical model. The variations in pressure were ignored apart from pressure in the turbines as well as pumps. The definition of vital rate of energy from the collector, with reference to Fig 2, is given as (Nishith et al., 2014) [2]:

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

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

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

Here, the mass flow rate of water passing from the collector is denoted by m_{CL} and its unit is kg/s, I stand for direct normal incidence radiation on the reflector in W/m^2, aperture area of the collector is denoted by A_P, whose unit is m^2, U_L denotes the loss co-efficient based on aperture area and its unit is W/m^2.K, $T_m = (T_6 + T_7)/2$, is the average temperature with unit ˚C, and T_a is the ambient temperature in ˚C 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:

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

From Eqs. (1) and (4), the following expression could be achieved:

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

It should be kept in mind that the rate of PTCs inlet mass flow normally stays firm throughout the operation. As a result, the difference of dryness factor and the average temperature will differ at the location of the load. The temperature difference could be calculated by ignoring the pump work and knowing the design point outlet dryness fraction at point 7, as in Fig. 2 of the PTCs field. Thomas 1996 [3] suggested 0.8 as a normal value of the dryness fraction.

By knowing the enthalpy at point 6, using equation 6, then, the temperature could be estimated.

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

Where, x is the dryness fraction, h is the Enthalpy. The value of PTCs field inlet temperature, T_6 could be calculated from the inlet pressure, P_6 and the inlet enthalpy, h_6.

The definition of aperture area is given as: you are indeed

$$A_p = (w - D_{co})L \quad (7)$$

Where: D_{co}, L, *and* w are the outer diameter, length and width of the receiver glass cover, respectively.

3.1.2. Optical efficiency η_o calculation

The estimation of the optical efficiency can be done as following [4-11]:

$$\eta_o = \rho_c \, \gamma \, \tau \, \alpha \, K_\theta \quad (8)$$

ρ_c is the reflectance of the mirror, γ is the intercept factor, τ denotes the transmittance of the glass cover, α denotes the absorbance of the receiver whereas K_θ stands for the angle of incidence modifier for a particular PTC system. Calculation could be performed using the equation (9) suggested by Forristall, 2003 and Nishith et al., 2015. [12 .13]:

$$K_\theta = \cos(\theta) + 0.000884(\theta) - 0.00005369(\theta)^2 \tag{9}$$

3.1.3 Calculation of overall heat-loss coefficient, U_L.

Solar collector heat loss coefficient between receiver and ambient was calculated as (Duffie, 2006) [4]:

$$U_L = \left[\frac{A_r}{(h_{c,ca}+h_{r,ca})A_C} + \frac{1}{h_{r,cr}}\right]^{-1} \tag{10}$$

Radiation heat coefficient between the cover and ambient was calculated as:

$$h_{r,ca} = \varepsilon_{Cv}\,\sigma\,(T_c + T_a)\left({T_c}^2 + {T_a}^2\right) \tag{11}$$

Here, ε_{Cv} denotes emittance of the cover and σ represents Stefane-Boltzmann constant. The definition of the radiation heat transfer coefficient between the receiver and the cover is given as:

$$h_{r,cr} = \frac{\sigma\,(T_c + T_{r,av})\left({T_c}^2+{T_a}^2\right)}{\frac{1-\varepsilon_r}{\varepsilon_r}+\frac{A_r}{A_c}\left(\frac{1}{\varepsilon_{cv}}-1\right)+\frac{1}{F_{12}}} \tag{12}$$

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

$$h_{c,ca} = \left(\frac{\mathrm{Nu}\,k_{air}}{D_{c.o}}\right) \tag{13}$$

Temperature of the cover is given by the following equation:

$$T_c = \frac{h_{r,cr}\,T_{r,a}+(h_{c,ca}+h_{r,ca})\,T_O\,\frac{A_c}{A_r}}{h_{r,cr}+(h_{c,ca}+h_{r,ca})\,\frac{A_c}{A_r}} \tag{14}$$

3.1.4 Calculation of the Power Block efficiency

The single collector efficiency is presented in equation (2). The quantity of solar radiation that shines on the solar collection system, is defined as (Al-Sulaimanin., 2013; Nishith et al., 2014; Nishith et al., 2015) [2, 13 and 14]:

$$Q_{solar} = \dot{m_{cl}}.(h_7 - h_6) + m.(h_1 - h_9) \quad (15)$$

Power, that steam turbine produces, is defined as:

$$W_{st} = \dot{m}\,(h_1 - h_2) \quad (16)$$

Here h denotes enthalpy and subscript $_{st}$ refers to steam. Steam Rankine cycle's net power with its calculation is shown below:

$$W_{net} = \eta_g\, W_{st} - (W_{F.W.P} + W_{Rc.P.W}) \quad (17)$$

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:

$$W_{F.W.P} = \dot{m}\,(h_4 - h_3) \quad (18)$$

$$W_{Rc.P.W} = \dot{m}\,(h_6 - h_5) \quad (19)$$

Efficiency of generator, solar collector field and turbine are important constraints on which the efficiency of the parabolic trough power plant, η_p depends upon. The parabolic trough power plant's efficiency is predicted following the definition of Reddy and Kumar, 2012 [17]:

$$\eta_P = \frac{Q_{solar}}{SM.I.L.W}.\eta_T.\eta_g \quad (20)$$

Description of the solar multiple (SM) is defined as the ratio of the thermal power generated through the solar field that is located at the point of design of the thermal power needed by the power block keeping the nominal conditions. In order to keep away from load working conditions of the power block throughout the extended cloudy weather and the times when there is no insolation, selection of SM was done for the purpose of designing the solar collector field, selection was made by adopting the recommendations of Montes et al., 2009; and Adel, 2010 [15, 16], has a value of 1.16. The amount of solar multiple must be more than unity to attain nominal conditions of the power block for a long duration.

3.2 – Modelling of solar parabolic trough collector field

For approximating the performance, choosing a place with desired weather condition is very important. Several tryouts were executed before selecting a certain year, month or a day. The inputs for performance assessment on a parabolic power plant present meteorological data for the solar radiant energy as per monthly average hourly global diffused solar radiations (insolations) along with speed of the wind and air temperatures at Universiti Teknologi Petronas. According to Adel, 2010 and Klein, 1977 [16,17], long-term performance, as well as actions of the system can be examined through hourly direct normal beam insolation from existing meteorological inputs on a certain day of every month.

3.2.1. Angle of incidence, θ

Incidence angle is the angle that lies between the beam radiation on a surface and a line that is normal to the collector rapture plane, as shown in Fig 5.

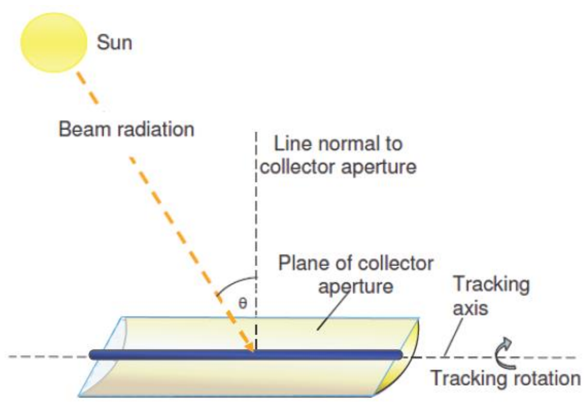


Fig.5 - Angle of incidence on a parabolic trough collector.

This angle varies over the course of the day and the outcomes obtained from the relationship among the location of the sun in the sky and the orientation of the collectors at a certain point. It is important to have identify some angles to verify the position of the sun.

Among these angles is declination angle. This angle is known to have an angular position of the sun at solar noon according to the plane of the equator. When the rotation of the Earth takes place around the Sun in a year, the declination angle varies in the range of $-23.45^{o} \leq \delta \leq 23.45^{o}$.

The declination angle is calculated as:

$$\delta = 23.45 \sin\left[360\ \frac{284+n}{365}\right] \tag{22}$$

Here n stands for day number of the year, with January is 1st. In addition to this, the position of the sun relies on the angle of hour. Hour angle is the angle that lies between the local meridian and the plane that contains the centre of the Sun. When the Sun is in line with the local meridian the angle of hour has the value equal to zero. This angle arises due to the rotation of the Earth, which spins on its axis at a rate of 15° per hour and has the unit degree.

$$\omega = 15\ (t_s - 12) \tag{23}$$

t_s stands for solar time given in hours.

$$t_s = L_{st} - 4(\psi_{stL} - \psi_{locL}) + E \tag{24}$$

Here ψ_{stL} stands for the standard time longitude, ψ_{locL} denotes the location longitude, and E stands for time correction equation for any day of year described by (Duffie and Beckman., 2006) [4]:

The equation of time is defined as the difference between the mean solar time and real solar time. Below is the equation of time that was used:

$$E = 229.2[0.000075 + 0.001868 \cos\beta - 0.032077 \sin\beta - 0.014615 \cos 2\beta - 0.04089 \sin 2\beta] \tag{25}$$

Where: $\beta = \frac{360}{365}\ (n-1)$ (26)

Lastly, Zenith angle is needed to verify the location of the Sun. It is the complementary angle of solar altitude angle. It can be defined as the angle that lies between the line of sight to the Sun and vertical. It is linked to the hour angle as well as the declination angle through the relationship mentioned below (Duffie and Beckman, 2006; Nishith et al, 2015) [4, 13]:

$$\cos\theta_z = \cos\delta \cos\emptyset \cos\omega + \sin\delta \sin\emptyset \tag{27}$$

Where, $\emptyset$ is the latitude of the place.

Calculation of angle of incidence can be done when the value of zenith angle, declination angle along with hour angle is given. The angle of incidence for a plane that is rotated about a horizontal north-south axis having a constant east-west tracking that reduces the incidence angle can be calculated as:

$$\text{Cos}\ \theta = \sqrt{cos^2\theta_z + cos^2\delta\ sin^2\omega} \tag{28}$$

3.2.2. Direct Normal beam Insolation (I_b)

The amount of solar radiation that is not scattered and has been absorbed by the environment, arriving at the surface of the Earth is represented by the direct normal irradiance. The term normal here mentions the direct radiation recorded on a plane normal to its direction. The design of direct normal irradiance, DNI is considered as average of maximum DNI and average DNI after 2 h of sunshine and before 2 h of sunset for a given location [17]. Furthermore, the global horizontal irradiance (GI), falling on the horizontal plane is calculated by adding the diffuse horizontal irradiance (DI) and the direct normal irradiance (DNI) together .

$$\text{GHI} = \text{DNI. Cos}\ \theta_z + \text{DHI} \tag{29}$$

Calculation of the aperture effective direct normal irradiance falling perpendicularly on the plane is done as:

$$I = \text{DNI} \,.\, \cos\theta \tag{30}$$

4. Case Study of Analytical Simulation.

Development of the Simulink hourly insolation model along with performance model that is grounded on the linked constraints and expressions was done by Matlab R2015a, and it was implemented in the specific PTC system. First of all, an analytical model was developed in order to look for direct normal beam hourly insolation on an aperture plane. The parameters taken as inputs for the purpose of evaluating the direct normal beam insolation were: days in a year, the position of the latitude and the longitude. Moreover, the inputs required to evaluate the performance of the model are air temperature from the environment, wind speed, features of PTC, the temperature of water at the inlet and the direct normal beam hourly insolation. Use of equations was made to calculate the performance of the collectors. A procedure, known as trial-and-error, was implemented to calculate the energy loss; this procedure provides the result very quickly as mentioned in [18]. Fig. 6 displays the algorithm for analytical simulation.

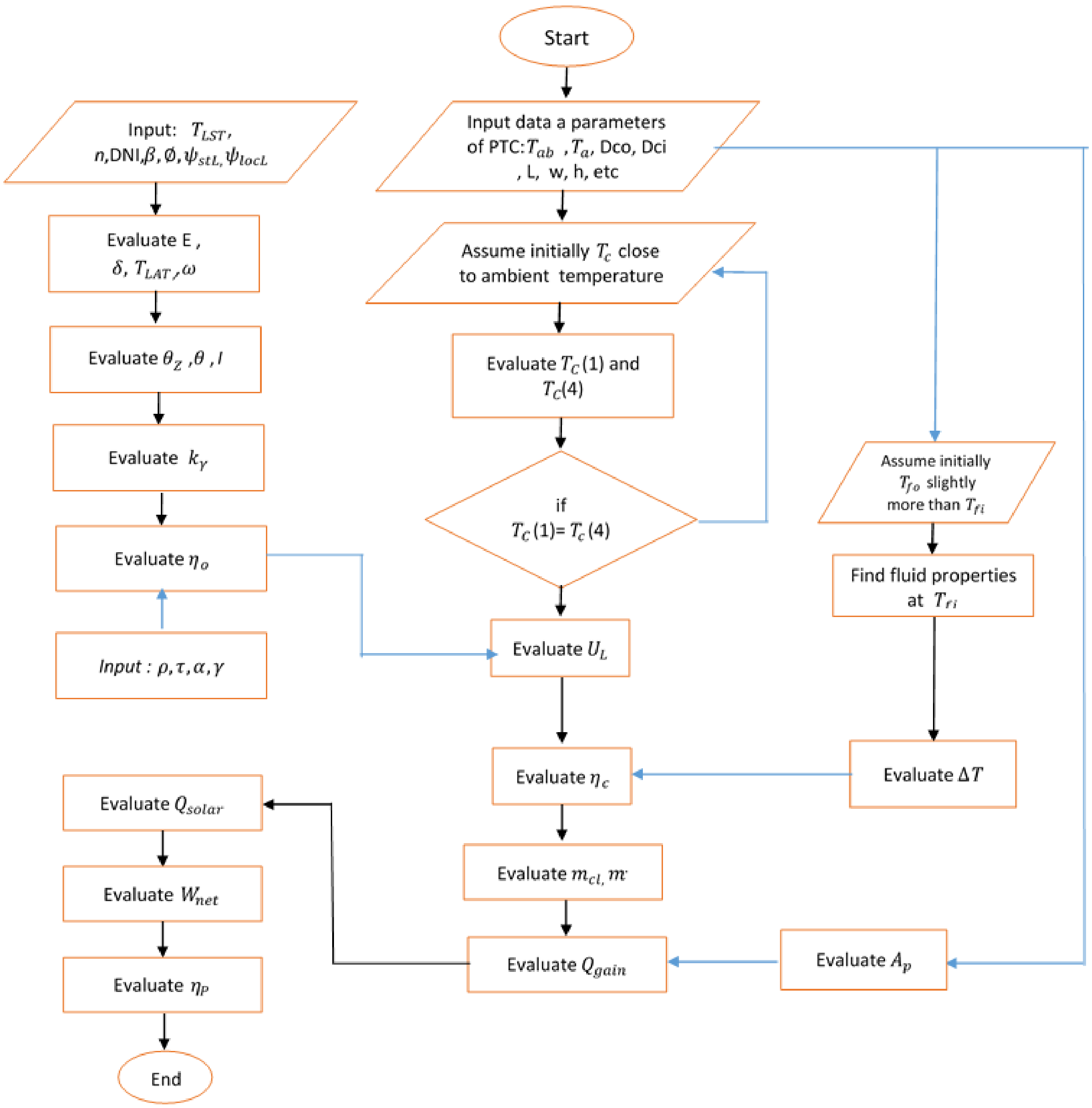


Fig.6. Algorithm for analytical simulation

5- Conclusion

The model was converted into a computer program within MATLAB environment, which enabled the prediction of the required hydrothermal parameters of the system. The control ideology of plant contains 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. The present model is found to be suitable for estimating heat loss coefficient based on aperture area , aperture effective direct normal irradiance (I) , heat gain and the thermal efficiency of Stand-alone Parabolic Trough Solar Thermal Power Plant under different operating conditions.

Appendix

State number	T (K)	P (kPa)	h (kJ/kg)
1	200	520	2854.76
2	45.8	10	2225.095
3	45.8	101.3	191.81
4	45.8	400	192.22
5	50	500	192.22
6	50	600	284.578
7	155	550	655.77
8	154	540	655.77
9	154	530	2751

State 1

S.h steam at P_1 = 5.2 bar , T_1= 200 ℃

h_1 **= 2854.76 kj/kg**

s_1 **= 7.04246 kj/kg.k**

State 2

P_2 **= 0.1 bar = 10 kpa ,** T_2 **= 45.8 ℃**

h_f **= 191.81 kj/kg ,** $s_1 = s_2$ **= 7.04246 kj/kg**

$X_2 = \frac{s_2 - s_f}{s_{fg}} = 0.85$

$h_2 = h_f + X_2\, h_{fg}$

= 191.81 + 0.85 (2392.1) = 2225.095 kj/kg

State 3

T_3 = 45.8℃ , P_3 = 1.013 bar

h_3= h_f = 191.81 kj/kg

State 4

P_4 = 4 bar , T_4 = 50 ℃

W_{pumpin}= v_3(p_4 -p_3)

= 0.001043(5 -1.0130)*100

= 0.4158441 kj/ kg

h_4 = h_3 + W_{pumpin}

= 191.81 + 0.4158441

h_4 = 192.22 kj/kg

State 5

P_5 = 5 bar , T_5= 50

h_5 = 192.22kj/kg

State 6

P_6 = 6 bar , T_6 = 154℃

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

h_6 = 284.578

State 7

P_7 = 5.5 bar , T_5= 155℃

h_f = 655.77 kj/kg , Assume X= 0.8

h_g = h_f + x h_{fg}

= 655.77 + 0.8 (2096.6) = 2333.05 kj/kg

State $\acute{8}$

$P_{\acute{8}}$ **= 5.3 bar , $T_{\acute{8}}$ = 154 ℃**

$h_{\acute{8}}$ **= 655.65 kj/kg**

State 9

P_{98} **= 5.3 bar , T_9 = 154 ℃**

h_9 **= 2751 kj/kg**

Amount of Steam Separated (X)

$h_s = h_w + Xh_{fg}$

670.56 = 655.65 + X (2101.4)

X= 0.0074= 0.74%

……………………………………………..

Flow rate

Rate of steam generation = 0.005 kg/s

Total flow to the steam separater

$\dot{m}_{fg} * h_{fg} = \dot{m}_s * h_s + \dot{m}_f * h_f$

$\dot{m}_{fg} = \dot{m}_s + \dot{m}_f$

$(\dot{m}_s + \dot{m}_f) * h_{fg} = \dot{m}_s * h_s + \dot{m}_f * h_f$

$(0.005 + \dot{m}_f) * 2101.4 = 0.005 * 2751 + \dot{m}_f * 655.65$

$\dot{m}_f$ **= 0.00224 kg/s**

$\dot{m}_{fg}$ **= 0.005 + 0.00224**

= 0.0072 kg/s

Rate of circulation water flow = 0.00224 kg/s

Nomenclature

UL: Heat loss coefficient based on aperture area (W/ (m^2. K))

$h_{r,ca}$: Radiation heat coefficient between the cover and ambient W/m^2 C.

$h_{r,cr}$: Radiation heat transfer coefficient between the receiver and the cover W/m^2 C.

$h_{c,ca}$: Heat loss coefficient between the cover and ambient W/m^2 C.

Tc : Temperature of the cover (K)

Ta: Temperature of ambient (K)

Ts : Temperature of sky (K)

Tc1: Temperature of receiver tube (K)

roh : density (kg/m^3)

v : velocity (m/s)

Dco : Diameter of cover(m)

Mu: viscosity ((N s/m^2)

kair : conductivity of air (kW/m)

segma : Stefane Boltzman constanat (kW/m^2 K^4)

epscv : emittance of the cover

Trav : Temperature of the receiver (K)

F12: collector efficiency factor

Epsr: emittance of the receiver

Dr : Diameter of receiver (m)

Nu: Nusselt number

Re: Reynolds number

hrca : The radiation heat coefficient between ambient and the cover (kW/m^2 K)

hrrc: The radiation heat coefficient between the receiver and the cover (kW/m2 K)

lati: Latitude angle(°)

n: Day of year

Lloc: Longitude angle (°)

Lst: Standard meridian

DNI: Direct normal raidation (W/m^2)

ST: Standard time (h)

IND: Aperture effective direct normal irradiance (W/m^2)

S: Declination angle (°)

E: The equation of time

w: Hour angle(°)

z: Zenith angle(°)

x: Angle of incidence (°)

IAM: Incidence angle modifier

Qg1: The collector field heat gain (W)

Ta: Ambient temperature (℃)

Relectancem; Reflectance of the mirror

Infacor: Intercept factor

Tranmg: transmittance of the glass

Absorbr : absorbance of the receiver

IAM: Incidence angle modifier

T6: Temperature inlet filed collector at point 6

T7: Temperature outlet filed collector at point 7

W: Collector width (m)

Dco: Diameter cover (m)

L: Collector length (m)

X7: Dryness at point 7

h1: Enthalpy at point 1

h2: Enthalpy at point 2

h3: Enthalpy at point 3

h4: Enthalpy at point 4

h5: Enthalpy at point 5

h6: Enthalpy at point 6

h7: Enthalpy at point 7

h8: Enthalpy at point 8

h9: Enthalpy at point 9

Tm = (T6+T7)/2

DT = Tm-Ta

Tm1 = (T9+T1)/2

DT1 = Tm1-Ta

Effo: Optical efficiency %

Effc: Thermal efficiency of collector %

Effc1: Thermal efficiency of superheater collector %

AP: Aperture area of collectors (m^2)

AP1: Aperture area of superheater collector (m^2)

MCL: The mass flow rate of steam passing through collector (kg/s)

M: The mass flow rate of steam through superheatercollector (kg/s)

Effg: Efficiency of generater

Efft: Efficiency of turbine

Qsolar: The amount of the solar radiation that shines upon the collector,considered as heat into the system (w)

Wst: Power from the steam turbine (W)

Wnet = Net power from the steam Rankine cycle(W)

SM : Solar multiple

EFFCP: The efficiency of the parabolic trough power plant %

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