Design and Analysis for 1MWe Parabolic Trough Solar Collector plant based on DSG method

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Abstract

Our objective to find the number of international standard size collector (ET-100) for designed output 1MWe parabolic trough solar thermal power plant based on DSG method. For evaluation of this designed output have done thermal analysis of steam cycle. In steam analysis, we have found steam flow rate, inlet temperature of collector, heat required per unit length for this design output and thermal efficiency of steam cycle. Further we have done optical and thermal analysis of PTSC. For, New Delhi locality took average annual air speed and solar irradiance then, we shall evaluate overall heat loss coefficient for receiver tube, correction glass cover temperature and inside tube fluid heat transfer coefficient. Further, compute collector efficiency factor, heat removal factor and total area of collector. For geometric analysis of parabolic mirror, EuroTrough-100 collector data to reckon the rim radius, focal distance, parabola curve length, geometric ratio, intercept factor, geometric factor and optical efficiency. Next, we have found collector efficiency and overall collector efficiency. We have also calculated percentage loss of heat from system and utilization of heat for power generation. Finally we have spied the number of collector for 1MWe electric output. We have designed a practical arrangement of collectors with a suitable diagram.

Keywords:-PTSC, DSG, HCE, optical & collector efficiency, receiver tube, geometric ratio, intercept factor heat removal factor, collector efficiency factor etc.

1. Introduction

The most serious challenges are scarcity of conventional energy resources in India & also world countries. The solar thermal systems play a vital role to deliver the non-polluting energy for domestic and industrial application. India has an ambitious target for the share of renewable energies in the national energy mix.

In this literature review the thorough study of research papers on the parabolic trough solar collector is done. Ari Rabl [1] matched a variety of different solar concentrators in terms of their most important general characteristics namely Concentration, acceptance angle, sensitivity to mirror errors, size of reflector area and average number of reflections. The connection between concentration, acceptance angle and operating temperature of a solar collector is analyzed in simple intuitive terms for designing for designing collectors with maximum concentration. M J Brookes et al. [2] explained the effect of the solar multiple on the annual performance of parabolic trough solar thermal power plants with direct steam generation (DSG). It has comprehend that number of collector will be solar field and also thermal storage. It is pointed out that the role of DSG with natural gas plant provides good outcome such as reduce cost of generation. Ming Qu et al. [3] calculated single dimensional heat transfer such radiative, convective, conductive and mass and energy that is solved by engineering equation solver. It plotted number of graph between operating parameters. It makes the program for PTC tracking to provide right focus on evacuated transparent glass receiver. Scott A. Jones et al. [4] created model of 30MWe SEGS VI in the TRNSYS simulation software to understand the behavior in respect of operating conditions. It anticipates effect on model in adequate environment condition. This software has capability to perform details analysis by which model could be improved Isabel Llorente Garcia et al. [5] explained the performance of parabolic trough solar thermal power plant with help of simulation model. The model is to anticipate the electric output during the various stages of planning, design, operation and construction. This model compares result to real data (50MWe operated by ACS Industrial group of Spain). Doing this comparison we can decide the mass flow rate of HTF. Balbir Singh Mahinder Singh et al. [6] focus active involvement of PTC in Malaysia. The importance due to scarcity of fossil fuel is used. It designed PTC by simulation software to the selection of certain parameter such the aperture area and obtained the geometric concentration ratio by receiver diameter. To evaluate the optical precision for thermal performance of CPTC, to the thermal losses reduced by aperture area. V.Siva Reddy et al. [7] did exergetic analysis of PTCSTPP to improve the performance of the plant to reduce the loss of the components to optimize the maximum efficiency. Land areas required for 50MWe for the location of Jodhpur and Delhi to increase exergetic efficiency from 23.66% to 24.32%. It also found that the exergetic and exergetic efficiency of PTC. Iman Niknia et al. [8] perform a transient simulation to integrating a new PTC collector with oil cycle and an auxiliary boiler. For analysis, a computer code is developed and experiments are performed to validate the simulation program. Based on the selected conditions, annual power generation of solar part and fossil section are determined and compared with fossil fuel plant. Comparison of the new system with previous arrangement illustrates that various integration schemes can be easily simulated and an appropriate system to satisfy the main design objectives can be chosen. Iman Niknia et al. [9] designed for 250 kW Shiraz solar thermal power plant power to promote the field of collectors by installing a large parabolic collector and combining the system with a 500 kW hybrid boiler. This hybrid plant performance is evaluated by simulation software to predict outcomes at the operating working condition. Due to this capability, this provides best strategies to control the operation. Hank Price et al. [10] reviewed the current state of the art of parabolic trough solar power technology and described the R&D efforts that are in progress to enhance this technology. The paper also shows how the economics of future parabolic trough solar power plants are expected to improve. The operating performance of the existing parabolic trough power plants has demonstrated this technology to be robust and an excellent performer in the commercial power industry and since the last commercial parabolic trough plant was built, substantial technological progress has been realized. The various alternative technologies are given for the tracking mechanisms, reflector materials, heat collection elements thermal characteristics, heat transfer fluids and power cycle to reduce the cost of the plant. Parabolic trough solar power technology appears to be capable of competing directly with conventional fossil-fuel power plants in mainstream markets in the relatively near term. Given that parabolic trough technology utilizes standard industrial manufacturing processes, materials, and power cycle equipment, the technology is poised for rapid deployment should the need emerge for a low-cost solar power option

S.K. Tyagi et al. [11] evaluated the exergetic performance of concentrating type solar collector and the parametric study is made using hourly solar radiation from the exergy output is optimized with respect to the inlet fluid temperature and the corresponding efficiencies are computed. R.Lugo-Leyte et al. [12] suggested to preventing the deflection due to long pipe/tube & high temperature. It has provided the composition of receiver tube material as copper (20%) and steel (80%). according this compound pipe is 75% less than gradient of the simple pipe in a time of ten second. Compound absorber pipe offer greater resistance to the deflection provoked by the direct steam generation. Amirtham Valan Arasu et al. [13] investigated the performance of a new parabolic trough collector hot water generation system with a well-mixed hot water storage tank. The storage tank water temperature is increased from 35ºC at 9.30 h to 73.84ºC at 16.00 h when no energy is withdrawn from the storage tank. The average beam radiation during the collection period is 699 W/m^2. The useful heat gain, collector instantaneous efficiency, energy gained by the storage tank water and the efficiency of the system as a whole are found to follow the variation of incident beam radiation as these parameters are strongly influenced by the incident beam radiation. The values of each of those parameters are observed maximum at noon. Soteris A. Kalogirou et al. [13] presented a parabolic trough solar collector system used for steam generation. A Modelling program called as PTCDES which is written in BASIC language is developed for determining the quantity of steam produced by the steam generation system. The flash vessel size, capacity and inventory determine how much energy is used at the beginning of the day for raising the temperature of the circulating water to saturation temperature before effective steam production begins. System performance tests indicate that the Modelling program is accurate to within 1.2% which is considered very accurate. The theoretical system energy analysis is presented in the form of Sankey diagram. The analysis shows that only 48.9% of the available solar radiation is used for steam generation. Martin Kaltschmitt et al. [15] described that solar energy has a share of more than 99.9 % of all the energy converted on earth. The solar radiation incident on the earth is weakened within the atmosphere and partially converted into other energy forms (e.g. wind, hydro power). Part of the solar radiation energy can be converted into heat by using absorbers (e.g. solar collectors). A. El Fadar et al. [21] presented a study of solar adsorption cooling machine, where the reactor is heated by a parabolic trough collector (PTC) and is coupled with a heat pipe (HP). This reactor contains a porous medium constituted of activated carbon, reacting by adsorption with ammonia. A model, based on the equilibrium equations of the refrigerant, adsorption isotherms, heat and mass transfer within the adsorbent bed and energy balance in the hybrid system components has been developed. From real climatic data, the model computes the performances of the machine. In comparison with other systems powered by flat plate or evacuated tube collectors. The numerical results show a great sensitivity of the performance coefficient of the machine to the radius of the absorber and the aperture width of collector. Ricardo Vasquez Padilla et al. [16] performed a one dimensional numerical heat transfer analysis of a PTSC. The receiver and envelope were divided into several segments and mass and energy balance were applied in each segment. Improvements either in the heat transfer correlations or radiative heat transfer analysis are presented as well. The partial differential equations were discretized and the nonlinear algebraic equations were solved simultaneously. Finally, to validate the numerical results, the model was compared with experimental data obtained from Sandia National Laboratory (SNL) and other one dimensional heat transfer models. The results showed a better agreement with experimental data compared to other models.

2. Description of PTSC plant based on DSG

A PTC is basically made up of a parabolic trough-shaped mirror that reflects direct solar radiation, concentrating it onto a receiver tube located in the focal line of the parabola. Concentration of the direct solar radiation reduces the absorber surface area with respect to the collector aperture area and thus significantly reduces the overall thermal losses. The concentrated radiation heats the fluid that circulates through the receiver tube, thus transforming the solar radiation into thermal energy in the form of the sensible heat of the fluid. The geometrical concentration ratio reaches about 20 to 100. This produced the temperature about 375^0C of receiver/absorber tube. We convey the water inside the absorber tube from given point 4 that is inlet for absorber tube. They take the heat converted into steam at required pressure and temperature as desired. This steam goes to steam turbine then expanded to produce mechanical work converted in electrical energy by generator. Exhaust steam goes to condenser which maintained the pressure below surrounding and also removed the heat to surrounding or cogeneration by external circuit of heating/cooling. Then pump suck water send it to inlet of collector at required pressure. Then inlet water gets heat then converts steam. This conversion of water into steam occurs slowly inlet to outlet of collector. [18].DSG has technical advantages that must be considered Zarza et al. [24]

- No danger of pollution or fire due to the use of thermal oil at temperatures of about 400^0C
- Opportunity to increase the maximum temperature of the Rankine cycle above 400^0C, the limit imposed by the thermal oil currently used
- Reduction in the size of the solar field, thus reducing the investment cost
- Reduction in operation and maintenance-related costs, as thermal oil-based systems require a certain amount of the oil inventory to be changed every year, as well as antifreeze protection when the air temperature is below 14^0C

3. Mathematical formulation

3.1. Analysis of Steam Cycle

- Constant pressure rejection in condenser (2-3)
- Constant pressure addition in evaporator/boiler (4-1)
- Adiabatic expansion process in steam turbine(1-2)
- Isochoric and adiabatic process in pump (3-4)

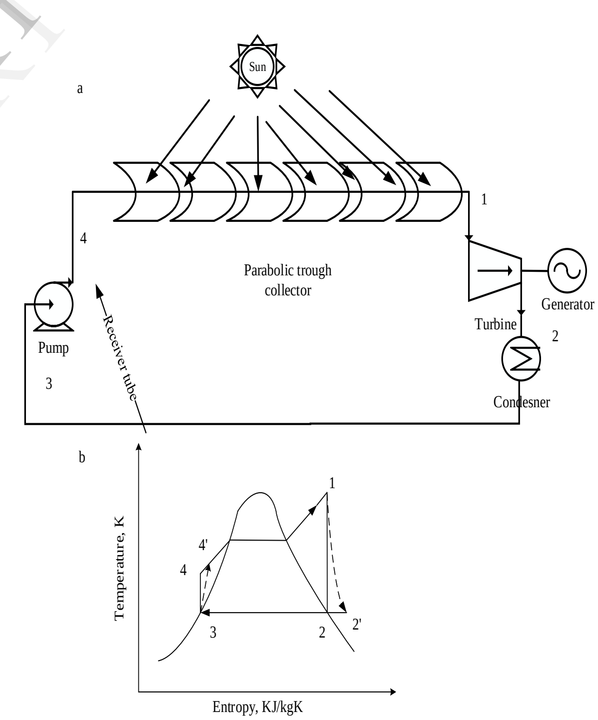

Fig.1. (a) Schematic diagram of direct steam generation with PTC as generator/receiver tube and (b) temperature versus entropy corresponding above power cycle

Table 1.Operational data for PTSC based on DSG

Net Grid power/generator power	1MWe
Inlet steam pressure	100 bar
Inlet steam temperature	375^0C
Condenser pressure	0.112 bar
Overall efficiency for electric generator and electric grid/parasitic loss	95%
Turbine shaft output	1.050 MWe
Dryness fraction at the outlet of condenser	saturated liquid =0
Steam turbine efficiency	88%
Pump efficiency	89%
Diameter of receiver /absorber tube $D_r = D_o$	0.07 m
Diameter of glass cover tube D_g	0.10 m
Inside diameter of receiver tube	0.055 m
Single module length of Euro Trough (ET-100)	12.27 m
Aperture width	5.76 m
Material for receiver tube	stainless steel
reflectance of the mirror ρ	0.94
transmittance of the glass cover τ	0.89
absorptance of the receiver α	0.94
angle of incidence θ	0°
intercept factor	0.95
Wind speed	3.03 m/s for New Delhi locality according Synergy co. India
Average annual Solar insolation	550w/m^2 for New Delhi according Synergy co. India

Important relations used for the analysis of steam Rankine cycle

Turbine efficiency

$$\eta_{ST} = \frac{h_1 - h_2'}{h_1 - h_2} \qquad (1)$$

Pump efficiency

$$\eta_{pump} = \frac{h_4 - h_3}{h_4' - h_3} \qquad (2)$$

Net power output

$$W_{net} = (h_1 - h_2') - (h_4' - h_3) \qquad (3)$$

Heat input

$$Q_{in} = (h_1 - h_4) \qquad (4)$$

Pump work

$$W_{pump} = (h_4' - h_3) = \frac{v(P_4 - P_3)}{\eta_{pump}} \qquad (5)$$

Thermal efficiency of steam cycle

$$\eta_{thermal} = \frac{W_{net}}{Q_{in}} \times 100 \qquad (6)$$

Steam flow rate $\dot{m}$ in Kg/sec

$$\dot{m} = \frac{\text{designed steam turbine shaft output}}{\text{net power output}} \qquad (7)$$

3.2. Optical analysis of parabolic trough collectors

For specular reflectors of perfect alignment, the size of the receiver (diameter D) required to intercept all the solar image can be obtained from trigonometry and Figure 2, given by Duffie et al. [17]

$$D = 2r_r \sin(\theta_m) \qquad (8)$$

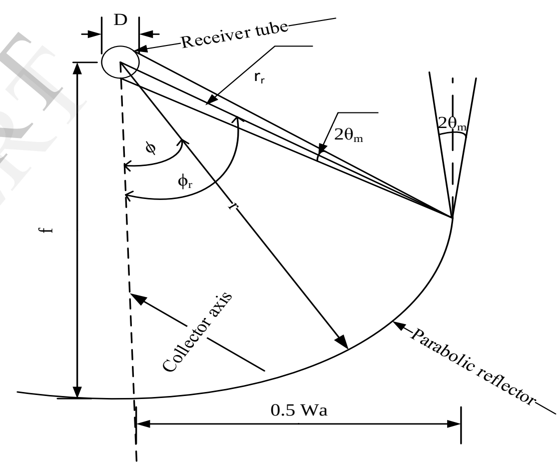

Fig. 2.cross-section of a parabolic trough collector with circular receiver

As φ varies from 0 to φ_r, r increases from f to r_r and the theoretical image size increases from 2 f sin (θ_m) to $2r_r \sin\theta_m / \cos(\varphi_r + \theta_m)$Therefore, there is an image spreading on a plane normal to the axis of the parabola

$$r_r = \frac{2f}{1+\cos\varphi_r} \qquad (9)$$

Another important parameter related to the rim angle is the aperture of the parabola, W_a

$$W_a = 2r_r \sin\varphi_r \qquad (10)$$

Equating above equation, again the aperture of the parabola

$$W_a = \frac{4f \sin \varphi_r}{1 + \cos \varphi_r} \quad (11)$$

Reduce it to like

$$W_a = 4f \tan \frac{\varphi_r}{2} \quad (12)$$

For a tubular receiver, the concentration ratio is given by

$$C = \frac{W_a}{\pi D} \quad (13)$$

By replacing D and W_a

$$C = \frac{\sin \varphi_r}{\pi \sin \theta_m} \quad (14)$$

The maximum concentration ratio occurs when φ_r is 90° and sin (φ_r) = 1. Therefore, by replacing sin (φ_r) = 1 in Eq. (14), the following maximum value can be obtained:

$$C_{max} = \frac{1}{\pi \sin \theta_m} \quad (15)$$

The curve length of the reflective surface is given by

$$S = \frac{H_p}{2}\left\{\sec\frac{\varphi_r}{2}\tan\frac{\varphi_r}{2} + \ln\left[\sec\frac{\varphi_r}{2} + \tan\frac{\varphi_r}{2}\right]\right\} \quad (16)$$

It is also shown that, for different rim angles, the focus-to-aperture ratio, which defines the curvature of the parabola, changes. It can be demonstrated that, with a 90° rim angle, the mean focus-to-reflector distance and hence the reflected beam spread is minimized, so that the slope and tracking errors are less pronounced. The collector's surface area, however, decreases as the rim angle is decreased. There is thus a temptation to use smaller rim angles because the sacrifice in optical efficiency is small, but the saving in reflective material cost is great

The amount of aperture area lost is

$$A_e = fW_a \tan \theta \left[1 + \frac{W_a^2}{48f^2}\right] \quad (17)$$

For a plate extending from rim to rim, the lost area is

$$A_b = \frac{2}{3} W_a h_p \tan \theta \quad (18)$$

Where h_p =height of parabola (m).

Therefore, to find the total loss in aperture area, A_l, the two areas, A_e and A_b, are added together without including the term tan θ. Jeter et al. [19]

$$A_l = \frac{2}{3} W_a h_p + fW_a \left[1 + \frac{W_a^2}{48f^2}\right] \quad (19)$$

$$\text{Geometry factor } A_f = \frac{A_l}{A_a} \quad (20)$$

Intercept factor

Using the universal error parameters, the formulation of the intercept factor, γ, is possible. Guven et al. [15]

$$\gamma = \frac{1 + \cos \theta_r}{2 \sin \theta_r} \int_0^{\theta_r} \mathrm{Erf}\left\{\frac{\sin\theta_r[1+\cos\theta_r][1-2d^*\sin\theta_r] - \pi\beta^*[1+\cos\theta_r]}{\sqrt{2}\pi\sigma^*[1+\cos\theta_r]}\right\} - \mathrm{Erf}\left\{-\frac{\sin\theta_r[1+\cos\theta_r][1+2d^*\sin\theta_r] + \pi\beta^*[1+\cos\theta_r]}{\sqrt{2}\pi\sigma^*[1+\cos\theta_r]}\right\} \frac{d\theta}{[1+\cos\theta_r]} \quad (21)$$

Optical efficiency is defined as the ratio of the energy absorbed by the receiver to the energy incident on the collector's aperture. The optical efficiency depends on the optical properties of the materials involved, the geometry of the collector, and the various imperfections arising from the construction of the collector. Sodha et al. [23]

$$\eta_O = \rho\tau\alpha\gamma(1 - \tan\theta\cos\theta) \quad (22)$$

3.3. Thermal analysis of parabolic trough collectors

The generalized thermal analysis of a concentrating solar collector is similar to that of a flat-plate collector. For a bare tube receiver and assuming no temperature gradients along the receiver, the loss coefficient considering convection and radiation from the surface and conduction through the support structure is given by

$$U_L = h_w + h_r + h_g \quad (23)$$

The linearized radiation coefficient can be estimated from

$$h_r = 4\sigma\varepsilon {T_r}^3 \quad (24)$$

If a single value of h_r is not acceptable due to large temperature variations along the flow direction, the collector can be divided into small segments, each with a constant h_r.

For the wind loss coefficient, the Nusselt number can be used.

For $0.1 < Re < 1000$,

$$\mathrm{Nu} = 0.4 + 0.54(\mathrm{Re})^{0.52} \quad (25)$$
$$1000 < \mathrm{Re} < 50{,}000$$

$$\mathrm{Nu} = 0.3(\mathrm{Re})^{0.6} \quad (26)$$

Estimation of the conduction losses requires knowledge of the construction of the collector, i.e., the way the receiver is supported. Usually, to reduce the heat losses, a concentric glass tube is employed around the receiver. The space between the receiver and the glass is usually evacuated, in which case the convection losses are negligible. In this case, U_L based on the receiver area A_r is given by

$$U_L = \left[\frac{A_r}{(h_w + h_{r,g-a})A_c} + \frac{1}{h_{r,r-g}}\right]^{-1} \quad (27)$$

$$h_{r,r-g} = \frac{\sigma\left(T_r^2 + T_g^2\right)\left(T_r + T_g\right)}{\left[\frac{1}{\varepsilon_r} + \frac{A_r}{A_g}\left(\frac{1}{\varepsilon_g} - 1\right)\right]} \quad (28)$$

To estimate the glass cover properties, the temperature of the glass cover, T_g, is required. This temperature is closer to the ambient temperature than the receiver temperature. Therefore, by ignoring the radiation absorbed by the cover, T_g may be obtained from an energy balance:

$$T_g = \frac{A_r h_{r,r-g} T_r + A_g\left(h_{r,g-a} + h_w\right)T_a}{A_r h_{r,r-g} + A_c\left(h_{r,g-a} + h_w\right)} \quad (29)$$

The procedure to find T_g is by iteration, estimate U_L from Eq. (27) by considering a random T_g (close to T_a). Then, if T_g obtained from Eq. (29) differs from original value, iterate. The radiation heat transfer from glass covers to air

$$h_{r,g-a} = \varepsilon_g \times \sigma \times \left(T_g + T_a\right) \times \left(T_g^2 + T_a^2\right) \quad (30)$$

Convection heat transfer, therefore based on the receiver area, the overall heat loss coefficient

$$U_O = \left[\frac{1}{U_L} + \frac{D_O}{h_{fi} D_i} + \frac{D_O \ln(D_O/D_i)}{2k}\right]^{-1} \quad (31)$$

The convective heat transfer coefficient, h_{fi} can be obtained from the standard pipe flow equation:

$$N_U = 0.023\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \quad (32)$$

$$\mathrm{Nu} = \frac{h_{fi} \times D_i}{K_f} \quad (33)$$

It should be noted that above equation is for turbulent flow (Re = 2300). For laminar flow,

Nu = 4.364 =constant

Empirical equations for the estimation of overall S.C. Mullick and Nanda have developed a semi empirical equation for directly calculating the overall heat transfer coefficient. This equation eliminates the need for an iterative calculation. Sukhatme [22]

$$\frac{1}{U_L} = \frac{1}{C_3(T_r - T_c)^{0.25} + \left[\sigma(T_r^2 + T_c^2)(T_r + T_c)/\left\{\frac{1}{\varepsilon_r} + \frac{D_O}{D_i}\left(\frac{1}{\varepsilon_c} - 1\right)\right\}\right.} + \left(\frac{D_O}{D_c}\right)\left(\frac{1}{h_w + \sigma\varepsilon_c(T_c^2 + T_a^2)(T_c + T_a)}\right) \quad (34)$$

The constant C_3 has been obtained from the correlation of Raithby and Hollands and is given by the expression

$$C_3 = \frac{17.74}{(T_r + T_c)^{0.4} D_O (D_O^{-0.75} + D_i^{-0.75})} \quad (35)$$

The cover temperature T_c is given by

$$\frac{(T_c - T_a)}{(T_r - T_a)} = 0.04075\left(\frac{D_O}{D_c}\right)^{0.4} h_w^{-0.67}\left[2 - 3\varepsilon_r + \frac{(6 + 9\varepsilon_r)T_r}{100}\right] \quad (36)$$

If $333<T_r<513$K and by

$$\frac{(T_c - T_a)}{(T_r - T_a)} = 0.04075\left(\frac{D_O}{D_c}\right)^{0.4} h_w^{-0.67}\left[2 - 3\varepsilon_r + \frac{(1 + 3\varepsilon_r)T_r}{100}\right] \quad (37)$$

If $333<T_r<623$K and by

The above equation has been developed for the following range:

$$0.1 \le \varepsilon_r \le 0.95, \quad 0.0125 \le D_O \le 0.15\,m, 15 \le h_w \le \frac{60W}{m^2K}, and\ 273 \le T_a \le 313K$$

The useful energy delivered from a concentrator is

$$Q_u = G_B\eta_O A_a - A_r U_L\left(T_g - T_a\right) \quad (38)$$

The useful energy gain per unit of collector length can be expressed in terms of the local receiver temperature, T_r as

$$q'_u = \frac{Q_U}{L} = \frac{G_B \eta_O A_a}{L} - \frac{A_r U_L (T_g - T_a)}{L} \quad (39)$$

In terms of the energy transfer to the fluid at the local fluid temperature, T_f. Solteris A.Kalogirou [20]

$$q'_u = \frac{\left(\frac{A_r}{L}\right)(T_r - T_f)}{\frac{D_O}{h_{fi} D_i} + \frac{D_O \ln(D_O/D_i)}{2k}} \quad (40)$$

Eliminating T_r

$$q'_u = F' \frac{A_a}{L}\left[G_B \eta_O - \frac{U_L}{C}(T_f - T_a)\right] \quad (41)$$

F' = the collector efficiency factor

$$F' = \frac{\frac{1}{U_L}}{\frac{1}{U_L} + \frac{D_o}{h_{fi} D_i} + \frac{D_o}{2K}\ln\frac{D_o}{D_i}} = \frac{U_O}{U_L} \quad (42)$$

The collector efficiency

$$\eta = F_R\left[\eta_O - \frac{U_L(T_i - T_a)}{G_B C}\right] \quad (43)$$

4. Results

4.1. Steam cycle result

Firstly we calculated the steam turbine shaft output by using of generator/electric grid efficiency, then calculated steam turbine output, then we used the steam cycle relation 1, 2, 3, 45, 6 & 7 to reckoned that what will be heat added to steam cycle to generate 1 MWe and also find steam flow rate for this power in kg/sec. all the important results of steam cycle are given in table 2.

Table 2.Important calculated data for steam cycle by cycle pad software

Mass flow rate in (Kg/sec)	1.11
Temperature at the inlet of PTC in ^{0}C	48.7
Steam turbine output in (MW)	1.050
Heat absorbed by receiver tube in (MW)	3.108
Heat rejected by condenser in(MW)	-2.070
Power required for pump in (KW)	-12.6
Generator & electric grid efficiency in (%)	95
η_{Carnot} in (%)	50.44
$\eta_{thermal}$ in (%)	33.38

Further we will have to analysis PTC & find collector efficiency, optical efficiency, intercept factor, how much heat collected and how much loss of heat due to three modes of heat transfer conduction, convention and radiation. How many of collectors and area of solar filed required for 1MWe.

4.2. Optical result data

We calculated further data for PTSC on basis of steam cycle data in given table 2, and the data PTSC are given in Table 1 PTSC Operating data for ET-100, and then we find optical parameter for find the optical efficiency. By using equation (8) to calculate half acceptance angle, rim radius find from equation (9), and calculate the concentration ratio from equation (14). next find the result of the curve length of the reflective surface from equation (16) .The amount of aperture area lost calculated from equation (17) and geometry factor calculated from equation (20), the formulation of the intercept factor, γ, is possible (Guven and Bannerot, 1985) from the equation (21), finally optical efficiency from equation (22). So that all the result finding above discuss, the value of results given in table 3

Table 3.Important result for optical performance

Half acceptance angle ,θ_m	0.6963
Aperture width ,H_p in meter	5.76
Focal distance, f in meter	1.44
Rim/parabolic trough radius ,r_r in meter	2.88
The length of parabola, S_p in meter	6.6113
Total area of single module, Aa in m^2	69.448
The length of single module, Lc in meter	12.057
the total loss in aperture area ,A_l in m^2	33.178
Geometry factor ,A_f	0.4777
Intercept factor , γ	0.94
Optical efficiency ,η_O in %	73.92

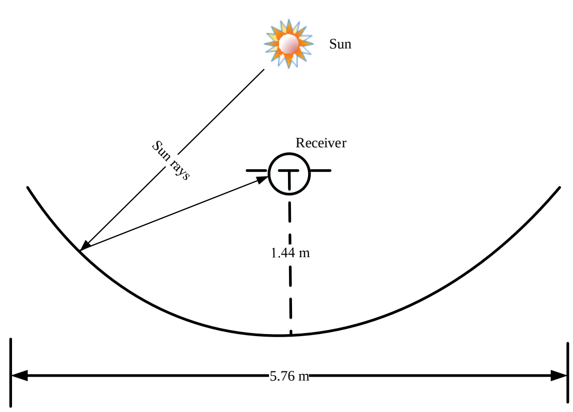

Fig. 3. Parabola geometry

4.3. Calculation of overall heat transfer coefficient for outside the receiver tube

The loss coefficient considering convection and radiation from the surface and conduction through the support structure is calculated by equation (23). This equation constituted three factors such as linearized radiation coefficient, linearized wind loss coefficient and linearized convection coefficient reckoned from equation (24), (27), and (28). Then guessing the value glass covers temperature used in equation (30) to find the overall heat transfer coefficient outside the receiver tube. Correct temperature of glass tube used the equation (29). The particular for correction temperature, we have used number of iteration and the result are given below in table 4 which calculated by MS excel software

Table 4.Calculation data for finding the correct temperature of glass cover by putting five Iterations

Parameter/Properties	1	2	3	4	5
Glass cover temperature Tg,^{0}C	60	153	162	163	163
Mean temperature T_m in ^{0}C	47.5	93.8	98.3	98.7	98.8
Density of air , ρ_{Air} in kg/m^3	1.1	1.1	1.1	1.1	1.1
μair in $\frac{Kg}{m-s}$*(10^{-5})	2.05	2.05	2.05	2.05	2.05
K_{Air} in $\frac{W}{mK}$*(10^{-2})	2.8	2.8	2.8	2.8	2.8
Reynolds number $Re_{(Air)}$ (10^4)	1.62	1.62	1.62	1.62	1.62
Nusselt number $Nu_{(Air)}$*(10^2)	1.05	1.05	1.05	1.05	1.05
Heat loss coefficient air h_w	28.1	28.1	28.1	28.1	28.1
Stefan coefficient $\sigma\,\frac{W}{m^2K^4}$*10^{-8}	5.67	5.67	5.67	5.67	5.67
The radiation heat transfer coefficient $h_{r,r\text{-}g}$ in $\frac{W}{m^2K}$	25.6	31.7	32.4	32.4	32.4
The radiation heat transfer coefficient $h_{r,g\text{-}a}$ in $\frac{W}{m^2K}$	6.58	10.1	10.5	10.5	10.5
The loss coefficient U_L in $\frac{W}{m^2K}$	16.9	20.0	20.4	20.4	20.4
Correction cover temp.T_g($^\circ$C)	153	162	163	162	163

In Above table 4, the data value will used for thermal analysis for PTSC, therefore I chosen 162.67 ^{0}C the correct temperature of glass cover and find the corresponding value of the heat loss coefficient U_L is 20.46721 $\frac{W}{m^2K}$ from table 4. This value used for further calculation for overall heat transfer coefficient, collector factor and heat removal factor etc.

4.4. Results calculation for collector heat removal factor, collector efficiency factor & overall heat transfer coefficient

The overall heat loss coefficient can be calculated by equation (31), and then we find the convective heat transfer coefficient for standard pipe flow equation (32).forward calculation different type of number by empirical equation for assessment of inside flow fluid heat transfer coefficient from equation (33), next find the collector efficiency factor from equation (42) and for the calculation of heat removal factor we will have to used equation (43). Foremost important data area of solar field used the same equation. Finally all the calculated data are shown in table 5.

Table 5.Calculation data for finding collector efficiency factor, heat flow factor and total aperture area

Steam flow rate, $\dot{m}$ in kg/sec	1.11	2	3
Outside dia. Of receiver, D_o in meter	0.07	0.042	0.042
Inside dia. Of receiver, D_i in meter	0.055	0.035	0.035
Thermal conductivity,K_{ss} in $\frac{W}{mK}$	20.2	20.2	20.2
Thermal conductivity, K_f in $\frac{W}{mK}$	6.20E-04	6.20E-04	6.20E-04
Dynamic viscosity water, μ in $\frac{Kg}{m-s}\mu$	1.28E-04	1.28E-04	1.28E-04
Constant pressure Specific heat, C_p	4.86E+00	4.86E+00	4.86E+00
Reynolds number $Re_{(Air)}$	2.01E+05	5.69E+05	8.53E+05
Prandtl number Pr	1.00E+00	1.00E+00	1.00E+00
Nusselt number, Nu	402.4919	92.54617	12800.63
Inside tube Heat transfer coefficient h_{fi} in $\frac{KW}{m^2K}$	7.129856	1.639389	226.7541
Collector efficiency factor , F'	0.987941	0.981488	0.996028
Heat removal Factor ,F_R	9.88E-01	9.81E-01	9.96E-01
Solar irradiance, S in W/m^2	5.50E+02	6.00E+02	7.00E+02
Heat absorbed by receiver tube per unit length ,Q_U/L	3.11E+06	3.11E+06	3.11E+06
Total area of collector	7740.385	5278.36	4458.1

4.5. Calculation result of collector efficiency and overall collector/solar field efficiency

For the calculation of collector efficiency uses equation (43) and we know that output of system is 1MWe, and input energy can be found by multiplication of total solar field & solar irradiance. The solar field efficiency is evaluated by dividing output of system to input of system. The results are given in table 5.

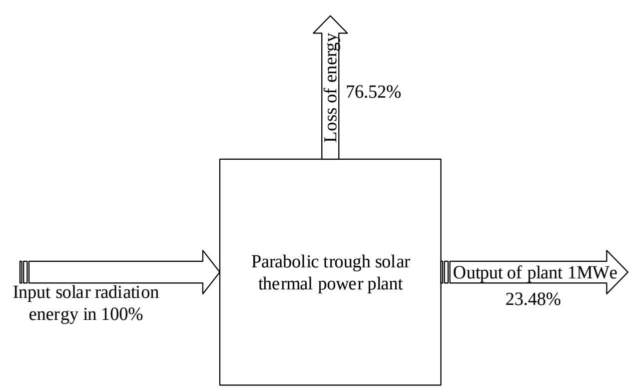

Fig.4.Percentage utilized and loss of energy to surrounding

Efficiency and energy results of collector
Input solar radiation energy = 4.257MW
Output of the plant = 1.000 MW
Total energy loss from the system = 3.257 MW
% loss of energy from the system =76.52
Collector efficiency = 71.05 %
Overall plant/collector/solar field efficiency =23.48%

4.6. Calculation for number collectors

This Euro Trough's mechanical rigidity to torsion is assured by a steel torque box of trusses and beams while the LS-3 steel structure is based on two "V-trusses" held together by Endplates. Basically, both the collector is same only difference the supporting structure and joint connection of pipe. My focus in find the number of collector, so all the data for both collectors such length of collector, width of collector, number of module per collector, and the rim angle. So I have taken ET-100 collector of data given below table and also formulation for number of collector is given in table 7.

Table 6.Find number of Euro Trough -100 collectors to bear 1MWe electric power

Overall length of a single collector (m)	98.5
Number of parabolic trough modules per collector	8
Gross length of every concentrator module (m)	12.27
Parabola width (m)	5.76
Number of ball joints between adjacent collectors	4
Net collector aperture per collector (m^2)	548.35
Total area of collector (m^2)	7740.385
Number of collector $=\frac{\text{Total area of collector } (m^2)}{\text{Net collector aperture per collector } (m^2)}$	14.11≅ 14 collectors & 1 module

We have found that for 1MWe electricity generation, we will have to use 14 collectors & 1 module. The length and width of collector, modules are given in above table.

5. Important performance graph/plot on based of above calculated data by EES & DPLOT scientific research software

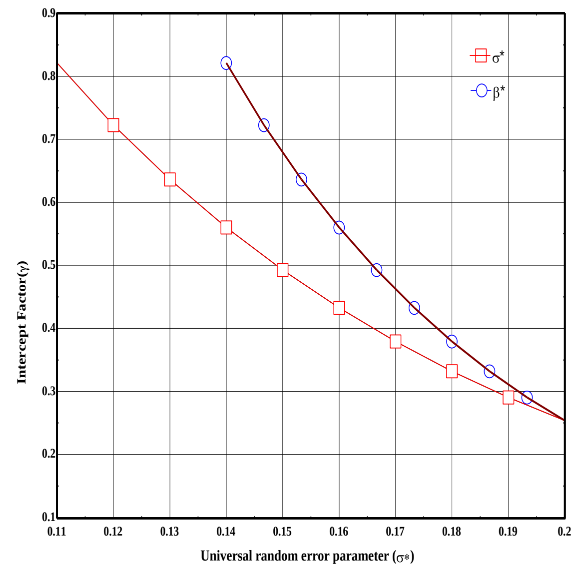

Fig.5.Intercept factor versus universal random error parameter

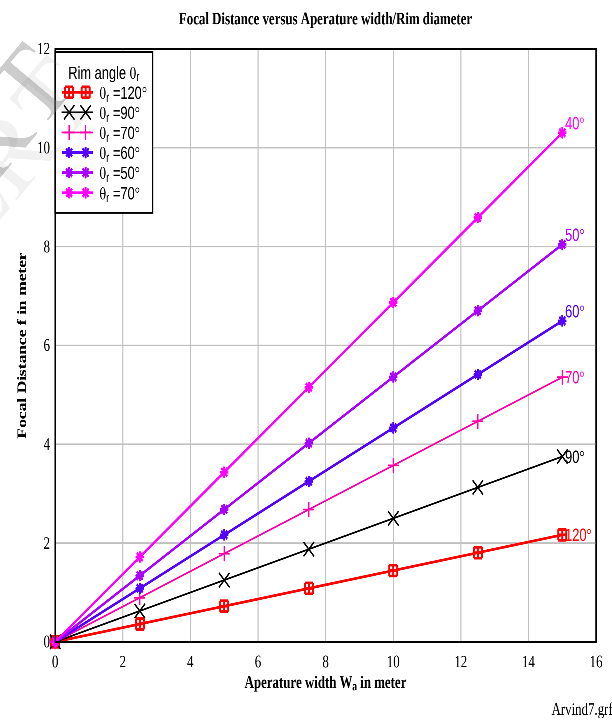

Fig.6.Plot between focal distance and rim angle of parabola

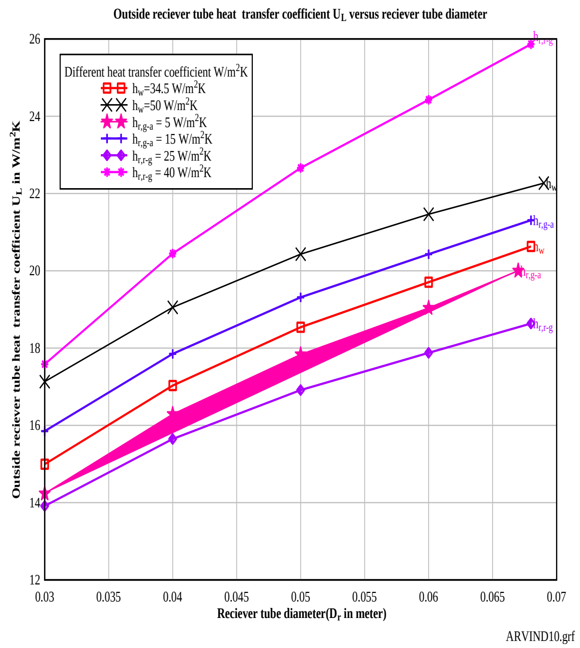

Fig.7.Plot between different heat transfer coefficient outside the receiver and receiver tube diameter

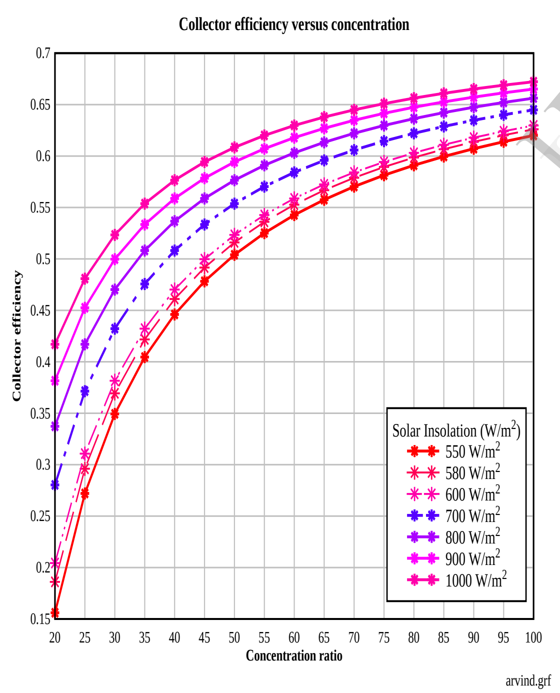

Fig.8.Plot between collector efficiency and solar Insolation

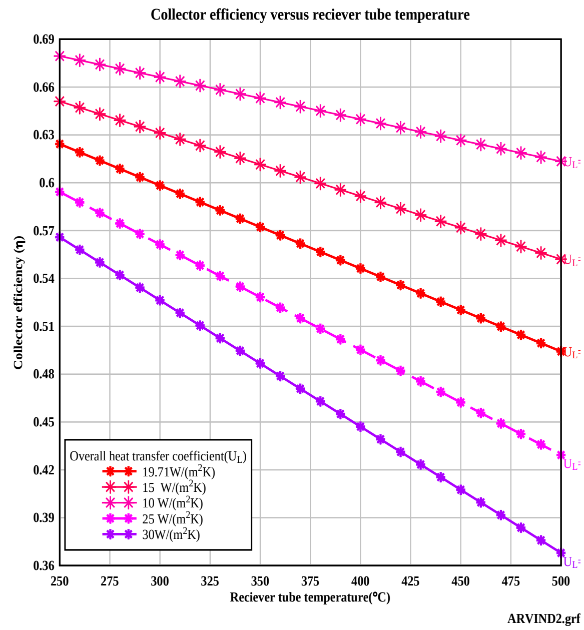

Fig.9.Collector efficiency and Overall heat transfer coefficient of receiver

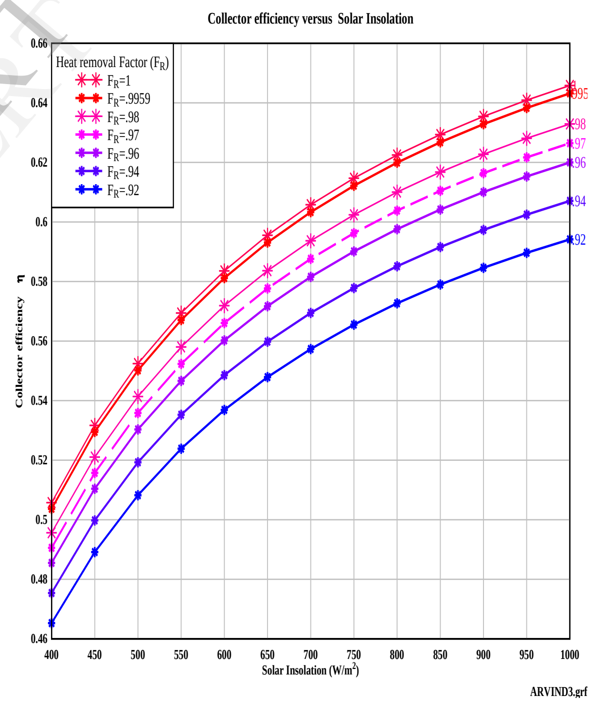

Fig.10.Plot versus Collector Efficiency and heat removal factor

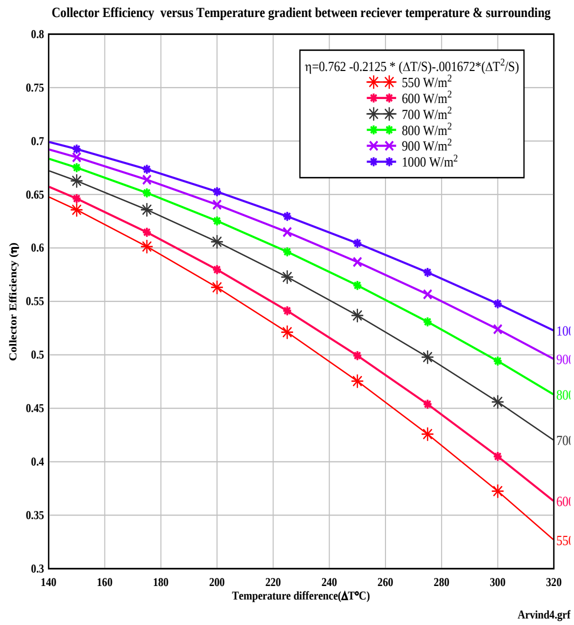

Fig. 11.Plot versus collector efficiency and solar insolation, according the performance equation of the IST (Industrial Solar Technologies

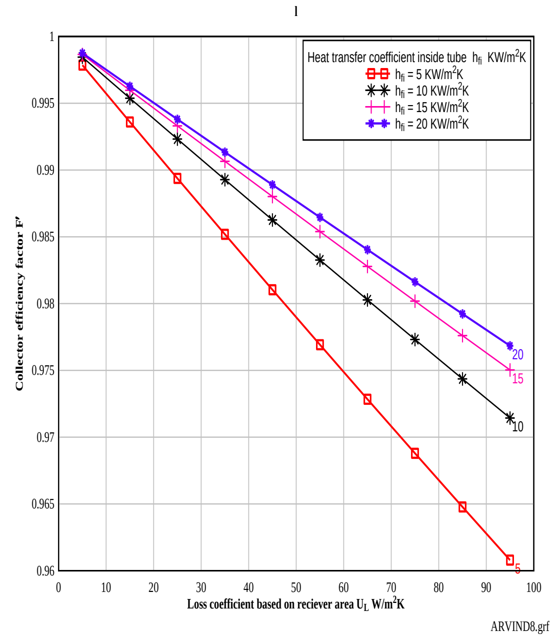

Fig. 13.Plot versus collector efficiency factor and heat transfer coefficient inside tube

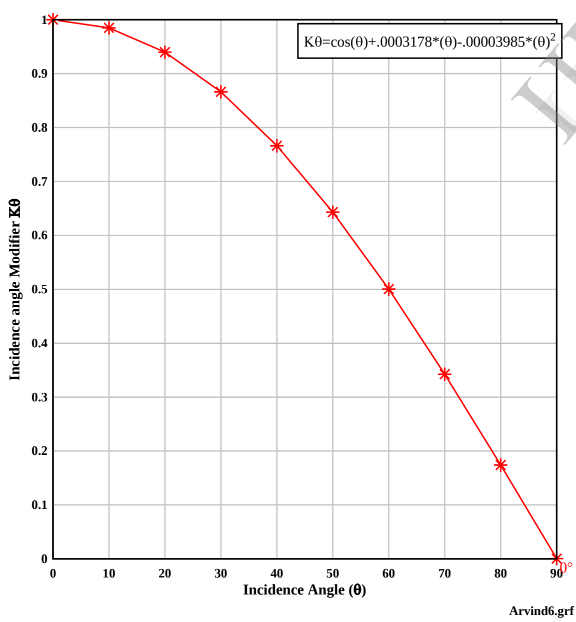

Fig. 12.Plot versus Incidence angle modifier and Incidence angle of sun ray on collector surface (For the IST collector)

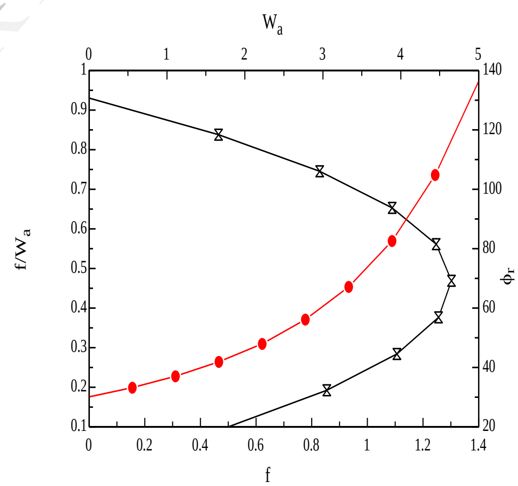

Fig. 14.Parabola focal length and rim angle

Conclusion

PTSC technology with DSG is very efficient & cheapest cost rate electricity generation as compared to solar energy power generation method. My designed electric output from grid is 1MWe. But, there are losses in electric grid & generator. We have accounted 95% efficiency of generator due to this losses; output of steam turbine must be 1.05 MWe

- In steam cycle, We found heat absorb in receiver tube per unit length, the mass flow rate, inlet temperature of water; pump work
- Input loss of heat from condenser and required trap heat in receiver tube to furnish 1.05MW steam turbine output, detailed result given in table 2.
- Then, we have reckoned the correct glass cover temperature by taking initial guess value which undergoes in five iterations, then calculated outside receiver tube heat loss coefficient, see results in table 4.
- We have found all the vital geometric parameter of PTSC of results see in table 3 such parameter as optical efficiency, geometry factor, intercept factor, rim radius, concentration ratio to make easy for analysis and manufacturing.
- We have done thermal analysis of PTSC to calculate the importance parameter/factor such overall heat loss/transfer coefficient, collector efficiency factor and heat removal factor is given in table 5, these factors provide the total collector area.
- Thereafter, we found the collector thermal efficiency ($\eta = 71.05\%$), overall plant/solar

  Field efficiency($\eta_{overall} = 23.48\%$) and 76.52% of heat lost from PTSC plant system.
- We have investigated 14 collectors and 1 module of collector for generation of 1MWe electricity generation, each collector consists of 8 modules and each module has length 12.27 m and width of 5.76m. The total area of collector/solar field is 7740.385 m^2. The investigated numbers of collector are based on standard collector Euro trough -100m, LS-3 data.
- We have investigated that parallel configuration of collectors (I-shape layout) is best to generate the steam at required temperature and pressure, also reduce heat loss and space accommodation. . We shall recommend four collector used in preheating, seven collector used in evaporation zone, and three collector used in superheated zone. We have modified diagram for practical use as shown in figure.14. In one module is not shown. This can adjusted to extend the length of module in superheated zone.

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Designed PTC field layout

Series configuration of collectors

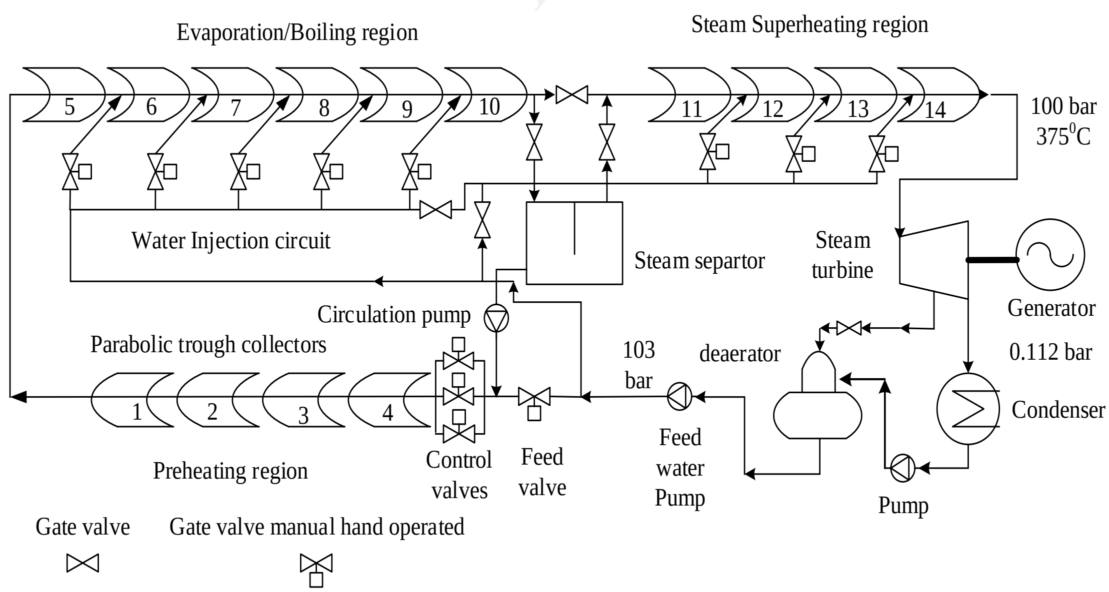

Fig.15. Designed schematic diagram for 1MWe PTSC plant based on DSG method

Parallel configuration of collectors

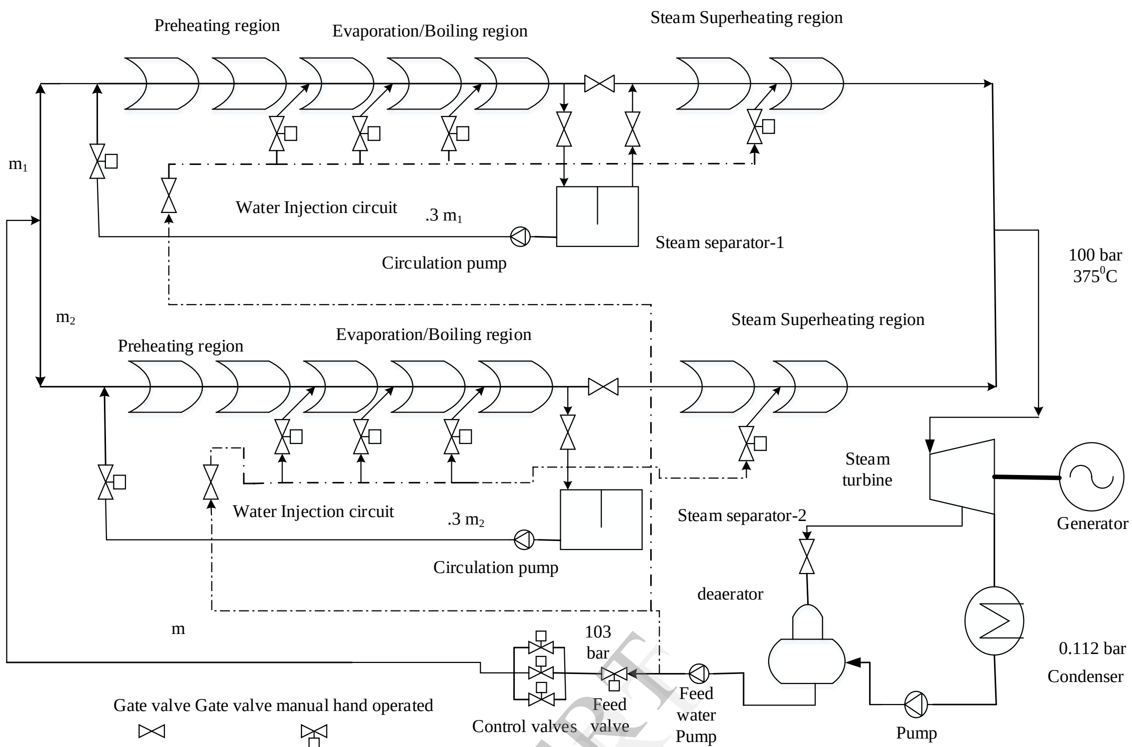

Fig. 16.parallel configuration of collector for 1MWe

Nomenclature

A_a	Aperture area of the collector, [m^2]
A_g	External surface area of glass cover, [m^2]
A_r	area of the receiver, [m^2]
c_{pw}	Specific heat of water, [J/kg.K]
c_{pa}	Specific heat of air at T_a , [J/kg.K]
C	Geometric concentration ratio
D_o	outer diameter of receiver tube [m]
D_i	Inner diameter of receiver tube [m]
T_r	Receiver temperature [K]
F_R	Collector heat-removal factor
f	focal length [mm]
G	Global irradiance on a horizontal surface, [W/m^2]
Gb	Beam irradiance incident on the aperture ($G_n \cos\theta$) , [W/m^2]
Gd	Diffuse irradiance, [W/m^2]
S	Direct normal (beam) irradiance, [W/m^2]
h_{fi}	Heat transfer coefficient inside the pipe,[W/m^2.K]
K_θ	Incidence angle modifier
Kss	Stainless steel Thermal conductivity of pipe, [W/m.K]

K_f	Thermal conductivity of water, [W/m.K]
K_{air}	Thermal conductivity of air , [W/m.K]
Lc	Length of one module, [m].
m_{flow}	Mass flow rate of the steam [kg/s]
C_{max}	Maximum concentration ratio
F'	Collector efficiency factor
$h_{r,g\text{-}a}$	linearized radiation coefficient from cover to ambient
$h_{r,r\text{-}g}$	linearized radiation coefficient from receiver to glass cover
h_r	The linearized radiation coefficient
h_c	Convective loss coefficient
h_w	loss coefficient for wind
A_e	Aperture area loss,[m^2]
A_b	Area loss due to plate extending from rim to rim,[m^2]
A_l	Total loss in aperture ,[m^2]
A_f	Geometry factor
r_r	Rim radius,[m]
d *	universal non-random error parameter due to receiver misallocation and reflector profile errors
Erf	Error function
ΔT	Temperature gradient between receiver and

	surrounding air,[K]
D_{rt}	riser tube outside diameter
d_r	displacement of receiver from focus
Tg	Glass cover temperature,[K]
D_g	Diameter of glass cover ,[m]
D_r	Diameter of receiver ,[m]
S_P	Curve length of parabola,[m]
H_P	Lactus rectum of parabola, [m]
n	Day number
Q_U	Solar collector useful output, [W/m^2]
t	Time, [s]
To	Heat transfer fluid outlet (from the collector) temperature, [K]
Ti	inlet temperature of fluid in the collector, [K]
Ta	Ambient temperature, [K]
T_m	Mean temperature of the heat transfer fluid across the collector or the solar field, [K]
V	Wind speed, [m/s]
U_L	Overall heat loss coefficient from absorber surface, [W/m^2.K]
U_O	Overall heat transfer coefficient of collector pipe, [W/m^2.K]
Wa	Collector width, [m]

Dimensionless groups

Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number

Greek symbols

η	Solar collector overall efficiency
$\eta_{overall}$	Solar field overall efficiency
η_o	Optical efficiency
ε	Emissivity coefficient
θ_r	Rim angle
β_r	misalignment angle error (degrees)
β *	universal non-random error parameter due to angular errors
σ *	universal random error parameter
θ_m	half acceptance angle [degrees]
ρ_w	Average density of the water between inlet and outlet temperatures.
ρ_w	Density of water, [kg/m^3]
ρ	Reflectivity constant
τ	Transmittance of the receiver glass envelope
α	Solar altitude angle
α_c	Absorptance of the absorber surface coating
γ	Intercept factor
σ_{total}	Standard deviation of the total errors, [mrad]
$\sigma_{optical}$	Standard deviation of the total optical errors, [mrad
σ_{slope}	Standard deviation of slope errors, [mrad]
$\sigma_{specular}$	Standard deviation of specular errors, [mrad]
$\sigma_{displacement}$	Standard deviation of receiver displacement errors, [mrad]
σ_{mirror}	Standard deviation of mirror, [mrad]
σ_{sun}	Root mean square (RMS) width of the sun, [mrad]
φ_r	Angle of Parabola
μ	Dynamic viscosity of water, [N.s/m^2]
μ_{air}	Dynamic viscosity of air, [N.s/m^2]

Abbreviations

DSG	Direct Steam Generation
INDI TEP	Integration of DSG Technology for electric production
SEGS	Solar electric generating system
ET100	Euro trough
PSA	Plata forma solar de Almeria
DSG	Direct steam generation
STTP	Solar trough thermal power plant
PTSC	Parabolic Trough solar Collector
PTC	Parabolic Trough Collector
IAM	Incidence Angle Modifier

Units

MWe	Megawatt electrical
kWe	kilowatt electric

References

[1] Ari Rabl, Comparison of Solar Concentrators, Pergamon Press 1976. Printed in Great Britain.

[2] M J Brookes 2005, thesis on Performance Of A Parabolic Collector, University Of Stellenbosch

[3] Ming Qu, David H. Archer, Sophie V. Masson, "A Linear Parabolic Trough Solar Collector Performance Model", Renewable Energy Resources and a Greener Future Vol.VII-3-3, 2006, ESL-IC-06-11-267.

[4] Scott A. Jones,Robert Pitz, TRSYS Modelling of SEGS VI parabolic trough solar electric generation system, proceeding of solar forum 2001: solar energy; the power to choose April 21-25,2001, Washington, DC

[5] Isabel Llorente Garcı, Jose Luis Alvarez, Daniel, Performance model for parabolic trough solar thermal power plants with thermal storage: Comparison to operating plant data, Solar Energy 85 (2011) 2443–2460, Elsevier

[6] Blanco Balbir Singh, Mahinder Singh, Designing a solar thermal cylindrical

parabolic trough concentrator by simulation, International Rio3 congress, world climate and energy event, Rio de Janeio,1-5,Dec 2003

[7] V. Siva Reddy , S.C. Kaushik , S.K. Tyagi Exergetic analysis and performance evaluation of parabolic trough concentrating solar thermal power plant (PTCSTPP), Energy 39 (2012) 258-273, Elsevier

[8] Iman Niknia, Mahmood Yaghoub 2011,Transient simulation for developing a combined solar thermal power plant, Applied Thermal Engineering 37 (2012) 196-207, Elsevier

[9] Iman Niknia, Mahmood Yaghoubi, Transient analysis of integrated Shiraz hybrid solar thermal power plant, Article in press, Renewable Energy xxx (2012) 1-6, Elsevier.

[10] Price, H., Luepfert, E., Kearney, D., Zarza, E., Cohen, G., Gee, R., and Mahoney, R. 2002. Advances in parabolic trough solar power technology. International Journal of Solar Energy Engineering, 124 109–125.

[11] S.K. Tyagi, Shengwei Wang, M.K. Singhal, S.R. Park, "Exergy Analysis and Parametric Study of Concentrating Type Solar Collectors", International Journal of Thermal Sciences 46 (2007) 1304-1310.R.Lugo-Leyte et al. (2012)

[12] Amirtham Valan Arasu and Samuel Thambu Sornakumar, "Performance Characteristics of the Solar Parabolic Trough Collector with Hot Water Generation System", BIBLID: 0354-9836, 10 (2006), 2, 167-174

[13] Soteris Kalogirou, Stephen Llyod and John Ward, "Modelling, Optimisation and Performance Evaluation of a Parabolic Trough Solar Collector Steam Generation System", Solar energy Vol. No.1, pp. 49-59, 1997

[14] Martin Kaltschmitt, Wolfgang Streicher Andreas Wiese, Renewable Energy Technology, Economics and Environment, Springer Berlin Heidelberg New York

[15] Halil M.Guven, Richard B. Bannerot, Optical analysis of parabolic trough solar collectors for technically less developed countries, June 1984,USAID

[16] Ricardo Vasquez Padilla, Gokmen Demirkaya, D. Yogi Goswami, Elias Stefanakos, Muhammad M. Rahman, "Heat transfer Analysis of Parabolic Trough Solar Receiver", Applied Energy 88 (2011) 5097–5110.

[17] Duffie; J. A.; Beckman, W. A.: Solar Engineering of Thermal Processes; John Wiley & Sons, New York, Brisbane, USA, 1991, 2nd edition

[18] Eck, M., Zarza, E., Eickhoff, M., Rheinlander, J., and Valenzuela, L. 2003. Applied research concerning the direct steam generation in parabolic troughs. Solar Energy. 74 341–351.

[19] Jeter, M.S., 1983. Geometrical effects on the performance of trough collectors. Solar Energy 30 , 109 – 113

[20] Soteris A. Kalogirou, Solar Energy Engineering Processes and Systems, Academic Press is an imprint of Elsevier

[21] A. Fadar, A. Mimet, A. Azzabakh, M. Pérez-García, J. Castaing, "Study of a new solar adsorption refrigerator powered by a parabolic trough collector", Applied Thermal Engineering 29 (2009) 1267–1270.

[22] Sukhatme, Solar Energy: Principles of Thermal Collection and Storage, third edition Publisher, Tata McGraw-Hill Education, 2008

[23] Sodha, M.S., Mathur, S.S., Malik, M.A.S., 1984. Wiley Eastern Limited, Singapore.

[24] Zarza, E., Hennecke, K., Hermann, U., Langenkamp, J., Goebel, O., Eck, M., Rheinlander, J., Ruiz, M., Valenzuela, L., Zunft, S., and Weyers, D.-H. 1999. DISS phase I project, Final project report, Editorial CIEMAT, Madrid