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Design of Solar Power Plant for One Megawatt Power with Central Cavity Receiver

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Abstract

Solar energy is a vital renewable form of energy and huge research has been done last few centuries for effective utilization of solar energy for the generation of power. Though solar power generation cannot compete economically at present, it is not too far away that the solar option is going to be techno-economical comparable with the primary source especially the fossil fuel resources depleting fast. The present work is an attempt to design the solar energy collection and storage system of a 1 MW Solar Central Cavity Receiver Power Plant (SCCRPP) in a unified way for its continuous operation on a 24 hour per day basis. The design is made by considering individually the design steps involved to design heliostat system, a detailed receiver system integrating three zones viz., preheating, evaporating, and superheating zones and storage system for nighttime operation of the plant and combine them to give a comprehensive design method. The study is restricted to the design of SCCRPP without the consideration of the cost economics of the components.

Disciplinary: Energy Engineering (Renewable Energy, Solar Energy), Power Plant Engineering, Thermal Engineering.

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1 Introduction

Energy is one of the most familiar terms during one's lifetime. The most energetic is the most capable of doing a thing (Purohit et al., 2010). Energy is required to sustain and improve the quality of life. Primitive man used his own muscles to help him convert energy into useful work. In course of time, man started cultivation, trained animals for work. Further, he began to use wind energy for sailing ships or driving windmills and the waterfalls to turn the water wheels. So far, he was using only renewable sources of energy. After the industrial revolution, he began to use coal, petrol, natural gas and other petroleum products. “The known resources of fossil fuels in the world are depleting very fast and it is estimated that (Sukhatme, 1984), by A.D 2040, man will have to increasingly depend upon renewable resources of energy”. Among the various energy alternatives, the nuclear and solar options stand out distinctly with their unique advantages and disadvantages.

2 Literature Review

2.1 Solar Central Cavity Receiver Power Plant

The central receiver concept is an old one, based in fact on the most ancient of solar energy events, the third century B.C, defense of Syracuse attributed to Archimedes (Brinkworth et al., 2005). The story as repeated by Galen was that Syracuse each with a polished shield reflected sunlight to a central point in the harbor, where the solar flux was amplified to a point where it repelled the invading Roman fleet.

The central receiver concept was the subject of a patent granted to CGO Barr in 1896. It was also the subject of several Soviet studies and experiments where a stationary boiler was surrounded by mirrors on railway cars that traveled around a semicircular track during the day.

2.1.1 Description of SCCRPP

The Solar Central Cavity Receiver Power Plant (SCCRPP) receives solar radiation on an array of the heliostat, which is placed around the tower for a multi-cavity receiver and in the north field for a single cavity receiver (Battleson 1981). The solar energy collected by the sun-tracking heliostats is reflected onto the top of a tower where the receiver is mounted. The receiver is inclined towards the heliostat field (Lipps et al., 1978). The working fluid in the receiver absorbs the heat, which is then used to generate power with the help of a thermal power plant.

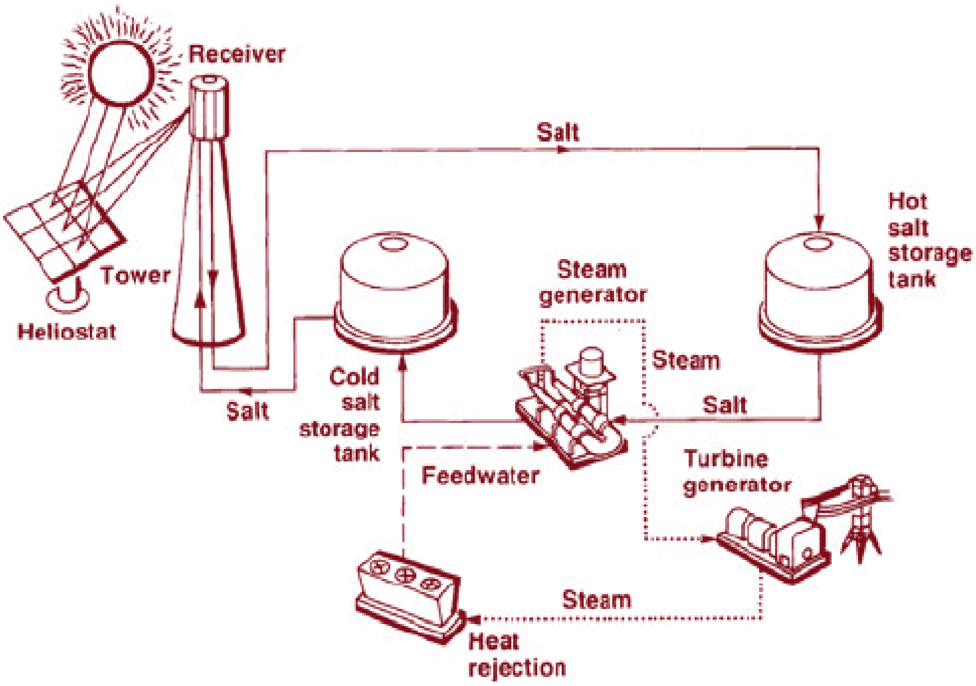

Figure 1: A simple solar central cavity receiver power plant (Gretz et al., 1984).

2.1.2 Heliostats

The heliostat assembly is comprised of Reflector Modules (Mirrors), Controller, Pedestal, Foundation and Tracking system.

The decision regarding the relative best position of receiver and heliostats is based on heliostat "loss" mechanisms and costs (Atif et al., 2015). Some of these loss mechanisms are Shadowing and Blocking, Atmospheric Transmittance and Optical Reflectance.

2.1.3 Shadowing and Blocking

The radial spacing ΔR and angular spacing ΔA given by Dellin et al. (1981) for high reflectance heliostats in large fields are as follows

$$\Delta R = HM(1.44\cot\theta - 1.094 + 3.068\theta - 1.1256\theta 2)\ (m)$$

$$\Delta A = WM(1.749 + 0.6396\theta + 0.2873/(\theta - 0.04902))\ (m)$$

The angle θ is the altitude angle to the receiver from the heliostat location of interest.

2.1.4 Atmospheric Transmittance

Vittetoe and Biggs have approximated atmospheric transmittance (τ) for two vital cases of the day

1. Clear day with 23 Km visibility,

$$\tau = 0.99326 - 0.1046*S + 0.017*S2 + 0.002845*S3$$

2. Hazy day with 5 km visibility,

$$\tau = 0.98707 - 0.2748*S + 0.3394*S2.$$

Where S is slant range from heliostat to receiver in Kilometers

2.1.5 Optical Reflectance

Of the reflective metals, aluminium and silver are the best performers with solar reflectance greater than 0.9. The solar reflectivity is generally in the range of 0.95-0.97 for good surfaces deposited by well-established mirroring procedures (Wei et al., 2010). The settling of dust particles also reduces the reflectance of the heliostats, thereby necessitating a frequent cleaning of mirrors to increase reflective efficiency. By using proper material these losses can be reduced to 5-7%.

2.1.6 Mirror Tracking

Although each mirror points in a different direction at the beginning and end of tracking correction, the angle of change and rate of angular change are identical for all mirrors (Collado et al., 2013). This means that once the mirrors are aligned to reflect energy onto the same target, they may be locked together and given the same tracking information for all tracking.

Both azimuth and elevation changes for every mirror are always equal to half of the angle of change of the sun over any time. Since the tracking information will be the same for all mirrors, sensors will be placed around the target boiler to sense off-target heating for the entire mirror group.

A simple tracking mode uses a heliostat rotating about a vertical axis. If the heliostat is tilted at an angle β from the horizontal and α is the sun's altitude angle, the incidence angle is $\cos\theta = \sin(\alpha+\beta)$.

3 Method: Thermal Design of SCCRPP

3.1 Estimation of Solar Radiation

For a specified power output, the total thermal energy collected through the heliostat-receiver system during daytime must be sufficient enough to produce power continuously over the night when solar radiation is not available. Hence thermal energy needed for nighttime operation should also be collected during daytime and stored in thermal storage through the storage system (Nithyanandam et al., 2014). The following set of relations can be used to estimate the intensity of radiation available on the heliostat installed at a given location. For this purpose, various terms are to be defined which are as under:

Declination

$$\delta = 23.45 * \sin (360 * (n+284)/365) \quad (1),$$

where n is the number of days counted from 1st January. December 21st is considered, as the day of the year as the value of the intensity of solar radiation is minimum on this day, therefore n is 355.

Solar hour angle is given by

$$\omega s = \cos-1 \ (-\tan\phi * \tan\delta) \quad (2).$$

Maximum day length is given by

$$Smax = (2/15) * \omega s \quad (3).$$

Daily extra-terrestrial radiation is given by

$$H0 = 24/\pi * 1.367 * 3600(1+0.03 \cos (360 * n/365)) * \omega s * \sin(\phi-\beta) \sin\delta + \cos(\phi-\beta) \cos\delta \sin\omega s \quad (4),$$

$$\beta = (\phi+15)0 \quad (5).$$

Daily global radiation at a location is given by

$$Hg = Ho(a+b * S/ Smax) \quad (6),$$

where a, b are constants and given in Table 1. The value of S can be obtained using a sunshine recorder and is taken as 8 hours

Hourly extra-terrestrial radiation is given by

$$Io = 1367(1+0.03 \cos (360 * n/365)) * \sin(\phi-\beta)\sin\delta + \cos(\phi-\beta)\cos\delta \cos\omega \quad (7).$$

Hourly global radiation is given by

$$Ig = Io*Hg/Ho(c+d*\cos\omega) \tag{8}$$

where c=0.409+0.5016 sin(ωs-60);d=0.6609-0.04767 sin(ωs-60).

For values of ω, flux at different hours and average flux over a day is calculated.

Table 1: Shows Constants *a* and *b* in Equation (6) for Indian Cities (Vittitoe et.al., 1979).

LOCATION	*a*	*b*
Ahmedabad	0.28	0.48
Bangalore	0.18	0.64
Hyderabad	0.28	0.48
Bhavanagar	0.28	0.47
Calcutta	0.28	0.42
Chennai	0.30	0.44
Goa	0.30	0.48
Jodhpur	0.33	0.46
Kodaikanal	0.32	0.55
Mangalore	0.27	0.43
Minicoy	0.26	0.39
Nagpur	0.27	0.50
New Delhi	0.25	0.57
Pune	0.31	0.43
Shillong	0.22	0.57
Srinagar	0.35	0.40
Trivandrum	0.37	0.39
Vizag	0.28	0.47

3.2 Estimation of Heliostat Dimensions

Solar radiation is available for only 'S' hours during a typical day. Power required at the receiver is given by

$$P_{rec} = (capacity*24/S)/(\eta_{gen}\eta_{tur}) \tag{9}$$

Power required at the heliostats is given by

$$P_{helio} = P_{rec}*(1-H_{loss}) \tag{10}$$

H_{loss} is taken as 65%.

The total effective area required is given by

$$A_t = P_{helio}*1000/I_g \tag{11}$$

The area of each heliostat is given by

$$A_h = WM*HM \tag{12}$$

The number of heliostats required is given by

$$N = A_t/A_h \tag{13}$$

The relation for spacing between the heliostats given by (Dellin et al., 1981) are as follows

$$\Delta R = HM*(\cot\theta-1.094+3.068\theta-1.1256\theta^2) \tag{14}$$

$$\Delta A = WM*(1.749+0.6396\theta+0.2873/(\theta-0.04902)) \tag{15}$$

θ is the heliostat angle from where the heliostat is located to the receiver tower. This angle is calculated as $\tan^{-1}(h/r)$. The height of the tower is initialized to 55m and the minimum radial distance from where the heliostats field can be arranged so that the shade of the tower does not fall on the heliostat field is given (15).

$$r = h/\tan(90-\delta-\phi) \quad (16).$$

The size of the aperture of the receiver is a function of the radial distance between

$$D_{rec} = r\,(32\pi/(60*180)+\varphi) \text{ where } \varphi=0.017\text{rad}. \quad (17).$$

The total land area required by the heliostat system is calculated, as the heliostats are arranged semi-circular

$$\text{Land area}=\pi*r^2/2.0 \quad (18).$$

The output data available from the calculations are set of spacing dimensions, Number of heliostats, minimum receiver diameter and land area for heliostats arrangement.

3.3 Design of Receiver

The central receiver mounted on a tower is a conical-shaped cavity-type receiver. It consists of a tube winded circumferentially inside the conical structure. The radiation from the heliostat is focused through an aperture at the bottom of the cone, into the cavity of the structure. The tube receives radiation, which is transferred to the fluid flowing through the tube. The tube section is divided into three sections namely, preheating, evaporating and superheating zones (Leonardi et al., 2011).

The problem of receiver design consists of estimating mass flow rate, for a given condition of temperatures and pressures. Estimation of tube diameter and length in all the zones and estimation of receiver dimensions. The thermal design of the receiver must satisfy the constraints namely, the allowable flux at receiver piping and minimum surface area of the receiver to lessen convective losses (Clausing 1981).

The receiver surface temperature T_{sur} is decided by considering the superheat temperature to be achieved by the working fluid. Therefore, the surface temperature must be at least 10^{o}c-20^{o}c more than the superheat temperature.

$$T_{sur} = T_{sur}+20^{O}C \quad (19).$$

Using the following relation total mass flow rate can be calculated as

$$M = P_{rec}/(C_p\Delta T+L+C_p\Delta T) \quad (20).$$

3.3.1 Preheating Zone

The preheating zone is designed using the heat flux at the receiver to be uniform throughout the section. The total mass flow rate of water needed M, and the inlet and outlet temperature of preheating zone T_{in} and T_{out} respectively as design input data. Initial tube diameter is assumed Mean Bulk temperature of the water

$$T_{mean} = (T_{in} + T_{out})/2 \quad (21).$$

The properties of water in preheating zone are taken at mean film temperature.

$$T_{min} = (T_{sur}+(T_{sat} + T_{in})/2)/2 \quad (22).$$

The velocity of flow is calculated by the relation

$$V_{pre} = m*4.0 /(\rho_{pre} * \pi * d_{pre}^2) \quad (23).$$

Reynolds number can then be calculated as

$$Re = V_{pre} * d_{pre} / \nu_{pre} \quad (24).$$

If the flow is turbulent, the Nusselt number can be calculated using the Dittus Boelter correlation

$$Nu = 0.023*Re^{0.8}*Pr^{0.4} \quad (25).$$

From this equation, the convective heat transfer coefficient is calculated

$$hc_{pre} = Kpre * Nu/d_{pre} \quad (26).$$

The radiative heat transfer coefficient for preheating zone of the receiver surface is given by

$$hr_{pre} = \sigma\varepsilon(T_{sur} + T_{mean})(T_{sur}^2 + T_{mean}^2) \quad (27).$$

The average overall heat transfer coefficient assuming negligible thermal conductivity of receiver pipe material is

$$1/U = 1/hc_{pre} + 1/hr_{pre} \quad (28).$$

From energy balance

$$m*C_p*(T_{out}-T_{in}) = U*A*(T_{sur}-T_{mean}) \quad (29).$$

From the above equation, one can calculate the area required. Therefore, the length of the tube is given by

$$L_{pre} = A/\pi*d_{pre} \quad (30).$$

The total heat absorbed in the preheating zone is

$$Q_{pre} = U*A*(T_{sur}-(T_{in}+T_{out})/2.0) \quad (31).$$

The heat flux in the preheating zone then is given by

$$Flux = Q_{pre}/((\pi*d_{pre}*L_{pre})/2) \quad (32).$$

If the value of flux calculated is greater than the allowable flux of water 0.7MW/m^2[refer Table 1] above which the pipe material gets overheated, then the diameter in preheating zone is changed and the above steps are continued till the constraint is satisfied.

3.3.2 Evaporating Zone

The heat absorbed in the evaporating zone is dependent on the saturation pressure and temperature. For a given saturation pressure the enthalpy of evaporation is known from the heat transfer data book (Kothandaraman et.al., 2021).

The total heat required in the evaporating zone is

$$Qevp = m^*h_{fg} \tag{33}$$

The velocity of steam in this zone is calculated using the relation

$$\mathrm{Vevp} = 4^*\mathrm{m}(\pi^*\mathrm{d}_{evp}^2) \tag{34}$$

The convective boilimg heat transfer coefficient is

$$\mathrm{hcevp} = 2.7[\mathrm{V}_{evp}\mathrm{K}_{evp}\rho_{evp}(\mathrm{h}_{fg}+\mathrm{h}_{fg}+0.4\mathrm{C}_{pevp}(\mathrm{T}_{evp}-\mathrm{T}_{sat}))/(\mathrm{d}_{evp}(\mathrm{T}_{evp}-\mathrm{T}_{sat}))]^{0.5} \tag{35}$$

The radiative heat transfer coefficient

$$\mathrm{hr}_{evp}=\sigma\varepsilon\,(\mathrm{T}_{evp}{}^4-\mathrm{T}_{sat}{}^4)/(\mathrm{T}_{evp}-\mathrm{T}_{sat}) \tag{36}$$

The overall heat transfer coefficient

$$\mathrm{U}=\mathrm{hc}_{evp}+3^*\mathrm{hr}_{evp}/4 \tag{37}$$

The total area required at the evaporating zone can be calculated using the relation

$$\mathrm{A}_{evp}=\mathrm{Q}_{evp}/\mathrm{U}^*(\mathrm{T}_{sur}-\mathrm{T}_{sat}) \tag{38}$$

The length of tube in evaporating zone is

$$\mathrm{L}_{evp}=\mathrm{A}_{evp}/(\pi^*\mathrm{d}_{evp}) \tag{39}$$

The flux in the evaporating zone is calculated

$$\mathrm{Flux}=\mathrm{Q}_{evp}/((\pi^*\mathrm{d}_{evp}{}^*\mathrm{L}_{evp})/2) \tag{40}$$

If the flux in the evaporating zone is higher than the allowable flux of steam 0.5MW/m^2 (refer to Table 1), then the diameter is varied till the flux constraint is satisfied.

3.3.3 Superheating Zone

The length of the tube required can be calculated using a similar procedure adopted for preheating zone. For the initial value of diameter, V_{sup} can be calculated using the mass flow rate m. The heat transfer coefficient, the total area required for the superheating zone and the length of the tube necessary can be estimated. The flux on the tube is compared with the allowable flux on steam. An iterative procedure can be carried out by changing the tube diameter till the matching occurs. The following relation can be used for the design of the superheating zone.

Total heat required at the superheating zone

$$\mathrm{Q}_{sup}=\mathrm{m}^*\mathrm{cp}_{sup}{}^*(\mathrm{T}_{sup}-\mathrm{T}_{sur}) \tag{41}$$

Reynolds number can be calculated using the relations

$$Re = V_{sup} * d_{sup}/\nu_{sup} \quad (42).$$

The Nusselt number can be calculated as

$$Nu = 0.24*Re^{0.6}*pr_{sup}^{0.3} \quad (43).$$

From this formula, tube side heat transfer coefficient can be calculated as

$$hc_{sup} = Nu*k_{sup}/d_{sup} \quad (44).$$

The radiation heat transfer coefficient is

$$hr_{sup} = \sigma\varepsilon(T_{sur} + T_{mean})(T_{sur}^{2} + T_{mean}^{2}) \quad (45).$$

The overall heat transfer coefficient assuming negligible thermal conductivity is

$$1/U = 1/hc_{sup}+1/hr_{sup} \quad (46).$$

Total surface area required at superheated zone can be calculated as

$$A_{sur} = Q_{sup}/U(T_{sur}-(T_{sat}+T_{sup})/2)) \quad (47)$$

The length of the tube is given by

$$L_{sup} = A_{sup}/(\pi*d_{sup}) \quad (48).$$

Flux in the superheating zone can be calculated

$$Flux = Q_{sup}/(A_{sup}/2) \quad (49).$$

If the flux calculated is greater than the allowable flux [refer to Table 1] the diameter of the tube is incremented, and the above iterative procedure is continued till the calculated flux matches the allowable flux.

The configuration of the cavity receiver is in the form of a truncated conical structure and is made from the total length of the tube which is wound around the inside surface of the truncated conical structure. The larger diameter of the cone cavity is equal to the aperture diameter. The diameter on the truncated side is initialized to 0.7 times the aperture diameter. The slant height of the truncated cone is calculated using

$$Sl_{rec} = (L_{pre}*d_{pre}+L_{evp}*d_{evp}+L_{sup}*d_{sup})/(\pi*(D_{rec}+d_{rec})/2) \quad (50).$$

The surface area of the receiver is calculated as

$$A_{rec} = \pi*SL_{rec}*(D_{rec}+d_{rec})/2.0 \quad (51).$$

If the allowable flux is not a major constraint under consideration and receiver surface area is a major constraint, then the surface area of the receiver is optimized. An iterative procedure is to be carried out again by varying the tube diameter till the surface area of the receiver is optimized.

3.4 Design of Storage Subsystem

The thermal energy for night operation is stored in tanks in the form of sensible heat using heat transfer fluids. If the liquid fluid in the receiver and the storage tanks are different then heat exchangers are used to transferring heat from one fluid to another to charge and discharge the heat. If the receiver fluid is air or steam and the sensible heat storage material is rock/sand, then heat storage tanks where the heat is absorbed by the rock/sand material (Bradshaw et al., 1988). In this type, estimation of tank dimensions to accommodate the required amount of sensible storage material is the only requirement. The inputs are the temperature limits of the fluids, storage capacity needed. In the present analysis fluid in the receiver is steam and the fluid in the storage subsystem is Nitrate salt (HITEC).

3.4.1 Estimation of Tank Dimensions

The following relations are used to estimate the tank dimensions.

In the present analysis, the storage capacity of the system is two-thirds of the rated power output.

$$Q_{st} = Capacity*2/3 \quad (52)$$

The mass flow rate of hot fluid is

$$m_h=Q_{st}/(c_{ph}(T_i-T_o)) \quad (53).$$

The mass flow rate of cold fluid is

$$m_c = Q_{st}/(c_{pc}(t_o-t_i)) \quad (54).$$

The mass of salt required is the product of the mass flow rate and the charging time. This amount of salt is stored in a tank. The volume and dimensions of the tank are calculated using

$$Volume = m_c*t/\rho_c \quad (55).$$

The tank is assumed to be cylindrical with an aspect ratio of 1:1.

The diameter given height of the cylinder tank is

$$D = H =[Volume*4/\pi]^{1/3} \quad (56).$$

The two tanks are of the same dimensions.

The tank is 50mm thick made of carbon steel (k_s =41.5 W/mk), with an insulation of 12 cn thickness using mineral wool (k_w=0.14W/mk). The thickness of insulation is taken so that the total heat loss from the tank surface doesn't exceed 5% of storage capacity. The heat loss through the tank surface is calculated using

$$\text{Heat loss}=(2\pi H(t_o-t_{amp})/[\ln((D+0.005)/k_{s+}\ln((D+t+0.005)/(D+0.005)/k_w)] + (\pi D^2/4)\,(t_o-t_{amp})/[0.05/k_s+0.12/k_w))1000.0 \quad (57).$$

Percentage loss in terms of the total capacity of storage is

$$\%loss = (Q_{st}-heatloss)/heatloss*100 \quad (58).$$

3.4.2 Heat Exchanger Design Calculations

The heat exchanger in between the hot fluid and cold fluid for charging and discharging the fluid is a shell and tube heat exchanger. The following are used to design the heat exchanger.

$$Area = \pi * d * l \quad (59),$$

$$V_h = 4*m_h/\rho_h \pi d_t 2 \quad (60),$$

$$Re = V_h * d_t/\nu_h \quad (61).$$

The Nusselt number is calculated using the correlation (Kothandaraman et.al., 2021)

$$Nu = 0.25*Re^{0.6}*Pr^{0.3} \quad (62).$$

The tube side heat transfer coefficient is calculated using

$$H_h = Nu*K/d_t \quad (63).$$

For the cold fluid flowing around the tube, velocity is given by

$$V_c = 4*m_c/\rho_c \pi\ (d_s-d_t) \quad (64),$$

$$Re = Vc*(\ d_s-d_t)/\nu_c \quad (65).$$

The Nusselt number is calculated using Equation (62). The shell side heat transfer coefficient is calculated using

$$h_c = Nu*K/d_s \quad (66).$$

The overall heat transfer coefficient is calculated

$$1/U=1/h_c+1/h_h \quad (67).$$

$$LMTD = [\Delta T_1-\Delta T_2]/[\ln(\Delta T_1/\Delta T_2)] \quad (68).$$

The effectiveness of heat exchanger is calculated by using the relations

$$NTU = U*Area/C_{min} \quad (69),$$

$$C = C_{min}/C_{max} \quad (70),$$

$$Eff = (1-\exp[-NTU(1-c)])/(1-C*\exp[-NTU(1-c)]) \quad (71).$$

The amount of heat is calculated

$$Qcal = U*Area*LMTD. \quad (72).$$

The calculated value is checked with the actual value, if the calculated value is less than the actual value the iterative procedure is carried varying the area of exchanger by assuming suitable dimension for length, tube, and shell diameters.

4 Result and Discussion

Thermal analysis of a 1MW SCCRPP is carried out using the design data given in Table 2

Table 2: Design Input Data Of 1MW SCCRPP

Location	Hyderabad
Number of the day(n)	355
Capacity of plant	1MW
Mass flow rate	3.42Kg/s
Saturated pressure of steam	50bar
Efficiency of generator	0.9
Efficiency of the turbine (thermal)	0.3
Total heliostat losses	0.35
Condenser pressure	0.06bar
Inlet temperature of the receiver	38^{o}c
Salt inlet temperature	275^{o}c
Steam inlet temp to storage	480^{o}c

The analysis is carried out for Hyderabad conditions. The solar condition is taken for December 21st as minimum solar radiation is the primary consideration.

4.1 Heliostat System

Heliostat dimensions = 12.5X12.5 m

Total number of panels = 144

Number of rows = 14

Tower height = 55 m

Ground area for heliostats = 128265 m^2

Table 3: Heliostat Systems Spacing

Radial Spacing	Axial Spacing	*Distance* To Tower	Number of Heliostats
27.66	32.98	65.40	6
33.24	33.54	93.06	9
41.04	35.08	126.30	11
51.85	37.77	167.34	14
66.64	42.07	219.19	16
86.65	48.84	285.83	18
113.55	59.84	372.48	20

4.2 Receiver System

Mass Flow rate = 3.42 Kg/s

Tube diameter = 0.05 m

Length of tube = 4486 m

Heat absorbed in receiver = 11111 KW

Receiver aperture diameter = 7.6 m

Receiver slant height = 11.1 m

Surface area = 224 m^2

Table 4: Heliostat Systems Spacing

Description	Preheat zone	Evaporator zone	Superheat zone
Length of tube, m	2469	79	1939
Diameter of tube, m	0.05	0.05	0.05
Flow Velocity, m/s	2.4	68.5	95.6
h_c, W/m^2k	14417	45567	2551
Heat absorbed, KW	4127	5612	1372

4.3 Storage System

Diameter of tank	=	11.5 m
Height of tank	=	11.5 m
Heat absorbed by tank	=	7436KW
Mass of HITEC salt	=	1331945 Kg
Pump Power	=	¼ HP

Heat loss in terms of the capacity of storage 3%.

5 Conclusion

The design is carried out on the thermal analysis of a 1MW SCCRPP, with a steam power cycle. For this purpose, the total heat energy needed to be absorbed by the receiver system is chosen to be equal to that of the requirement of a 1MW steam power cycle. Based on 8 hours of effective solar collection at the heliostat, the number of heliostat panels needed that can collect the energy needed for continuous operation of the plant over 24 hours a day is estimated. It is seen that 2/3 rd of heat energy collected from solar sources is to be stored in a thermal storage system for the nighttime operation of SCCRPP.

6 Availability of Data and Material

Data can be made available by contacting the corresponding author.

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