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MW Level Solar Powered Combined Cycle Plant with Thermal Storage: Thermodynamic Performance Prediction

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Abstract- The renewable resource, mainly the solar energy, can be used to produce electric energy on a large scale in solar thermal power stations, which concentrate sunlight at temperatures which range between 200° to 1200° C and even more. This paper presents a conceptual configuration of a solar powered combined cycle power plant with a topping air Brayton cycle and a bottoming steam Rankine cycle. The conventional gas turbine (GT) combustion chamber is replaced by a high-temperature solar thermal air heating system. During the daytime, a part of the exhaust air from the GT is bypassed to produce superheated steam in the heat recovery steam generator (HRSG), which in turn runs a steam turbine and the remaining exhaust air from GT is utilized to charging molten salt, which acts as a storage medium. The heat energy of the molten salt is utilized to generate steam for 4 hours in another HRSG, when sunlight is not available. From the thermodynamic analysis, it is found that for the base case GT pressure ratio of 4, power obtained from the GT block is 1.75 MW, while total power obtained from the combined cycle is 2.28 MW. The overall thermal efficiency of the combined cycle at this pressure ratio is 25.39%. The pressure ratio of the gas turbine has been varied from 2 to 20 and the optimum pressure ratio has been found out where total power output of the combined cycle plant is maximum.
Keywords- Solar energy; power tower; combined cycle; thermal storage; energy analysis.

Nomenclature

T	[K]	Temperature
r	[-]	Pressure ratio
k	[-]	Ratio of specific heats
m	[kg/s]	Mass flow rate
P	[MW]	Power
h	[MJ/kg]	Enthalpy
Q	[MW]	Required solar insolation
C_p	[MJ/kg-K]	Specific heat of air

Special Characters		
η	[-]	Efficiency
Subscripts		
e	[-]	Expansion
GT	[-]	Gas Turbine
ST	[-]	Steam Turbine
C	[-]	Compression
w	[-]	Water/Steam
a	[-]	Air
A, B, C….	[-]	State Points
mech	[-]	Mechanical
G	[-]	Generator
th	[-]	Thermal

1. Introduction

Generally non-renewable energy sources like coal, oil, gas etc. are used for the generation of electricity in the conventional power plants. These energy sources are finite in nature and require millions of years to become usable again. Another major disadvantage of these non-renewable energy sources is that they produce pollutant by-products, which create global warming and greenhouse effect.

On the other hand, renewable energy sources are free from these kinds of hazardous effects. These sources are safe today as well as tomorrow and do not create any burden for future generations with unnecessary risk. Out of various renewable energy sources available today, solar energy is one of the best promising options due to following reasons [1]:

- It is inexhaustible in nature
- It can meet present and future energy demands

The problems like pollution, global warming and green house effects can be reduced rapidly by more and more utilization of renewable energy such as solar power. It is environmentally friendly renewable energy option [2]. Concentrating solar power technology is one area where many countries are focusing nowadays like Southwestern United States which is one of the world's best areas for sunlight. The abundance of solar energy makes concentrating solar power plants an attractive alternative to conventional power plants. The main benefits of concentrating solar power plants include low operating costs, and the ability to produce power during peak energy demand periods. The solar energy can also be stored in proper storage medium which in turn can be used when the solar energy is not available [3].

Out of various concentrating solar thermal power technologies available today like parabolic dish, Fresnel collectors, parabolic trough etc., solar power tower or central receiver technology is a very promising option for generating solar power for utility scale applications. The general construction of central receiver technology is that a large number of heliostats or mirrors reflect the sunlight at the top of a tower, where a solar receiver is mounted. The receivers can be used to heat air or water. They can also be used to heat liquid sodium or molten salt, which can be used to store the energy before using it to boil water to drive turbines [4]. Two demonstration projects on solar power tower i.e. PS 10 and PS 20 are in operational stage in Seville, Spain which are producing 11 MWe and 20 MWe respectively [5]. Ivanpah in USA will be the world's largest solar power tower electric generating system of net power of 377 MW and it consists of three units: Ivanpah 1 has a total capacity of 126 MW and Ivanpah 2 and 3 are both 133 MW each [6].

Heat storage in solar thermal plants is an essential issue since solar energy is not available throughout day and night. Various thermal storage mediums are available like Therminol VP-1, molten salt, HITEC etc. Out of various heat transfer fluids available which can be used as heat storage medium, molten salt can be used over the other two since it remains in the liquid stage in a wide range of temperatures (290°C to 550°C). Two-tank storage is a prevailing design options for liquid thermal energy storage over single-tank storage since two-tank storage was found to reduce the levelized electricity cost (LEC). In a two-tank storage system, the volumes of hot and cold liquid (typically, a molten salt) are maintained in separate tanks. The storage system is charged by transferring excess heated HTF (heat transfer fluid) to the hot tank from the cold tank. To discharge the storage system, the process is reversed with the stored heat being delivered to the power block [7].

In combined cycle power plants, the heat rejected by the higher temperature cycle is recovered and used by the lower temperature cycle to produce additional power and gives high efficiency. Some of the combined cycles are: gas-steam, steam-organic fluids and gas-organic fluids. In the combined cycle power plant or combined cycle gas turbine, which has been conceptualized in the present study, a gas turbine generator generates electricity and waste heat is used to make steam to generate additional electricity via a steam turbine. Limited works have been done on central tower power plants and storage systems [8, 9, 10].

In the present study, a central receiver gas turbine combined cycle has been conceptualized, where air has been considered as the working fluid in the topping Brayton cycle, and the heat energy remaining in the air coming out from the gas turbine is utilized to generate steam in another HRSG as well as to heat the molten salt HTF. An in-house calculation code in C has been developed performing a thermodynamic performance analysis of the combined cycle.

2. Model Description

Figure1 shows the schematic of a solar powered gas turbine combined cycle considered in the present study. Ambient air at point 1 (300K and 1.01325 bar pressure) enters the compressor (**COMP**) and leaves at point 2. Compressed air then enters the solar receiver (**R**) which raises the air temperature in two stages until a final temperature of 1000°C (1273 K) is achieved. Heated air enters the gas turbine **(GT)** at point 3 and expands to point 4.

A part of the exhaust from GT is passed through a HRSG to produce superheated steam which in turn will be utilized to run a steam turbine in the bottoming cycle. In the HRSG, air is first used to raise the temperature of the steam in the superheater (**SUP**) from point ‘F’ to the point ‘A’. Then the air is used to evaporate the water from point ‘E’ to ‘F’ in the evaporator (**EVAP**) and finally, it is utilized to heat the feed water in the economiser (**ECO**) i.e. to raise the temperature of the feed water to saturation temperature. The superheheated steam at point ‘A’ (65 bar pressure and 460° C) enters into the steam turbine (**ST**) and after doing work in the steam turbine, steam is exhausted in the condenser (**COND**) at point ‘B’ at 0.075 bar pressure. After being condensed in the condenser, water is pumped from point ‘C’ to point ‘D’ i.e. at the boiler pressure by a feed pump (P). The system as well as the programming is developed in such a way that the temperature

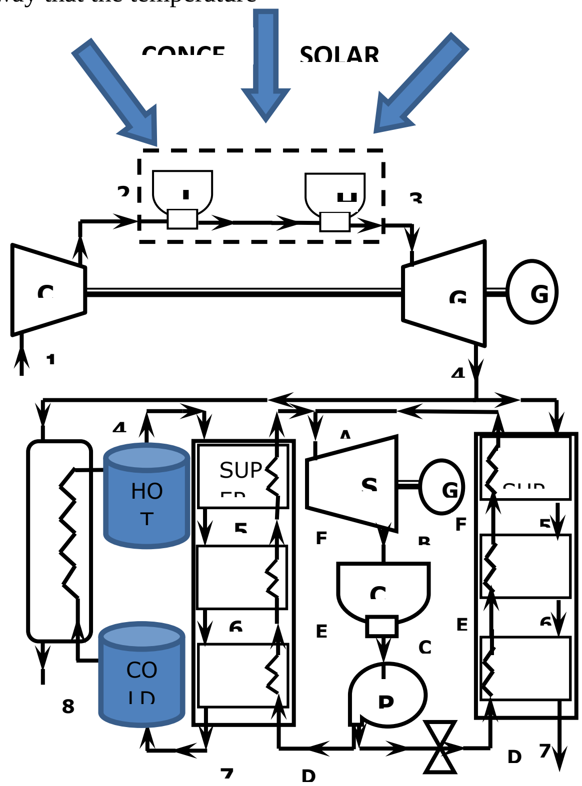

Figure 1. Schematic of the Proposed Model

difference between the evaporator exit temperature of air (i.e. T_6) and the saturation temperature of water (i.e. $T_E = T_F$) does not fall below 15° C for better heat transfer.

The other part of the hot exhaust air from gas turbine is utilized to heat up the molten salt in the heat exchanger (**HX1**), and heats up the molten salt during the daytime, coming out from the cold salt storage tank, which acts as the storage medium. The hot salt coming from HX1 is stored in the hot salt storage tank. While sufficient sunlight is not available, the hot salt from the hot salt storage tank flows to another heat exchanger (**HX2**), which comprises of economiser, evaporator and the superheater. The HX2 produces the superheated steam in the same condition as produced during the daytime (i.e. at 65 bar and 460°C). The same steam turbine is used to generate the same power for 4 hours as produced during the daytime. The simulation is done in such a way that molten salt temperature at the exit of the economiser does not fall below 290°C.

The air from the compressor is heated by two solar receiver (**R**) modules in series: low temperature (**LT**) and high temperature (**HT**) module. In the receiver cluster the air from the compressor of the gas turbine is heated up to 1000°C by concentrated solar energy. Figure 2 shows a schematic of a solar concentrator-receiver system which traps the incoming solar flux into the working fluid. A number of heliostats concentrate the solar radiation into the solar receivers, which are mounted on top of a tower.

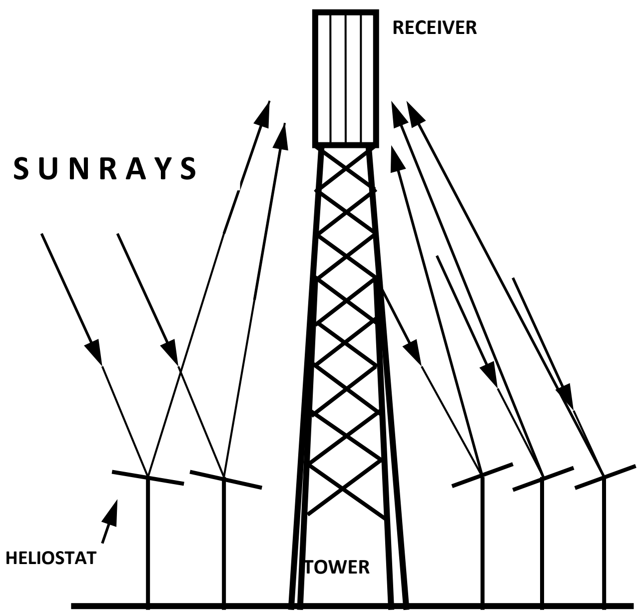

Figure 2. Solar Concentrator Receiver System

3. Solar Receiver Model

The solar receiver considered here is having two modules: low and medium temperature modules. Both the modules are similar in construction as used in SOLGATE project [11]. The low temperature (LT) module is nothing but a tubular receiver which consists of parallel tubes as shown in Fig. 3. The outlet temperature of air from the LT module is 550°C. For the low temperature receiver, the aim of the development was to achieve an overall cost reduction at the first, low temperature stage of the receiver cluster by utilizing simple, less expensive modules [12].

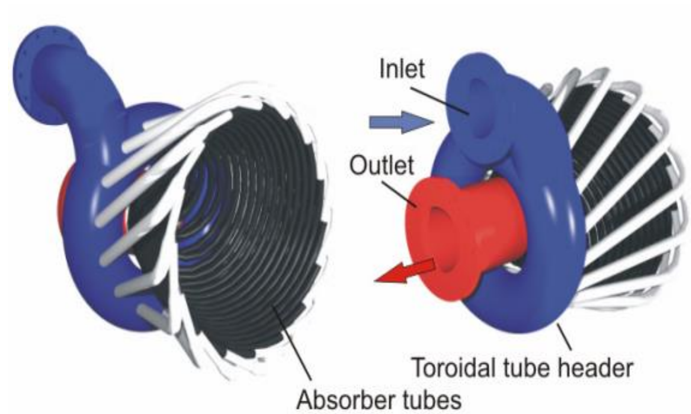

Figure 3. Schematic of LT Module [11]

For the high temperature (HT) module, ceramic foam absorber is used, installed on a ceramic mounting structure. For the absorber structures, SiC have been applied. To increase the absorption to 96%, the foam is coated with a silica layer and tempered for 100 h at 1400°C. For the absorber mounting structure a fiber-reinforced alumina-based structure was selected. The schematic of the above pressurized volumetric receiver is shown in Fig. 4 [11].

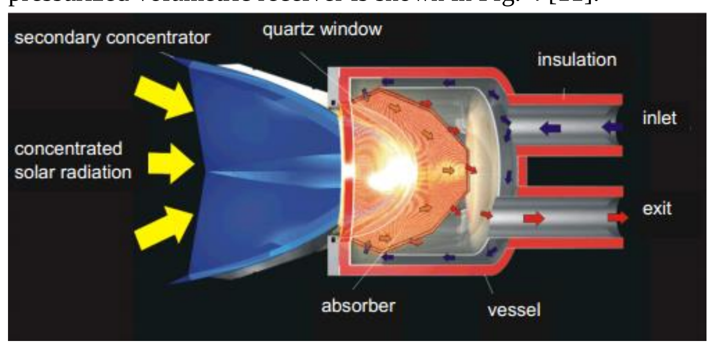

Figure 4. Schematic of HT Module [11]

4. Mathematical Model Descriptions of The System

Initially, mass flow rate of air (m_a) is considered as 3 kg/s. Since temperature of air at the entry to the compressor and pressure ratio (r) is known, temperature after isentropic compression is given by the following equation [13]

$$T_{2'} = T_1 (r)^{\frac{k-1}{k}} \tag{1}$$

But after actual compression in the compressor, temperature is obtained from the following equation [13]

$$T_2 = T_1 + \frac{1}{\eta_C}(T_{2'} - T_1) \tag{2}$$

Similarly, considering the isentropic efficiency of expansion ($\eta_{e,GT}$), temperature after expansion from gas turbine is given by the following equation [13]

$$T_4 = T_3 - \eta_{e,GT}(T_3 - T_{4'}) \tag{3}$$

HRSG Plant: Initially, outlet temperature of air from economizer is considered as 120^{O}C and later on it is corrected. Approximate mass flow rate of water (m_w) in the bottoming steam turbine plant can be found out from the overall energy balance in the HRSG and the enthalpy (h) at different points. Approximate mass flow rate of water in HRSG can be calculated from the following equation:

$$m_w = \frac{m_a \int_{T_7}^{T_4} C_{pa}\, dT}{(h_A - h_D)} \tag{4}$$

Where C_{pa} is the specific heat of air which follows the following polynomial equation where a_0, b, c, d are constants [14].

$$C_{pa} = a_0 + bT + cT^2 + dT^3 \tag{5}$$

In the superheater, steam is superheated by the hot exhaust air from the gas turbine. Evaporator inlet temperature of air (T_5) can be calculated from the energy balance equation in the superheater and is given by

$$T_5 = T_4 - \frac{m_w(h_A - h_F)}{m_a C_{pa}} \tag{6}$$

In the evaporator also, water is evaporated by taking the heat energy from air. Outlet temperature of air (T_6) from the evaporator can be calculated from the energy balance equation in evaporator:

$$T_6 = T_5 - \frac{m_w(h_F - h_E)}{m_a C_{pa}} \tag{7}$$

For better heat transfer, T_6 should be 15°C more than the saturation temperature of water. If the calculated T_6 does not satisfy this condition, custom-made codes in 'C' are developed in such a way that m_w is reduced by 0.001 kg/s and T_6 is recalculated. The process is repeated until T_6 satisfies the above condition and finally corrected T_7 is calculated from the below energy balance equation in economiser:

$$T_7 = T_6 - \frac{m_w(h_E - h_D)}{m_a C_{pa}} \tag{8}$$

Power obtained from steam turbine is calculated from the following equation [13]

$$P_{ST} = m_w(h_A - h_B) \tag{9}$$

Power required for the pump [13]

$$P_{Pump} = m_w(h_D - h_C) \tag{10}$$

Net power obtained from steam turbine [13]

$$\left(P_{net}\right)_{ST} = (P_{ST} - P_{Pump})\eta_{mech.}\eta_G \tag{11}$$

Thermal efficiency of the steam cycle [13]

$$\left(\eta_{th}\right)_{ST} = \frac{(P_{net})_{ST}}{m_w(h_A - h_D)} \tag{12}$$

4.1 Calculation for Mass Flow Rate of Salt in HX2 and Total Amount of Salt

Thermal energy required to be stored in the salt storage to run steam turbine at the rated output for 4 hours when sunlight is not available is given by the following equation

$$EnergyStored = \frac{4 \times (P_{net})_{ST} \times 1000}{(\eta_{th})_{ST}} \tag{13}$$

Approximate mass flow rate of salt through the HX2 can be calculated from the following equation

$$m_{salt} = \frac{EnergyStored}{1.6 \times (838 - 563) \times 4 \times 3600} \tag{14}$$

where specific heat of molten salt is 1.6 kJ/kg-K; maximum and minimum temperatures of molten salt are 565°C (838 K) and 290°C (563 K) [15].

Temperature of molten salt after superheater is given by the below equation

$$T_{5'} = T_{4'} - \frac{m_w(h_A - h_F)}{m_{salt} \times 1.6} \tag{15}$$

Temperature of molten salt after evaporator is given by the below equation

$$T_{6'} = T_{5'} - \frac{m_w (h_F - h_E)}{m_{salt} \times 1.6} \tag{16}$$

Evaporator outlet temperature of salt must be 15°C more than the saturation temperature of water for better heat transfer. If this condition is not followed, the mass flow rate of salt is adjusted accordingly.

Economiser outlet temperature of salt is given by the below equation

$$T_{7'} = T_{6'} - \frac{m_w (h_E - h_D)}{m_{salt} \times 1.6} \tag{17}$$

The temperature of the molten salt is maintained between 290°C (563 K) and 565°C (838 K) [15]. If T_7' is less than 290°C, mass flow rate of molten salt is revised so that T_7' is more than 290^{O}C.

Amount of salt required to be stored to run the steam turbine for 4 hours during non-availability of sun-light can be obtained from the below equation

$$M_{salt} = (4 \times 3600 \times m_{salt}) \tag{18}$$

4.2 Calculation for Additional Mass Flow Rate of Air

Heat energy required to be stored in molten salt is given by

$$E_{salt} = \frac{M_{salt} \times C_{p-salt} \times (838 - 563)}{\eta_{storage}} \tag{19}$$

Considering 6 peak sun-shine hours, heat stored in salt per second is given by

$$e_{salt} = \frac{E_{salt}}{(6 \times 3600)} \tag{20}$$

So, additional air required to heat the salt to required temperature is given by the below equation

$$m_{a(addl.)} = \frac{e_{salt}}{\eta_{effectiveness} \times \int_{563}^{T_4} C_{pa}\, dT} \tag{21}$$

4.3 Calculation for Overall Thermal Efficiency

Required solar energy to heat compressed air to the gas turbine inlet temperature can be obtained from the following equation [13]

$$Q = (m_a + m_{a(addl.)}) \int_{T_2}^{T_3} C_{pa}\, dT \tag{22}$$

Net power obtained from the gas turbine [13]

$$(P_{net})_{GT} = (P_{GT} - P_{comp}) \tag{23}$$

Overall thermal efficiency of the combined cycle [13]

$$\eta_{th,ovrl} = \frac{(P_{net})_{GT} + (P_{net})_{ST}}{Q} \tag{24}$$

5. Results and Discussions

This section first describes the assumed data, then the base case performance at pressure ratio 4 and subsequently, the parametric analysis based on energetic performance of the combined cycle at varying pressure ratios of the compressor.

a) Gas turbine inlet temperature of air is assumed as 27°C.

b) Steam turbine inlet temperature and pressure of superheated steam as 460°C and 65 bar.

c) Condenser pressure has been taken as 0.07 bar.

d) Gas turbine inlet temperature of air is 1000°C

e) The molten salt at hot salt storage tank is kept at 565°C and salt at cold salt storage tank is at minimum 290°C [15]

f) Specific heat of molten salt is 1.6 kJ/kg-K

The base case performance of the solar powered gas turbine combined cycle power plant is shown in Table 1 at compressor pressure ratio of 4. At the topping cycle (GT) pressure ratio of 4, the plant delivers a net power of 2.28 MW while both the GT shares a load of 1.75 MW and the balance is supplied by ST. To generate this power, the plant is required to absorb solar power of about 9 MW which is equivalent to about 11250 m² of collector surface exposed to an average insolation level of 800 W/m².

Table 1. Base case performance of the solar powered gas turbine combined plant

Parameter	Unit	
Gas Turbine Pressure Ratio	--	4
Required Solar Insolation	[MW]	9
GT Power	[MW]	1.75
GT Cycle Thermal	[%]	19.42
Mass Flow Rate of Steam	[kg/hour]	1876
Required Salt Storage	[MT]	193277
Net Plant Output (GT+ST)	[MW]	2.28
Overall Thermal Efficiency	[%]	25.39

As mentioned in Table 1, for the base case performance of the combined solar power tower plant at pressure ratio 4, the solar insolation requirement is 9 MW. For this solar insolation, the required number of heliostats is 750 considering the aperture area of each heliostat is 15.0 m² [16].

5.1 Parametric Analysis

The parametric analysis has been carried out at varying pressure ratios of the compressor to find out the energetic performance of the combined cycle plant. The pressure ratio has been varied from 2 to 20.

The net power obtained from the gas turbine cycle initially increases; takes a maximum value 1.96 MW at pressure ratio 7 and then decreases with the increase of pressure ratio. For the steam turbine cycle, work obtained from the cycle gradually decreases with the increase of pressure ratio since the mass flow rate of water decreases with the increase of pressure ratio as explained in Fig. 6. Since the gas turbine power takes the major share in

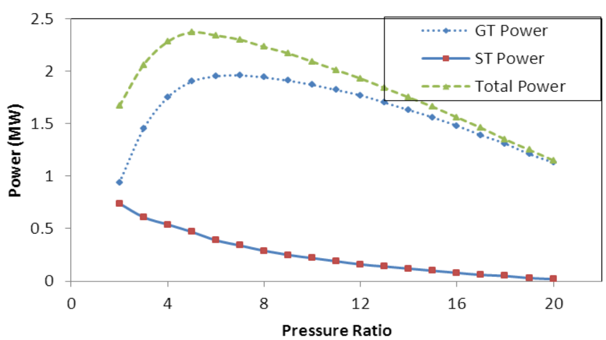

Figure 5. Variation of Power Obtained from Gas Turbine, Steam Turbine and Combined Cycle

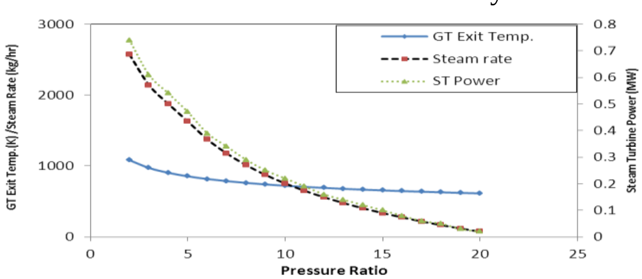

Figure 6. Variation of GT Exit Temperature, Steam Rate and Steam Power with Pressure Ratio

the total power obtained from the combined cycle, total power curve also takes initially the upward trend and then a downward trend with the increase of pressure ratio, which is evident in Fig. 5.

From Fig. 6, it is evident that air temperature at gas turbine exit is decreasing with the increase of pressure ratio. For the combined cycle, since the pinch point temperature difference is set at minimum 15°C for better heat transfer and temperature after gas turbine expansion is decreasing with the increase of pressure ratio, the evaporation rate for the HRSG is decreasing with the increase in pressure ratio. Since the mass flow rate of water is decreasing with the increase of pressure ratio, power obtained from steam turbine is decreasing with the increase of pressure ratio.

Figure 7 shows the variation of Thermal Efficiency of Gas Turbine and Combined Cycle with pressure ratio. For the gas turbine, thermal efficiency increases with the

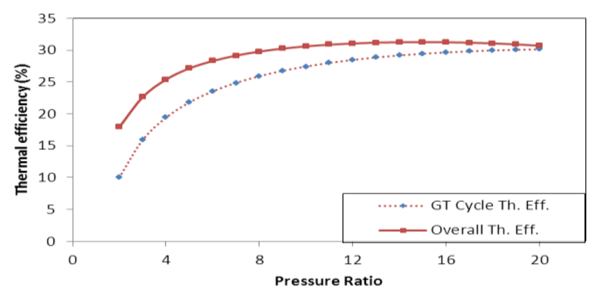

Figure 7. Variation of Thermal Efficiency of Gas Turbine and Combined Cycle with Pressure Ratio

increase of pressure ratio. But for the combined cycle, overall thermal efficiency increases initially but decreases for the higher pressure ratios.

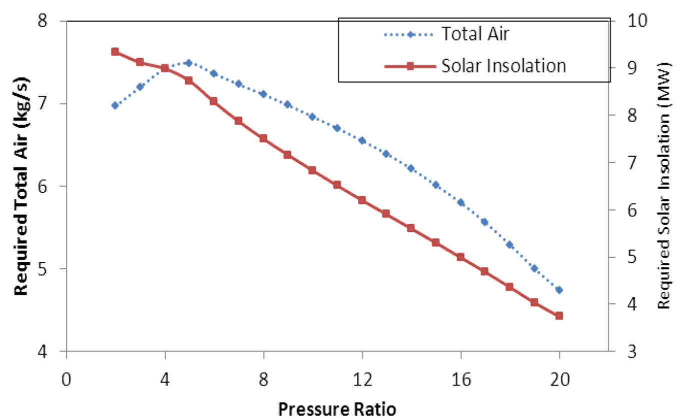

Figure 8. Variation of Required Total Air and Solar Insolation with Pressure Ratio

It is evident from Fig. 8 that the total air requirement initially increases and then decreases with the increase of pressure ratio. The total air after producing power in the gas turbine cycle is utilized for two purposes: one part is utilized for producing superheated steam in the HRSG plant and the other part is used for charging molten salt coming from cold salt storage tank. But the solar insolation requirement decreases with the increase of pressure ratio.

As already explained that mass flow rate of steam decreases with the increase of pressure ratio. Since the mass flow rate of steam decreases with the increase of pressure ratio, mass flow rate of salt also decreases with the increase of pressure ratio as shown in Fig. 9, which signifies that size of the salt storage will decrease with the increase of pressure ratio.

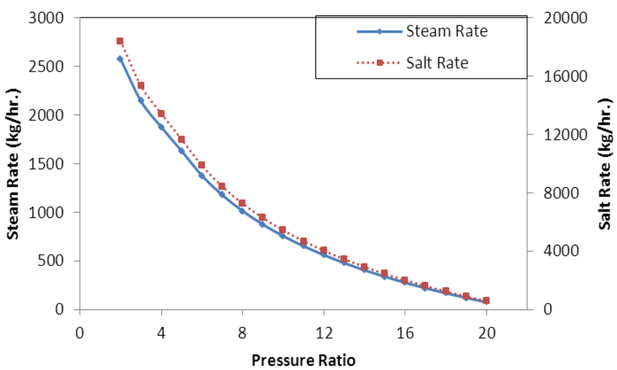

Figure 9. Variation of Steam Rate and Salt Flow Rate with Pressure Ratio

6. Conclusions

The present study proposes a conceptual configuration of solar tower combined cycle plant with molten salt storage: topping GT cycle with air as the working fluid and a bottoming HRSG plant. The hot molten salt is utilized to produce steam for 4 hours while sufficient sunlight is not available.

Parametric Analysis of the combined cycle has been done for the varying pressure ratio of the compressor and fixed gas turbine inlet temperature. It suggests that power output from combined cycle increases initially, reaches a maximum value (2.37 MW) at pressure ratio of 5 and then decreases. The overall thermal efficiency at this pressure ratio is 27.14%. So, the combined cycle can be operated at pressure ratio of 5 to get maximum total power.

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