Thermodynamic Study of Environment Friendly Air / Steam Combined Cycle

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ABSTRACT

Gas/Steam combined cycle power plants are extensively used due to their better thermal efficiency but the natural gas or other hydrocarbon fueled gas turbine plants create adversities for the environment. The present work deals with a combined cycle having a gas turbine cycle run on air which is brought up to the state of being expanded using a compressor and a heliostat based heating system. The discharge from the gas turbine is sent to a heat recovery steam generator for generating steam at the desired pressure and temperature for expansion in the steam turbine. The considered combined cycle configuration has been analyzed using thermodynamic modeling based on the first law of thermodynamics. The study involves parametric investigations for insight into the combined cycle which is free from carbon emission and environment friendly.

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Nomenclature

1, 2, 3...18	Cycle states as shown in figure 1
A	Heliostat Area
CEP	Condensate extraction pump
CSP	Concentrated solar power
f_r	Fraction of steam bleed
G	Irradiation
HPC	High pressure compressor
HPGT	High pressure gas turbine
HRSG	Heat recovery steam generator
HPST	High pressure steam turbine
LPC	Low pressure compressor
LPGT	Low pressure gas turbine
LPST	Low pressure steam turbine
m	Mass
P	Pressure
p	Pump
Q	Heat
T	Temperature
W	Work

h	Enthalpy
η	Efficiency
γ	Ratio of specific heats

Subscripts

a	Air
abs	Heat absorbed by molten salt
C_p	Specific heat polynomial for heat
Cond.	Condenser
gt	Gas turbine
h	Heliostat
Int.	Intercooler
loss	Heat loss in the central receiver
over	Overall
P	Polytropic
s	Solar
sa	Salt
st	Steam turbine
w	Water

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1. Introduction

Energy demand is growing day by day and traditional source of energy like fossil fuel, which have the major contribution in energy production are depleting at alarming rate. These fossil fuel based energy options are largely responsible for environmental degradation so it is imperative to find an alternative of reliable and long lasting source of energy which should not only reduce the harmful impact on environment but also help in improving the performance of available power plants. This can be done safely by incorporating renewable energy sources like solar energy. The most direct method of harnessing solar energy is through concentrated solar power (CSP) which has gained attention as a best solution to the sustainable production of energy. Sun is the abundant source of energy for a greener planet hence called energy of future which never runs out. It is getting attention for power production from the most of researchers as this helps in reducing dependency on fossil fuel and thus resulting in cutting down the cost. From an International Energy Agency study, it is estimated that by 2050 CSP could provide 11.3% of global electricity [Shahnazari, and Lari, 2017]. Combined cycle power plant have become most favored thermal power cycles because of their better efficiency. Here in case of combined cycle, the bottoming cycle runs on the exhaust of topping cycle using heat recovery steam generator for steam generation. It has potential to run the low temperature cycle which otherwise will be emitted to atmosphere. This idea necessitates new power plants and improvement in the performance of available power plants by incorporating renewable energy resources which help in carbon free power generation with minimum environmental degradation. The possibility of using air as a working fluid in gas turbine cycle and integrating it with steam cycle using heat recovery steam generator is the motivation behind present thermodynamic study of the solarized combined cycle for power production.

Pioneering work done in the field of solar energy utilization shows that the solar energy has highest global potential as the renewable source because it is the safe form of inexhaustible free energy so adoption of solar based technologies would lessen the issues related to energy security, climate change, and global carbon emission level [Kabir et al., 2018]. It also decreases the consumption of fuel or transportation charges of these non-renewable sources. Different types of solar technologies currently in working are parabolic trough collector (60°C -500°C), solar dish (100°C -700°C) and heliostat field (150°C-1500°C). From this, it is obvious that heliostat has the largest and most flexible operating temperature range [Kalogirou, 2004]. Amongst all the clean technologies, solar energy serves as an effective renewable energy source to mitigate greenhouse gas emissions and helps to reduce global warming [Ramachandra et al., 2011]. The amount of solar energy reaching the earth's surface is 6000 times the present global consumption of energy [Bosshard, 2006]. There is abundant availability of solar radiation of 1700-1900 kWh per kW peak for more than 300 clear sky days in a year. The total installed capacity of CSP in India is 12.28883 GW till March 31, 2017, which increased four times in two years which was 3.74397 GW in March 2015 [Kumar, 2017]. Wang et al. [2017] modelled and compared performance of different cycles for applications in central receiver system from the aspects of efficiency, the specific work output, and the ability to install thermal energy storage. Among all the CSP technologies, the solar power tower technology offer highest efficiency and greatest potential for scaling. Shahin et al. [2016] concluded in their investigations that the use of a reheat system between the two stage turbines will increase the net power output.

A brief literature review carried out in respect to the present work is detailed herein [Shin et al. 2002] showed how the combined cycle continues to gain acceptance as a reliable, flexible and efficient base load power generation. Bijarniya et al. [2016] discussed about the problems with conventional power plants, overview, and design of various CSP technologies and site selection criteria of CSP in India is outlined. Mahtta et al. [2014] identified the potential sites for installation of CSP and provided solar irradiation data in India. Shaaban [2016] worked on Integrated Solar Combined Cycles (ISSC) with steam Rankine cycle and organic Rankine cycle as bottoming cycles. Binamer, [2019] developed a mathematical model of a typical integrated solar combined cycle system using engineering equation solver, to evaluate the performance of commercial ISSC power plants. Hogerwaad et al. [2017] studied a solar based multigenerational system for power generation, refrigeration, and desalination. Stein, and Buck, [2017] reviewed advanced power cycle for CSP technologies and analysed that the gas turbine combined cycles driven by CSP are highest efficiency power cycles. Xu et al. [2011] worked on energy and exergy analysis of solar power tower plants and presented their performances. Calise et al. [2018] presented a thermo-economic and environmental comparison between the integrated solar

combined cycle and a conventional combined cycle.

Looking upon the potential of solar energy and the ongoing studies, it is evident that the solar powered thermal energy is quite attractive. The present paper deals with the solar-powered air run gas turbine cycle combined with the steam run steam turbine cycle. The use of heliostat based heating system for creating working fluid capable of producing positive work in gas turbine offers the advantage of no contamination of working fluid. This offers the possibility of recycling the air as working fluid in gas turbine installation while using the energy carried by gas turbine exhaust for generation of steam in heat recovery steam generator (HRSG). The performance investigation of the air/steam run combined cycle is carried out based on thermodynamic modeling of the considered cycle on Ist law of thermodynamics. The results obtained have been analyzed and presented herein.

2. System description

Figure 1 shows the schematic of the combined cycle consisting of gas-turbine cycle coupled with a steam Rankine cycle along with the heliostat field. The heliostat field reflects solar radiation to a receiver, which carries molten salt. In the current analysis, a mixture of salt KNO₃ (40 %, wt./wt.) & NaNO₃ (60 %, wt./wt.) is used as a salt [Xu et al. 2011]. Air enters the compressor at state 1 for being compressed to a state 2 in low pressure compressor then goes to the intercooler. Here perfect intercooling is considered and then it is compressed from state 3 to higher pressure and temperature at state 4 in HPC. The compressed air then enters a molten salt to the air heat exchanger, where it is heated using heat from the molten salt. The high pressure and temperature air is then expanded in the high pressure gas turbine from state 5 to state 6 from where it again goes to molten salt to air heat exchanger and after expansion in the low pressure gas turbine from state 7 to state 8, the low pressure and moderate temperature air passes through the HRSG, where it exchanges heat with the water delivered by the feed pump of the reheat steam Rankine cycle at state 18. The high temperature superheated steam leaving the HRSG enters high pressure steam turbine at state 10 and expands to state 11 from where it again goes to HRSG and gets the temperature similar to state 10 and work on low pressure steam turbine at state 12 and expands up to the state 14 with a provision of steam bleeding at state 13 for the deaerating purpose and condenser condense steam from state 14 to state 15 which is then pumped with the help of feed pump from state 17 to 18 and again goes to HRSG.

3. Thermodynamic modelling

The salient features of thermodynamic modelling of the considered configuration are detailed ahead. Air is considered the working fluid for topping cycle with molar composition of 21% oxygen and 79 % nitrogen. The specific heat of air as a function of temperature in kJ/kg K is given by [1],

$$C_p(T) = 0.999639 - 0.055205 \times 10^{-3} \times T + 0.346320 \times 10^{-6} \times T^2 - 0.140119 \times 10^{-9} \times T^3$$

3.1. Air run topping cycle

Heat gained by air due to temperature variation between any two states 1 and 2 is given as:

$$Q = \int_{T_1}^{T_2} m_a \cdot c_p(T) dT \quad (1)$$

Considering the polytropic compression process the temperature of air after compression is given by:

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma \cdot \eta_{pc}}} \quad (2)$$

$$\text{Considering perfect intercooling, } T_2 = T_3 \quad (3)$$

Compressor work, considering the deployment of intercooler in between, is given by:

$$W_c = m_a \cdot (h_2 - h_1) + m_a \cdot (h_4 - h_3) \quad (4)$$

In the present solarized combined cycle the temperature at inlet to gas turbine depends upon the heat added to the compressed air in molten salt heat exchanger. Turbine inlet temperature T_5 is obtained from the energy balance equation no (5)

$$Q_{in} = m_a \cdot \int_{T_4}^{T_5} C_p(T) \cdot dt + m_a \cdot \int_{T_6}^{T_7} C_p(T) \cdot dt \quad (5)$$

Figure 1 Schematic of solar powered air / steam combined cycle

In the gas turbine run with air as working fluid, considering the polytropic efficiency of turbine, the temperature of air after the expansion is given by:

$$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{\frac{\gamma-1}{\gamma}\eta_{pt}} \quad (6)$$

$$\text{Considering perfect reheating, } T_5 = T_7 \quad (7)$$

Heat received from the solar radiation on Heliostat field is given by:

$$Q_s = G \cdot A \quad (8)$$

Energy balance in receiver expressed as-

$$Q_{in} = Q_{abs} + Q_{loss} \quad (9)$$

The rate of heat absorbed by the molten salt can be calculated as:

$$Q_{abs} = m_{sa} C_{psa} (T_{sa,out} - T_{sa,in}) \quad (10)$$

$$Q_{loss} = Q_{loss,cond} + Q_{loss,conv} + Q_{loss,ems} + Q_{loss,ref} \quad (11)$$

Work output from the gas turbine can be given as:

$$W_{gt} = m_a \cdot (h_5 - h_6) + m_a \cdot (h_7 - h_8) \quad (12)$$

Heliostat efficiency is given as:

$$h = \frac{Q_{in}}{Q_s} \quad (13)$$

For calculating efficiency of gas turbine cycle the total solar heat falling on heliostat (Q_s) is considered and the efficiency of gas turbine cycle is given as:

$$\eta_{gt} = \frac{W_{net \text{ of gas turbine}}}{Q_s} \quad (14)$$

Where, $W_{net \text{ of gas turbine}} = W_{gt} - W_c$

3.2. Steam run bottoming cycle

In HRSG, the energy balance yields:

$$m_a \cdot (h_8 - h_9) = m_w \cdot ((h_{10} - h_{18}) + (h_{11} - h_{12})) \quad (15)$$

$$\text{Considering perfect reheating, } T_{10} = T_{12} \quad (16)$$

In steam turbines, the work output from HPST and LPST is given as:

$$W_{HPST} = m_w \cdot (h_{10} - h_{11}) \quad (17)$$

$$W_{LPST} = m_w \cdot (h_{12} - h_{13}) + m_w \cdot (1 - fr) \cdot (h_{13} - h_{14}) \quad (18)$$

$$W_{pump} = \int v dp \quad (19)$$

$$Q_{Cond.} = m_w \cdot (1 - fr) \cdot (h_{14} - h_{15}) \quad (20)$$

Efficiency of steam turbine cycle:

$$\eta_{st} = \frac{W_{st}}{Q_{st}} \quad (21)$$

Where, $W_{st} = W_{HPST} + W_{LPST} - W_{pump}$

$$Q_{st} = m_w \cdot ((h_{18} - h_{10}) + (h_{12} - h_{11})) \quad (22)$$

Where Q_{st} refers to the heat carried from HRSG for steam generation.

For the total solar heat falling on heliostat being given as Q_s the overall efficiency of the cycle without considering the heat lost in intercooler is given as:

$$\eta_{over} = \frac{W_{over}}{Q_s} \quad (23)$$

Where, $W_{over} = W_{net \text{ of gas turbine}} + W_{st}$

Heat lost in intercooler is recovered for process heat is given by:

$$Q_{int.} = m_a \cdot (h_2 - h_3) \quad (24)$$

Efficiency of solar powered air/steam combined cycle considering the utilization of heat rejected in intercooling for process heating is given as:

$$\eta_{over} = \frac{W_{over} + Q_{int.}}{Q_s} \quad (25)$$

An exhaustive computer program in C language has been written based on thermodynamic modeling for studying the influence of variation of independent parameters namely the ambient temperature and cycle pressure ratio on various dependent parameters in the considered solar powered air / steam combined cycle for other parameters kept constant.

4. Results and discussion

Results are obtained from thermodynamic investigations of the considered air/steam combined cycle based on input parameters given in Table 1. The combined cycle performance has been analyzed for cycle pressure ratio varying from 14, 16, 18 & 20 and the ambient temperatures varying as 283 K, 288 K & 293 K [Shaaban, 2016]. The results have been graphically plotted and analyzed ahead.

Table 1 Input Parameters [Hogerward et al. 2017, Rabbani et al. 2015, Shaaban 2016, and Xu et al. 2011]

Parameter, Unit	Value
Solar Irradiation, W/m ²	1000
Heliostat Area, m ²	16,666
Heliostat efficiency, %	75
Receiver efficiency, %	80
Polytropic efficiency of compressor, %	90
Polytropic efficiency of turbine, %	92
Ambient Temperature, K	283, 288, 293
Ambient pressure, bar	1
Cycle pressure ratio	14, 16, 18, 20
Exit pressure of air at gas turbine, bar	1.2
HP steam turbine pressure, bar	90
LP steam turbine pressure, bar	5
Condenser pressure, bar	0.07
Bleed pressure, bar	1.2

Figure 2 shows the variation of gas turbine work output with the cycle pressure ratio for different ambient temperatures. It depicts that as cycle pressure ratio increases the work output of gas turbine also increases at every ambient temperature. At any ambient temperature, the increase in pressure ratio will need more work for compression and shall also yield more expansion work. Here the increase in the work output with the increase in cycle pressure ratio is attributed to the fact that the increase in positive work is more than the increase in negative work, which results in overall increase in the work output. But with increasing ambient temperature the work output of the gas turbine is seen to be decreasing at any cycle pressure ratio for a constant solar irradiations. At any cycle pressure ratio, the lower ambient temperature will incur lesser compressor work. For constant heat available, the temperature at inlet to expander may also get lowered resulting in decline in work output from it. But the net effect is seen to be in the form of overall enhancement of work output from the cycle due to the reduction in negative work being more than the reduction in positive work and thus the net gain in the overall work output. For fixed amount of solar irradiations and ambient temperature of 293 K, the turbine inlet temperature is seen to be varying as 1051 K, 1048 K, 1046 K and 1045 K at cycle pressure ratio of 14, 16, 18 and 20 respectively. At any ambient temperature, the turbine inlet temperature is seen to decline gradually with increasing cycle pressure ratio due to the combined effect of increase in cycle pressure and reheating deployed during expansion process. Here the maximum work output of 0.455 MW per kg of air is obtained from air powered gas turbine at cycle pressure ratio of 20, ambient temperature of 283 K and turbine inlet temperature of 1032 K.

Figure 3 shows the variation of work output of steam cycle with cycle pressure ratio at different ambient temperatures. The graphical pattern shows that at any ambient temperature as the cycle pressure ratio increases, for the fixed quantity of solar irradiations, the increase in expansion results in the decrease of gas turbine exhaust temperature. This reduction in gas turbine exhaust temperature eventually reduces the quantum of heat available for generation of steam in the heat recovery steam generator and thus steam turbine work output decreases with the increase in cycle pressure ratio. Further, at any cycle pressure ratio, the steam cycle work output is seen to be increasing with the increase in ambient temperature. It can be seen from the cycle arrangement that the temperature of air admitting into compressor i.e. the ambient temperature will directly affect the temperature after compression. For definite amount of solar irradiations the higher temperature after compression will result in higher temperature at inlet to gas turbine and thus offer higher temperature of exhaust gases from the gas turbine. This high temperature exhaust from gas turbine holds better potential to generate steam in HRSG and thus exhibit the increase in the work from the steam cycle. Maximum steam cycle work output of 0.212 MW/kg of air entering gas turbine cycle is obtained at the cycle pressure ratio 14, ambient temperature of 293 K with turbine inlet temperature of 1051 K.

Figure 2 Variation of work output of gas turbine with cycle pressure ratio at different ambient temperatures

Figure 3 Variation of work output of steam turbine with cycle pressure ratio at different ambient temperatures

Figure 4 shows the variation of overall work output with cycle pressure ratio at different ambient temperatures. It depicts that as cycle pressure ratio increases, the overall work output of the cycle decreases at any ambient temperature. Also, at any cycle pressure ratio, the overall work output increases with the increase in ambient temperature. At some cycle pressure ratio, the increase in overall work output with increase in ambient temperature is attributed to the higher gas turbine inlet temperature for given solar irradiations due to higher temperature of air entering the molten salt heat exchanger. This results in higher gas turbine exhaust temperature which offers more steam generation in HRSG and thus there is increased steam cycle work output along with increased gas turbine work output. With the increase in cycle pressure ratio, the overall cycle work output diminishes due to the fixed heat input from the solar irradiations and the larger pressure ratio yielding larger expansion and thus resulting in lower exhaust gas temperature which eventually reduces

the output from HRSG leading to reduced steam cycle work output. The maximum overall work output of 0.646 MW/kg of air entering gas turbine is found at cycle pressure ratio 14, ambient temperature of 293 K and turbine inlet temperature of 1051 K.

Figure 5 shows the variation of the gas turbine cycle efficiency with the cycle pressure ratio at different ambient temperatures. In the considered range of cycle pressure ratio of 14 to 20, the gas turbine cycle efficiency is seen to increase with cycle pressure ratio at any ambient temperature. For given solar irradiations, this increase in gas turbine cycle efficiency occurs due to the increment in positive work being more than the increase in the negative work. Further, at any cycle pressure ratio, the gas turbine cycle efficiency is seen to be decreasing with increasing ambient temperature. It shows that the lower ambient temperature offers high efficiency at any cycle pressure ratio, which may be due to the reduction in the compressor work requirement.

Figure 6a shows the variation of overall efficiency of the cycle with the cycle pressure ratio for different ambient temperatures. It is evident that the overall efficiency of the combined cycle decreases with the increase in the cycle pressure ratio at each ambient temperature. This pattern of variation is due to the increase in negative work being more than the increase in positive work with the increasing cycle pressure ratio. Also, at any cycle pressure ratio, the increase in ambient temperature offers the increase in the overall efficiency. The increase in ambient temperature

Figure 4 Variation of overall work output of cycle with cycle pressure ratio at different ambient temperatures

Figure 5 Variation of efficiency of gas turbine cycle with cycle pressure ratio at different ambient temperatures

Figure 6a Variation of overall efficiency of cycle with cycle pressure ratio at different ambient temperatures

Figure 6b Variation of efficiency of overall cycle and compressor work requirement with cycle pressure ratio with or without intercooler at ambient temperature of 283

leads to the corresponding increase in gas turbine inlet temperature which results in the higher gas turbine exhaust temperature yielding more steam generation and thus more contribution from steam turbine; however, the efficiency of gas turbine cycle is lesser at higher ambient temperatures. Hence the higher ambient temperature at inlet to compressor exhibits higher overall efficiency due to better utilization of solar energy, which yields more turbine inlet temperature and thus the improved performance of gas turbine, HRSG and steam turbine. The maximum overall cycle efficiency is found to be 38.80 % at cycle pressure ratio of 14 and ambient temperature of 293 K having turbine inlet temperature of 1051 K while ignoring the heat lost in intercooling. If the heat lost in intercooler is recovered for process heating, then the maximum value of overall efficiency is found to be 49.68% at cycle pressure ratio of 20 and ambient temperature of 283 K. The overall cycle efficiency of the solar powered air/steam combined cycle having intercooling and reheating in the topping cycle studied in this paper is found to be better than the published work of Rabbani et al. [2015] which deals with the energy and exergy analysis of the solar powered air/steam combined cycle due to the use of intercooling and reheating in the topping cycle and the feed water heating in steam cycle in the present work.

The relevance of deploying intercooling has been analyzed by assessing the compressor work requirement in the two different arrangements of the combined cycle with intercooler and without intercooler. Figure 6b shows the variation of efficiency of overall combined cycle and compressor work requirement with cycle pressure ratio for the cycle configuration with intercooler and without intercooler at ambient temperature of 283 K. It depicts that at any cycle pressure ratio, the compression work requirement decreases with the use of intercooler in the compression process. The reduction in compression work requirement due to use of intercooler is seen to be by 22.5%, 23.73%, 24.8% and 25.7% at cycle pressure ratios of 14, 16, 18 and 20 respectively. The intercooling offers the added opportunity of utilizing the heat rejected in intercooler for process heating and consideration of this heat in overall efficiency estimation shows that the overall efficiency of combined cycle having topping cycle with intercooling is more than the combined cycle having topping cycle without intercooling.

Figures. 2, 3, 4, 5, 6a to 6b are showing the graphical variation of efficiency and work output from the gas turbine based topping cycle, steam turbine based bottoming cycle and the overall air/steam combined cycle along with the assessment of the impact of deployment of intercooling during compression in the topping cycle. The presence of error bars in each of these graphical depictions explicitly exhibit the range of variation in respective parameter.

5. Potential applications and future scope of this technology

The present study pertaining to thermodynamic analysis of solar powered air/steam combined cycle consisting of gas turbine in topping cycle and steam turbine in bottoming cycle holds the potential to produce power without using any hydrocarbon fuel subjected to the availability of solar energy. The absence of use of any fuel makes the solar powered air/steam combined cycle environment friendly due to zero direct emissions. The deployment of intercooler in the topping cycle paves the way for enhancing the power output from the topping cycle along with the possibility of using the rejected heat at intercooler of gas turbine for process heating applications. It also facilitates the possibility of using the heat rejected at condenser of steam Rankine cycle for process heating.

The present thermodynamic study can be used for developing a prototype of air/steam combined cycle power plant that is environment friendly. The combined cycle efficiency can be further enhanced by improving the solar energy recovery for air heating and identifying the sources of inefficiency by carrying out the exergy analysis of the considered combined cycle based power generation system.

6. Conclusions

Following conclusions have been obtained from this study of solar powered air/steam combined cycle:

1. The considered solar powered air / steam combined cycle consisting of air run gas turbine cycle and steam Rankine cycle does not yield any emissions during its functioning.
2. Efficiency of solar powered air/steam combined cycle is 38.80% at cycle pressure ratio of 14 and ambient temperature of 293 K.
3. Efficiency of solar powered air/steam combined cycle considering the utilization of heat rejected in intercooling for process heating yields the efficiency of 49.68% at cycle pressure ratio of 20 and ambient temperature of 283 K.
4. Use of intercooling results in 25.7 % reduction in the negative work requirement in compressor and 4.04 % gain in the net overall work output from the combined cycle.
5. The increase in ambient temperature leads to an increase in the overall efficiency of the combined cycle; however, the efficiency of gas turbine cycle is lesser at higher ambient temperatures.

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