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Thermo-economic Analysis of Air Powered Solarized Intercooled-reheat GT Cycle Combined with Steam Rankine Cycle and Organic Rankine Cycle for Power and Cooling

Gaitry Arora[1] and Onkar Singh[2]

[1]PG Scholar, Mechanical Engg. Deptt. HBTU, Kanpur (U.P.) – INDIA
2Professor, Mechanical Engg. Deptt. HBTU, Kanpur (U.P.) – INDIA
E-mail: [1]gaitry24arora8@gmail.com, [2]onkpar@rediffmail.com

Abstract—*Global challenges in respect to the energy availability can be met through the improvement in performance of existing energy technologies or through newer energy generation options or combination of the both. The renewable energy specially the solar energy is the potential source for powering the thermodynamic cycles for producing the electricity with minimum adverse impact on surroundings. The solar powered intercooled-reheat type gas turbine cycle can be suitably integrated with the steam Rankine cycle for utilizing the energy carried by GT exhaust in heat recovery steam generator (HRSG) and can be further utilized in heat recovery vapour generator (HRVG) for running the low temperature organic Rankine cycle. The present study considers the Brayton cycle run on air combined with steam Rankine cycle and low temperature organic Rankine cycle (ORC) run on R1233zd(E) as working fluid. Also, the heat rejected during intercooling in multistage compressor is utilized for running the low temperature vapour absorption refrigeration system to get the cooling effect in addition to the power from gas turbine, steam turbine and ORC turbine. Here thermodynamic modeling has been carried out for analyzing the performance of considered combined cycle configuration for power and cooling. The study also includes the economic analysis for yielding the cost per unit cost of power and cooling. Here the air powered solarized GT cycle combined with steam Rankine cycle and organic Rankine cycle for power and cooling offers the thermal efficiency of 60.56% and cooling effect of 15.26 ton for cycle pressure ratio 8 and 293K ambient temperature.The cost of unit carbon free power and cooling has been obtained as 4904 US$/kW.*
Keywords: Combined cycle, Intercooling, Reheating, Vapourabsorption refrigeration cycle, R1233zd (E), Organic Rankine cycle

Nomenclature	
1, 2, 3….	Cycle states as shown in schematic
A	Heliostat Area, m^2
CSP	Concentrated solar power
evap.	Evaporator
f_r	Fraction of steam bleed
G	Generator
HPC	High-Pressure compressor
HPGT	High-Pressure gas turbine
HPST	High-Pressure steam turbine
HRSG	Heat recovery steam generator
HRVG	Heat recovery vapour generator
HX	Heat exchanger
I	Solar radiation
ISCC	Integrated solar combined cycle
LPC	Low pressure compressor
LPGT	Low pressure gas turbine
LPST	Low pressure steam turbine
m	Mass
ORC	Organic Rankine cycle
p	Pump
Q	Heat
ST	Steam Rankine cycle
VT	ORC vapour turbine
η	Efficiency
γ	Ratio of specific heat

Subscripts	
a	Air
abs	Heat absorbed by molten salt
amb.	Ambient
c	Compressor
cond	Conductivity
conv	Convectivity
cep	Condensate extraction pump
cond.	Condenser
ems	Emissive
exhaust	Exhaust air
GT	Gas turbine
h	Heliostat
ms	Molten salt
loss	Heat loss in the central receiver
n	Normal
over	Overall

p	Polytropic
ref	Reflective
s	Solar
ST	Steam turbine
w	Water

1. INTRODUCTION

Worldwide increasing energy requirementhas been demanding for simultaneous increase in the energy generation from different power generation technologies. In spite of the numerous energy sources, the major share of energy is still generated using the fossil fuels which createadversities and also the depletion of fossil fuels calls for an urgent attention towards finding the sustainable source of energy. It is evident that in general the renewable sources of energy hold great potential in this regard and the harnessing of solar energy in particular offers attractive power generation option. The solar energy offers limitless potential in India the average daily solar energy in India lies between 4 and 7 kWh/m^{2}days and the same needs to be harnessed for efficient and cost effective power production. Combined cycle power plant offers the effective technological option to utilize the energy available in the system as it uses the energy available with the exhaust gases. Literature review in respect to solarized power plants shows that there are two ways of generating electricity from solar energy, one is photovoltaic (PV) and another one is concentrated solar power (CSP). In CSP the flat plate collector is used for up to 100° C whereas the concentrating solar collectors such as parabolic trough collector (PTC), dish collector (DC), heliostats, etc., can be used for high-temperature range such as 60–500°C, 100–700°C, and 150–1500°C respectively observed by Kalogirou [1].The organic Rankine cycle (ORC) is a cycle for lower temperature range such as lesser than 400°C.Shaaban[2] performed the thermodynamic analysis for an integrated solar combined cycle (ISCC)including steam Rankine cycle (ST) and ORC as two bottoming cycles. 15 different organic fluids for organic Rankine cycle are analyzed and R1234ze(z) comes out to be the suitable based on thermo-economic, safety, and environmental considerations. The absence of combustion in the topping cycle yields zero emission and carbon free power.Caliseet al. [4] presented a thermo-economic and environmental comparison between the integrated solar combined cycle and a conventional combined cycle.Hogerwaardet al. [7] studied a multigenerational system for additional commodity production from a single heat source. Combined power and cooling cycles have been explored for improving their overall energy conversion efficiency and decreasing the cost of energy produced.Worldwide many electric companies faced rapid growth in electricity demand peaks in the last years.

The present study deals with Thermo-economic analysis of air powered solarized intercooled-reheat GT cycle combined with steam Rankine cycle and organic Rankine cycle for power and coolingwith no carbon emission.The system consists of a intercooled-reheat type topping Brayton cycle and two bottoming cycles out of which one is steam run steam Rankine cycle and the other is organic fluid run organic Rankine cycle. The air run GT cycle employs two stage intercooled compression and a reheating during expansion while utilizing the solar energy collected through heliostats and used through a molten salt heat exchanger. During intercooling, the heat rejected in intercooler is utilized by the generator of vapour absorption refrigeration cycle. The energy carried by the exhaust air having potential leaving the GT is utilized by steam Rankine cycle through HRSG and the exhaust air exiting from HRSG goes to HRVG for rejecting the heat available with it to power the ORC. The considered combined cycle configuration has been analyzed using thermodynamic modeling based on the first law of thermodynamics along with its economic analysis. The results obtained have been analyzed and presented herein.

2. SYSTEM DESCRIPTION

Figure 1 shows the schematic of the solarized combined power and cooling cycle consisting of heliostat field to power the intercooled-reheat type gas-turbine cycle coupled with steam Rankine cycle and organic Rankine cycle. The heliostat field reflects solar radiations to a receiver, carries molten salt. In this study a mixture of work KNO_3 (40 wt. %) & $NaNO_3$ (60 wt. %) is used as a salt in molten salt heat exchanger,Xu et al. [3]. Air enters the compressor at state 1 for being compressed to state 2 in LPC and then goes to the intercooler for being cooled before subsequent compression. The heat rejected by the compressed air in intercooler is used in generator of ammonia-water based vapour absorption refrigeration cycle for getting the cooling effect. Here perfect intercooling is considered and the air is compressed from state 3 to4in HPC. The compressed air then enters a molten salt heat exchanger, where it is heated using heat from the molten salt. The air at high pressure and temperature is then expanded in the HPGT up to state 6 from where it again goes to molten salt heat exchanger for getting reheated. After reheating the further expansion occurs in the LPGT up to state 8 from where the low pressure and moderate temperature air passes to HRSG, where it transfers it's heat for steam generation from the water delivered by the feed pump of the reheat steam Rankine cycle at state 18.The superheated steam leaving HRSG entersHPST at state 10 and expands to 11 from where it goes to HRSG and again attains the temperature similar to that at 10. The reheated steam enters LPST at state 12 and expands up to state 14 with the provision of steam bleeding at state 13 for deaerating purpose.The condensate is then pumped by feed pump from state 17 to 18 and againenters the HRSG for steam generation. The air leaving HRSG at state 9 goes to HRVG and exits at state 1. The low temperature heat available with air passing through HRVG is used for evaporating refrigerantR1233zd (E) to act as the working fluidin vapour turbine. The high pressure refrigerant from state 28 expands to 29 and goes to ORC condenser. The condensate moves from state 30 to 31 and is again sent to HRVG for getting the working fluid for vapour turbine.

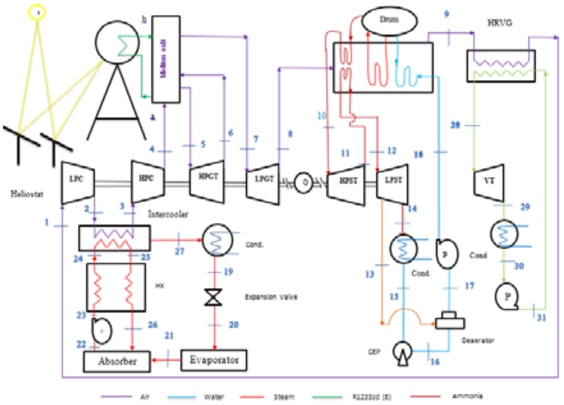

Figure 1 Schematicof air powered solarized intercooled-reheat GT cycle combined with steam Rankine cycle and organic Rankine cycle for power and cooling

3. THERMODYNAMIC MODELING

The salient features of thermodynamic modelling of the considered configuration are detailed ahead.

3.1 Air properties

Air is considered as the working fluid for topping Brayton 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[11]–

$Cp = 0.99963 – 0.055205x\ 10^{-3}\ x\ T + 0.34632\ x\ 10^{-6}\ x\ T^2- 0.14012\ x\ 10^{-9}\ x\ T^3$ (1)

3.2 Gas Turbine

Air temperature after the compression, considering polytropic efficiencyis expressed as-

$\frac{T_2}{T_1} = (\frac{P_2}{P_1})^{[\frac{\gamma-1}{\gamma.\eta p,c}]}$ (2)

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

$Q = \int_{T1}^{T2} m_a \cdot C_p(T) dT$ (3)

Taking perfect intercooling $T_2 = T_3$(4)

Work of compressor is given by:

$W_c = m_a \cdot (h_2 - h_1) + m_a \cdot (h_4 - h_3)$ (5)

Considering polytropic efficiency of turbine, the pressure and temperature variation in expansion is given by:

$\frac{T_5}{T_6} = \left(\frac{P_5}{P_6}\right)^{[\frac{\gamma-1}{\gamma}].\eta_{p,t}}$ (6)

Considering perfect reheating, $T_5 = T_7$ (7)

Gas turbine work outputisobtained as:

$W_{gt} = m_a \cdot (h_5 - h_6) + m_a \cdot (h_7 - h_8)$ (8)

Gas turbine cycle efficiencycan beexpressedas:

$\eta_{gt} = \frac{Wgt - Wc}{Q_s}$ (9)

3.3. Heliostat field, Central receiver and molten salt

Heat received on Heliostat field can be expressed as:-

$Q_s = I_n.A$ (10)

$Q_{in} = Q_{abs}+Q_{loss}$ (11)

Heat absorbed by the molten salt expressed as:

$Q_{abs}= m_{sa}C_{psa.}(T_{sa,out}–T_{sa,in})$ (12)

Heat loss in the central receiver can be expressed as -

$Q_{loss} = Q_{loss,cond}+ Q_{loss,conv}+ Q_{loss,ems}+ Q_{loss,ref}$(13)

Receiver's efficiency is given by-

$\eta_r = \frac{Q_{abs}}{Q_{in}}$ (14)

Heliostat efficiency can be calculated as-

$\eta_h= \frac{Q_{in}}{Q_s}$ (15)

3.4. HRSGand steam Rankine cycle

In HRSG, the energy balance yields:

$m_a.\ (h_8 – h_9) = m_{w.}\ ((h_{10} – h_{18}) + (h_{11} – h_{12}))$ (16)

Here perfect reheating consideration yields,

$T_{10} = T_{12}$ (17)

HPST work output as-

$W_{HPST} = m_w . (h_{10} - h_{11})$ (18)

LPST work output as

$W_{LPST} = mw.(h_{12} – h_{13}) +m_w.(1-f_r).(h_{13} – h_{14})$ (19)$W_{pump} = \int vdp = h_{18} - h_{17}$ (20)

$W_{cep}=h_{16} - h_{15}$ (21)

$Q_{Cond.} = m_w.(1 - fr).(h_{14} - h_{15})$ (22)

The energy balance in the deaerator is given by:

$m_w.h_{17}= m_w.f_r.h13 + m_w.(1-f_r).h_{16}$ (23)

Efficiency of steam turbine cycle:

$\eta_{ST}= \frac{W_{ST}}{Q_{exhaust}}$ (24)

$Q_{exhaust} = m_a\ (h_8 - h_9)$ (25)

$W_{ST} = W_{HPST} + W_{LPST} - W_{pump} - W_{cep}$ (26)

$Q_{ST}= m_w\ ((h_{18} – h_{10}) + (h_{12} – h_{11}))$ (27)

3.5.HRVG and organic Rankine cycle

HRSG's exhaust air goes to the HRVG for energyutilizationin ORC vapour turbine having refrigerant R1233zd (E). The energy balance in the HRVG expressed as-

$\eta_{HRVG} . m_a . c_p(T) . (T_9 - T_1) = m_r . (h_{28} - h_{31})$(28)

ORCvapour turbine's work outputis obtained as-

$W_{T,ORC} = m_r.(h_{28} - h_{29})$ (29)

Heat extracted in ORC condenser is obtained as;

$Q_{cond.,ORC} = m_r.(h_{29} - h_{30})$ (30)

The pumping work requirement can be given as;

$W_{P,ORC} = m_r.(h_{31} - h_{30})$ (31)

Efficiency of organic Rankine cycle

$\eta_{orc} = \frac{W_{T,ORC} - W_{P,ORC}}{h_9 - h_1}$ (32)

3.6. Vapour Absorption Refrigeration cycle

Heat from the intercooled compressor for generator of VARS is given by-

$Q_g = h_2-h_3$ (33)

Refrigeration effect of the evaporator per kg of refrigerant is given by-

$Q_{evap} = h_{21} - h_{20}$ (34)

The coefficient of performance of VARS is given as

$COP = \frac{Qevap}{Qg}$ (35)

3.7. Overall Efficiency of the system

The overall work output from the combined cycle comes from the intercooled – reheat type Brayton cycle, reheat steam Rankine cycle and organic Rankine cycle. The consideration for the heat rejected in the condensers of the bottoming cycles yields the overall work output from the considered cycle as shown ahead;

$W_{overall} = W_{net\ of\ gas\ turbine} + W_{ST} + W_{ORC} + Q_{Cond.,ORC} + Q_{Cond.}$ (36)

The overall efficiency of the considered combined cycle-

$\eta_{overall} = W_{overall}/ Q_s$ (37)

3.8Economic analysis

An economic analysis has been carried out for analyzing the economics of the considered. The cost of unit carbon free power and cooling has been obtained as;

Cost of unit carbon free power and cooling = Total cost/$W_{overall}$

Table 1. Cost details of the considered cycle [4, 13, 14, 15, 16 and 17]

Elements	Cost, in US$
Heliostat component cost	1107/m^2
Installation and checkout cost	175/m^2
Life cycle cost	134/m^2
Air compressor capital cost	$CC_{A,COP}$ =((39.5m_a)/(0.9-$E_{poly,comp}$))(P_2/P_1)ln(P_2/P_1) CC_{GT} = (-98.328 lnW_{GT}+ 1318.5W_{GT}
Gas turbine capital cost	CC_{GT} = (-98.328 lnW_{GT}+ 1318.5W_{GT})
HRSG	$4745(\frac{Q_{HRSHG}}{\log \Delta T_{HRSG}})^{0.8}$ + 11820 m_w +658m_g
Steam turbine	6000$W_{ST}^{0.7}$
Steam Condenser	1773m_w
Feed water pump	3540$W_{FWP}^{0.71}$
Organic Rankine cycle capital cost	2345000($W_{net,ORC}$/1115kW
Absorption cooling system	10/kW

4. Results and discussions:

Results are obtained from the thermodynamic modeling and computer simulation of the air powered solarized intercooled-reheat GT cycle combined with steam Rankine cycle and organic Rankine cycle for carbon free power and cooling based on the input parameters given in Table 2.

Table 2. Input Parameters [2, 5, 8, 10 and 12]

Parameter, Unit	Value
I_n , Wm^{-2}	1000
Heliostat area, m^2	16,666
Heliostat efficiency,%	75
Receiver efficiency,%	80
Polytropic efficiency of compressor,%	90
Polytropic efficiency of turbine, %	92
T_{amb}., K	283, 288, 293
P_{amb}, bar	1
Cycle pressure ratio	8,10,12,14,16,18
P_{HPST}, bar	90
P_{LPST}, bar	5
Condenser pressure, bar	0.07
Bleed pressure, bar Isentropic efficiency of steam turbine	1.2 80
Inlet pressure of ORC vapour turbine, bar	8.336
Inlet temperature of ORC vapour turbine, K	388
Condenser temperature of ORC, K	288
Strong solution temperature,	40^{o}C
Weak solution temperature, ^{o}C ORC Refrigerant fluid	156 R1233zd(E)

Figure 2illustrates the variation of overall efficiency of the cycle at 90 bar steam generation pressure with cycle pressure ratio at different ambient temperatures which depicts that the overall efficiency of the combined cycle decreases with the increase in the cycle pressure ratio at each ambient temperature. Also, at any cycle pressure ratio, the increase in ambient temperature offers the increase in the overall efficiency. The maximum overall cycle efficiency is found to be 60.24% at cycle pressure ratio of 8 and ambient temperature of 293 K.

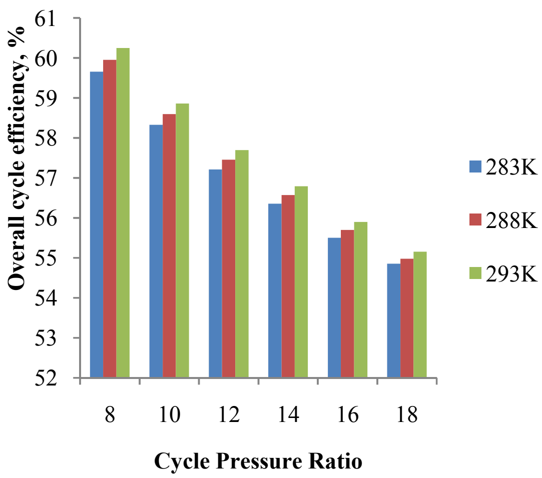

Figure 2: Variation of overall efficiency of cycle at 90 bar with cycle pressure ratio at different ambient temperatures

The effect of varying the cycle pressure ratio on the e efficiency of bottoming cycles namely steam Rankine cycle & ORC and the overall combined cycle efficiency at optimum conditions of steam generation pressure 90 bar, steam generation temperature 715 K and steam bleeding fraction of 9.8 % is shown in Figure 3. It is seen that while the overall cycle efficiency exhibits the decreasing trend with the increase in cycle pressure ratio the steam Rankine cycle efficiency decreases while the overall combined cycle efficiency decreases and the gas turbine cycle efficiency increasesHowever, the ORC efficiency is constant

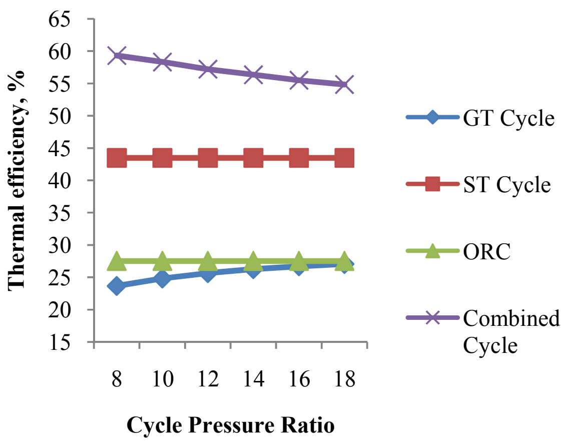

Figure 3. Variation of thermal efficiency with cycle pressure ratio

Figure 4 shows the variation of cooling capacity available from VARS with the increase in topping cycle pressure ratio at different ambient temperatures. It details that as the cycle pressure ratio is increases, for any ambient temperature the cooling capacity of VARS is increasing. Similarly at any cycle pressure ratio, with the increase in ambient temperature, the cooling capacity available from VARS also increases.

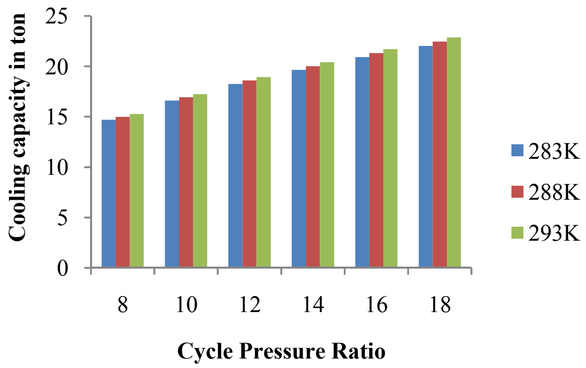

Figure 4. Variation of cooling capacity with cycle pressure ratio

Figure 5 shows the variation of overall cycle efficiency and cooling capacity with cycle pressure ratio. It depicts that as the cycle pressure ratio increases the cooling capacity also tends to increase whereas overall cycle efficiency is decreasing. Graphically at the point of intersection of the variation of overall cycle efficiency and cooling capacity referring to the tradeoff between the power generation and cooling capacity yields, the optimum values for a combined power and cooling cycle at 56.78% overall cycle efficiency and 20.4 tons of refrigeration at 293K ambient temperature, 90 bar steam generation pressure, 715 K steam generation temperature and 14 cycle pressure ratio.

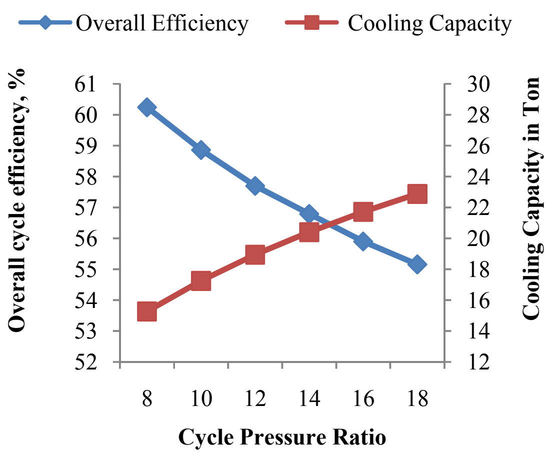

Figure 5: Variation of Overall cycle efficiency and cooling capacity with cycle pressure ratio

4. CONCLUSIONS:

Following conclusions have been obtained from the thermodynamic investigation and economic analysis of the considered cycle.

1. There is absence of combustion which does not yield any emission in generation of power along with cooling effect. Availability of cooling effect of 0.00021 ton per MW/kg of air using the heat liberated in intercooling in gas turbine cycle is a greater advantage.
2. Maximum overall cycle efficiency of 60.56 % is achieved with lower cycle pressure ratio of 8 and at higher ambient temperature of 293K.
3. The increase in ambient temperature from 283 K to 293 K leads to an increase in the overall efficiency of the combined cycle, and Cost per unit carbon free power withcooling is 4904 US$/kW.

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