Parametric evaluation of solar driven combined supercritical organic Rankine cycle and vapor absorption refrigeration cycle for trigeneration

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Abstract. Current study deals with the parametric evaluation of combined supercritical organic Rankine cycle and vapour absorption refrigeration cycle driven by solar power tower. It was obtained from the results, exergy and thermal efficiency of the combined system improved with solar irradiation. Maximum thermal and exergy efficiency were obtained 46.60% and 68.25% respectively at 950 W/m² while maximum exergy destruction was obtained 7589.46 kW at 500W/m². COP of the system decreased with generator and condenser temperature. The maximum COP for heating and cooling were found 1.4452 and 0.4448 respectively at 90℃ of generator temperature.

Keywords: Parametric analysis, solar power tower, ORC, vapor absorption refrigeration cycle, cogeneration

1. Introduction

Due to industrial expansion and population growth, world energy demand is increasing on a regular basis [1, 2]. Consequently, the use of fossil fuels is constantly growing, contributing to higher carbon emissions. Fossil-fuel stocks are declining slowly, however. This leads to the challenge of building energy options that are safe and secure [3, 4]. Geothermal, solar, wind and biomass are the various renewable energy options currently being used to generate clean and environmentally friendly electricity. Among these unconventional energy resources, solar energy is more appropriate for production of power due to its availability, low cost and noise-free operation [5].

For cooling, heating, and producing electricity, solar collectors are used to absorb sunlight [6]. Solar collectors were tested by Tian and Zhao [7] and found that at low temperature (less than100℃) the flat plate collectors can be used. Concentrated solar collectors are, however, used for high temperature applications, such as parabolic trough collectors, dish collectors, heliostats, etc. Kalogirou [8] reported the working temperature range of the parabolic trough collector (60-500℃), dish collector (750-1000 ℃), and heliostat (150-1500 ℃). Concentrated solar power (CSP) technology plays a key responsibility in solving the current and future problem of power generation in tropical countries by using solar heat that is readily available in nature, among other solar technologies [9].

Solar power tower (SPT) is the latest technology among the different CSP technologies. The SPT system groups of a number of complex sub-systems such as a receiver, a 75-150 m high tower, a thermal storage system (optional), a 50-150 m² heliostat field and a power conversion system per heliostat area. Solar radiation focuses on heliostat field receivers where it has been used to produce high temperature heat for power production through a high energy loop or for the supply of industrial processes [10].
Several studies were performed on the SPT driven cycles such as combined recompression $\text{sCO}_2$ cycle and trans-critical $\text{CO}_2$ cycle [11], $\text{sCO}_2$ Brayton cycle [12], recompression $\text{sCO}_2$ with and without main compressor inter cooling [13], triple combined cycle [14], multi generation combine cycle [15].

Singh and Mishra [16] evaluated the thermodynamic performance of the solar operated supercritical organic Rankine cycle (SORC) system. They concluded that exergetic parameters were found such as irreversibility ratio, fuel depletion ratio, improvement potential, 0.9296, 0.579 and 11859 kW respectively. Among other selected fluids, $\text{R600a}$ was the best fluid. Wang and Dai [17] used the waste heat was used from the $\text{sCO}_2$ recompression cycle by ORC to produce electricity and also carried out an exergo-economic study using $\text{R245fa}$, isobutene, $\text{R123}$, cyclohexane, toluene and isopentane as an ORC working fluid. In the end, they found that the optimal product cost of combined ($\text{sCO}_2$/ORC) cycle was much lower. Kalra et al. [18] studied that the SORC operating fluid is above and below its critical pressure and found that over its subcritical ORC, the SORC has various benefits as its SORC heat source cooling curve aligning with working fluids heating curve. Furthermore, Chen, Goswami and Stefanakos [19] reported that the SORC's heating cycle did not move through the same double phase region as in the subcritical ORC, results in excellent thermal matching in the evaporator and thus less irreversibility in the SORC. Pan, Wang, and Shi [20] found in various studies that SORC frequently selected when the critical temperature of working fluid is exceptionally less than the source heat temperature.

Apart from this many researchers are associated with the cogeneration system. Several studies were conducted on the combined ORC-VCC system recent years. Pektezel and Acar [21] performed thermal evaluation of the combined ORC and single and double evaporator VCC system. They found that $\text{R600a}$ was considered as the best fluid for the combined system. They also found that the COP and combined system’s exergetic efficiency with a single evaporator were more than that of a dual evaporator. Saleh [22] carried out thermal evaluation of the ORC integrated VCC system. He concluded that $\text{R602}$ was an acceptable working fluid for system performance and environmental considerations. Finally, he concluded that the system ‘s highest COP, exergy efficiency and expansion ratio using $\text{R602}$ were 0.99, 53.8% and 12.2 respectively at 25 °C temperature of condenser and maintaining other parameters constant.

Javanshir et al. [23] performed thermodynamic and exergo-economic analyzes on the regenerative ORC-VCC system. Their results indicated that among other selected working fluids, the $\text{R134a}$ gave the lowest exergy and thermal efficiencies while $\text{R143a}$ and $\text{R22}$ showed the highest exergy and energy efficiency respectively. The maximum exergy destruction rate was found in the boiler followed by turbine. Hu et al. [24] performed a study of the ORC-VCC system operated by solar power. They concluded that the ice production and the cooling power depend on the condensation and generation temperature. $\text{R245fa}$ was selected as appropriate fluid for this purpose. Using $\text{R245fa}$ cooling power and ice production per unit meter square collector area were 126.44W/m² and 7.61 kg/m²-day respectively. Moles et.al [25] performed a study on low temperature heat activated combined ORC-VCC system. They estimated that the variation of the COP was found between 0.30 and 1.10, while the ORC-VCC system's electrical COP ranged from 15 to 110. Additionally, HFO-1234ze(E) was considered an acceptable working fluid for enhanced efficiencies. Li et.al [26] performed a working fluids selection study for the combined ORC-VCC system. They concluded that for ORC–VCC system butane was chosen as the best refrigerant, with temperature variations ranging from 60 to 90 °C, −15 to 15 °C and 30 to 55°C at boiler exit temperature, evaporation temperature and condensation temperatures varying, respectively. Based on butane the COP of the whole system was observed as 0.47 at boiler exit temperature of 90°C.

In some studies, in integrated cooling, heating and power production systems, vapour absorption cooling (VAR) systems have been used in addition to VCC. In this direction Li et al. [27] performed a comparative analysis focused on single and multi-objective optimization methods between the combined LiBr-water/$\text{sCO}_2$ and NH$_3$-water/$\text{sCO}_2$. Their findings show that the LiBr-water/$\text{sCO}_2$ model shows a refrigeration COP gain of 0.3112 compared to LiBr-water/$\text{sCO}_2$ vapor. A recent combined $\text{sCO}_2$ and VAR recompression cycle parametric and exergo-economic analysis was performed by Wu et al. [28]. They find that there is maximum exergy loss in the reactor and there is minimal exergy loss in the components of the VAR cycle. Finally, their results of exergo-economic optimization indicate that the
combined system gave 26.12% and 2.73% greater thermal and exergy efficiency compared to sCO₂ cycle, respectively.

It has been noted from the above-mentioned literature section that different research were carried out on a cogeneration system for combined heating, cooling, and power generation. However, no study was conducted on the combination of supercritical ORC and VAR powered by concentrated solar power.

Therefore, the objective of the present research is to evaluate the performance of the solar power tower-operated combined ORC-VAR. The effect of solar irradiation, topping and bottoming cycle constraint has been further analyzed on combined system performance. The solar system's analysis and architecture is outside the reach of the present research.

2. System description

This model consists of a SORC power system and a VAR system for combined cooling and heating application as shown in figure 1. This whole combine system is driven by solar power tower system. In solar circuit heat transfer fluid (HTF) (molten salt mixture) is flowing. Working fluid (toluene) flowing in the SORC takes heat from HTF through evaporator 1 (EV1) (process 8 to 1). This heated fluid stream expanded in the turbine (process 1 to 2). Still stream have high temperature passing through recuperator (process 3 to 4) where it is absorbed by low temperature stream. This stream then goes to generator where this gives the heat to VAR cycle (process 4 to 5). Passing through the condenser (process 5 to 6) it goes the pump 1 (process 6 to 7). This low temperature stream of working fluid is passing through the recuperator (process 7 to 8) to get the heat from high temperature stream. Again, it goes to the EV1 then cycle repeats again.

Now come across the VAR cycle consist ammonia and water fluid system. Ammonia works as the coolant while the water works as the absorbent to produce cooling and heating effects. After taking the heat through the evaporator 2 (EV2) to produce the cooling effect, ammonia vapor having low pressure enter in to the absorber where ammonia vapor absorbed by cold water (process 9 to 10). A solution called aqua-ammonia is produced here. The ammonia vapour water absorption would reduce the absorber pressure, evaporator much vapor has been drawn, thus increasing the temperature of solution. Water cooling systems have been used to eliminate the heat generated in the absorber and improve the ability to absorb water. Subsequently, a strong absorber solution is injected into a solution heat exchange unit (SHE) (process 12-13), where this efficient solution is heated by a weak solution from the generator (state 13). Then the strong solution enters the generator (state 12) where it is activated by heat transferred from the SORC working fluid (toluene) and heated at high pressure when heated, The solution removes ammonia vapour from the hot, weak ammonia solution that trickles down through the pressure-reducing valve instantly (process 14-15). Then, high-pressure ammonia vapour enters the condenser and induces heat effects (process 15 to 16) and converts liquid ammonia into high-pressure ammonia. After passing through the expansion valve (process 17-18), low pressure liquid ammonia enters the evaporator. The design working parameters of the SPT model, the SORC and the VAR cycle are listed in table 1.

3. Thermodynamic approach

3.1 Assumptions for simulation

(1) All system components are under conditions of steady state. (2) Friction and heat loss are ignored in piping. (4) Heliostat and the receiver parameters have remained constant and the assumed input data are given in table 1. (5) Temperature molten salt to the inlet at EV1 was taken as 700°C [29]. (6) Refrigerant after the evaporator and condenser should be the saturated vapor and saturated liquid respectively [28]. (7) Strong and weak solution after the generator and absorber considered as saturated liquid [28].
**Figure 1.** Schematic diagram of integrated SPT combined SORC-VAR system.

**Table 1.** Input parameters for simulation of the proposed model

| Geometric and operating parameters for SPT |  |
|-------------------------------------------|------------------------------------------------|
| Direct normal irradiation                 | 970 W/m$^2$ [11] |
| Sun temperature                           | 4500 K [30] |
| Solar’s multiple                          | 2.8 [11] |
| Efficiency of heliostat                   | 58.71 % [35] |
| Number of heliostat                       | 141 [32] |
| Heliostat’s total mirror area             | 9.04×7.89 [11] |
| Initial temperature difference            | 15 K [11] |
| Solar receiver’s temperature approach     | 423.15 K [35] |
| Concentration ratio                       | 900 [35] |
| Convective heat loss coefficient          | 10 W/m$^2$-K [35] |
| Tower height                              | 74.62 m [32] |
| Convective heat loss factor               | 1 [35] |
| View factor                               | 0.8 [35] |
| Absorptance                               | 0.95 [35] |
| Thermal emittance                         | 0.85 [35] |

**Combined cycle’s input data**

| Parameter                                           | Value                        |
|-----------------------------------------------------|------------------------------|
| Maximum pressure                                    | 25 MPa [16]                  |
| Maximum temperature                                 | 470 °C                       |
| Turbine’s isentropic efficiency                     | 0.87 [16]                    |
| Pump’s isentropic efficiency                        | 0.85 [16]                    |
| Heat exchanger effectiveness                        | 0.95 [34]                    |
| Recuperator effectiveness                           | 0.95 [34]                    |
| Mass flow rate in SORC                               | 1.5 kg/s                     |
| VAR cycle maximum pressure                          | 1.08 MPa [31]                |
| VAR cycle minimum pressure                          | 0.14 MPa [31]                |
| Effectiveness of SHE                                 | 0.73 [33]                    |
3.2 Thermal modeling of SPT

Thermal modeling equations of the proposed system were developed in this part based on the conservation of exergy and energy equations, taking into consideration of assumptions those are made in above section. Direct solar heat incidence upon heliostat field is defined as [2, 11]

\[ \dot{Q}_{\text{solar}} = DNI \cdot A_h \cdot N_h \]  

(1)

Where, DNI is the solar irradiation (W/m²), \( A_h \) is single heliostat area (m²) and \( N_h \) is the heliostats number. However, due to heliostat efficiency, some of that heat is lost in the surroundings. The quantity of actual heat obtained through the heliostats is therefore specified as [2, 11];

\[ \dot{Q}_h = \dot{Q}_{\text{solar}} \cdot \eta_h \]  

(2)

Where, \( \eta_h \) is the efficiency of heliostat. This amount of heat is directed to the solar receiver where the heat transfer fluid flows. But a some amount of the heat is lost in the atmosphere. The heat available at the solar center receiver is therefore determined as [2, 11];

\[ \dot{Q}_r = \dot{Q}_h \cdot \eta_r = \dot{Q}_h - \dot{Q}_{\text{loss}} \]  

(3)

Where, \( \eta_r \) is the receiver thermal efficiency, is defined as [11];

\[ \eta_r = \alpha - \frac{\zeta \cdot f_{\text{view}} \cdot \sigma \cdot T_R^4 + h_{\text{conv}} \cdot f_{\text{conv}} \cdot (T_R - T_{\text{air}})}{G_{\text{in}} \cdot \eta_h \cdot CR} \]  

(4)

Where, \( T_R \) is the solar receiver’s surface temperature and CR is concentrated ratio. \( \zeta \) is the solar emittance. To calculate heat loss, this can be approximated as [11];

\[ T_R = T_i + \delta T_R \]  

(5)

Where, \( T_i \) is maximum temperature of cycle and \( \delta T_R \) is approach temperature of solar receiver.

Furthermore, control volume exergy balance equation is determined as [37];

\[ \sum \left( 1 - \frac{T_S}{T_j} \right) \dot{Q}_j - W_{\text{C,V}} - \sum (\dot{m}_j E_j) - \sum (\dot{m}_j E_{\text{e},j}) - E_D = 0 \]  

(6)

Where, \( E_D \) is the exergy destruction rate and subscript j refers to thermal property at particular state. Solar exergy inlet to the combined system is determined as [11];

\[ $E_{\text{solar}} = \left( \frac{Q_0}{\eta_h \eta_r} \right) E_s \]  

(7)

Where, \( E_s \) is the dimensionless useful maximum work obtained from the solar irradiation. \( E_s \) is expressed as [11];

\[ E_s = 1 + \frac{1}{3} \left( \frac{T_s}{T_m} \right)^4 - \frac{4}{3} \left( \frac{T_s}{T_m} \right) (1 - \cos \beta)^{1/4} \]  

(8)

Where, \( T_{\text{su}} \) and \( T_{\text{b}} \) are the sun and reference temperature respectively. \( \beta \) is the sun’s disc subtended half cone angle. Its value has been taken 0.005rad on solar energy limiting efficiency [36]. Further, in the receiver, useful exergy obtained by the molten salt is defined as

\[ E_r = \dot{m}_{\text{ms}} \cdot C_{p \text{ms}} \cdot \left[ (T_B - T_a) - (T_0 \cdot \ln \left( \frac{T_B}{T_a} \right) \right] \]  

(9)
3.3 Thermal modeling of the combined SORC-VAR system

Heat received by the combined cycle from the SPT field is given by the heat balance equation in the evaporator 1:

\[
\dot{Q}_r = \dot{Q}_{EV1} = \dot{m}_{ms} \cdot C_{p_{ms}} \cdot (h_b - h_a) = \dot{m}_{SORC} \cdot (h_1 - h_b)
\]  \hspace{1cm} (10)

Heat transfer from the evaporator-2 is determined as:

\[
\dot{Q}_{EV2} = [\dot{m}_{NH3} \cdot h_9 + \dot{m}_W \cdot h_{19}] - [\dot{m}_{NH3} \cdot h_{18} + \dot{m}_W \cdot h_{20}]
\]  \hspace{1cm} (11)

NH3 mass flow rate in evaporator-2 is determined by:

\[
\dot{m}_9 = \frac{\dot{Q}_{EV2}}{(h_9 - h_{18})}
\]  \hspace{1cm} (12)

Material and mass balance in the absorber tube given by:

\[
\dot{m}_9 + \dot{m}_{15} = \dot{m}_{10}
\]  \hspace{1cm} (13)

And

\[
\dot{m}_9 \cdot X_9 + \dot{m}_{15} \cdot X_{15} = \dot{m}_{10} \cdot X_{10}
\]  \hspace{1cm} (14)

Further heat rejection in the absorber and condenser has been used heating applications. Therefore, heat rejection in condenser and absorber can be defined as:

\[
\dot{Q}_{Absorber} = [\dot{m}_9 \cdot h_9 + \dot{m}_{15} \cdot h_{15}] - [\dot{m}_9 \cdot h_{10}]
\]  \hspace{1cm} (15)

\[
\dot{Q}_{Condenser} = [\dot{m}_{16} \cdot h_{17} - \dot{m}_{17} \cdot h_{17}]
\]  \hspace{1cm} (16)

Heat lost in PRV and EXV can be determined as:

\[
\dot{Q}_{PRV} = \dot{m}_{14} \cdot (h_{14} - h_{15})
\]  \hspace{1cm} (17)

\[
\dot{Q}_{EXV} = \dot{m}_{17} \cdot (h_{17} - h_{18})
\]  \hspace{1cm} (18)

Heat exchanger in generator is given by:

\[
\dot{Q}_{Generator} = [\dot{m}_{16} \cdot h_{16} + \dot{m}_{12} \cdot h_{12}] - [\dot{m}_{13} \cdot h_{13}]
\]  \hspace{1cm} (19)

Further chemical exergy of the system is constant throughout. After neglecting energy due to velocity and height, specific physical exergy at \(j^{th}\) point is defined as [1, 2,11]:

\[
E_j = (h_j - h_0) - T_0(s_j - s_0)
\]  \hspace{1cm} (20)

For cooling purpose exergy available in EV2 is determined as [31]:

\[
\dot{E}_{Cooling} = \dot{Q}_{EV2} \cdot \left[1 - \left(\frac{T_0}{T_{EV2}}\right)^{\gamma-1}\right]
\]  \hspace{1cm} (21)

Further destruction of the exergy in each part has been determined. Exergy destruction in the EV1 is defined as:

\[
\dot{m}_{SORC} \cdot [(h_b - h_1) - T_0(s_a - s_b)] + \dot{m}_{ms} \cdot [(h_a - h_b) - T_0(s_a - s_b)] - ED_{EV1} = 0
\]  \hspace{1cm} (22)

Exergy balance equation for the EV2 can be defined as:

\[
\dot{m}_{NH3} \cdot [(h_9 - h_{18}) - T_0(s_9 - s_{18})] + \dot{m}_W \cdot [(h_{19} - h_{20}) - T_0(s_{19} - s_{20})] - ED_{EV2} = 0
\]  \hspace{1cm} (23)

Exergy destruction in absorber is defined as:
Exergy destruction in condenser 2 is expressed as;
\[ ED_{\text{Cond2}} = \left[ (E_{17} - E_{16}) - (E_{24} - E_{23}) \right] \]  
(25)

Exergy destruction in SHE expressed as;
\[ ED_{\text{SHE}} = \left[ (E_{13} - E_{14}) - (E_{12} - E_{11}) \right] \]  
(26)

Exergy destruction in PRV and EXV defined as;
\[ ED_{\text{PRV}} = \left[ E_{14} - E_{15} \right] \]  
(27)
\[ ED_{\text{EXV}} = \left[ E_{17} - E_{18} \right] \]  
(28)

3.4 Proposed system’s performance parameters

On the basis of the thermal modeling, following performance parameters were discussed;

Net power output obtained from the combined cycle is defined as;
\[ W_{\text{net}} = W_{\text{Turbine}} - W_{\text{Pum}} - W_{\text{Pump2}} \]  
(29)

Combined system’s thermal efficiency is determined as;
\[ \eta_{\text{Thermal}} = \frac{W_{\text{net}}}{Q_{\text{sol}}} \]  
(30)

Combined system’s exergy efficiency is defined as [28];
\[ \eta_{\text{Exergy}} = \frac{W_{\text{net}} + E_{\text{Cooling}}}{E_{\text{solar}}} \]  
(31)

Combined cycle exergy efficiency is determined as [6, 37];
\[ \eta_{\text{Exergy}} = 1 - \frac{ED_{\text{Total}}}{E_{\text{solar}}} \]  
(32)

Combined cycle’s thermal efficiency can also be defined by the relation between combined cycle’s exergy and thermal efficiency [6, 37];
\[ \eta_{\text{Thermal}} = \eta_{\text{Exergy}} \cdot \eta_{\text{Carnot}} \]  
(33)

At last COP for cooling is defined as [33],
\[ \text{COP}_{\text{Cooling}} = \frac{Q_{\text{EV2}}}{Q_{\text{Generator}} + W_{\text{Pump2}}} \]  
(34)

Pump work is very less as compared to the \( Q_{\text{Generator}} \). Therefore Pump 2 work can be neglected. Therefore, COP for cooling can also be defined as [31];
\[ \text{COP}_{\text{Cooling}} = \frac{Q_{\text{EV2}}}{Q_{\text{Generator}} + W_{\text{Pump2}}} \]  
(35)

Rejection of the heat in condenser and absorber can be considered as heating purpose. Therefore, heating COP of the system can be defined as [31, 33];
\[ \text{COP}_{\text{Heating}} = \frac{Q_{\text{Condenser}} + Q_{\text{Absorber}}}{Q_{\text{Generator}}} \]  
(36)

3.5 Working fluids selection

When choosing the working fluid for every thermodynamic cycle, caution must be taken because it impacts the performance of the cycle, economic viability and environmental aspects [38]. A mixture of
magnesium dichloride (MgCl₂) and potassium chloride was used in the receiver as molten salt HTF with a mass fraction of 32% and 68% respectively in the receiver [11]. The reason behind choosing this HTF is that, compared to liquid sodium (Na) and solar salt, this is the cheapest alternative for the SPT driven cycle. Molten salt’s thermophysical data are listed in table 2 [39].

Further, toluene was selected as working fluid in current study for SORC due to its high temperature stability 480℃ [40]. Organic Rankine cycle works above its critical temperature of the working fluid because the critical temperature of toluene is 318.55℃ [40] which is less than the maximum temperature of the combined system.

Table 2. Thermo-physical properties of molten salt (magnesium dichloride + potassium chloride) [41].

| Parameters                  | Values               |
|-----------------------------|----------------------|
| Density                     | 1593 (kg/m³)         |
| Solidification temperature  | 699 K                |
| Thermal conductivity        | 0.39 (W/m-K)         |
| Stability limit             | 1691 K               |
| Specific heat               | 1.028 (kJ/kg-K)      |

3.6 Validation of the model
To check the correctness of the modeling equations for the current system the model has been validated to the existing studies. SORC has been validated with previous study Singh and Mishra [16]. Thermal efficiency is considered as the validation parameter considering same baseline conditions. Table 3 displays that there is very less % deviation in efficiency with reference. Therefore, this model for analysis is acceptable. Bottoming VAR cycle is also validated with the previous study Gupta et al. [33]. It is seen in table 4 that VAR has good agreement with the reference in terms of COP. Therefore, this model is ready for further analysis.

Table 3. Validation of the SORC system

| Baseline conditions in reference | Thermal efficiency with Isobutene Singh and Mishra [16] | Thermal efficiency of Current model | Deviation |
|---------------------------------|--------------------------------------------------------|-----------------------------------|-----------|
| T₁ = 390(℃), P₁ = 25MPa, η₁Turbine = 0.87, η₁pump = 0.85 | 50.89%                                                  | 48.13%                             | -5.42%    |

Table 4. Validation of the VAR system

| Baseline conditions in reference | Reference                | COP<sub>Cooling</sub> | COP<sub>Heating</sub> |
|---------------------------------|--------------------------|------------------------|------------------------|
| T₁₄₆ = 94.95℃, T₁₇ = 34.95℃, T₈ = 4.95℃, T₉ = 4.95℃ | Gupta et al. [33]        | 0.4042                 | 1.404                  |
|                                | Current model            | 0.3825                 | 1.3974                 |
4. Result and discussion
Exergy and energy analysis of concentrated solar power based combined SORC and VAR was performed in the current study. In the EES software, a numerical computer program is designed to simulate a model. The impact of the various variables on system performance has been investigated. Impact of independent parameters such as solar irradiation or direct normal irradiation (DNI), generator temperature, inlet temperature and pressure of turbine, condenser and absorber temperature on dependent variable such as thermal and exergy efficiency, COP for cooling and heating have been carried out.

4.1 Performance evaluation with solar irradiation
Figure 2 shows exergy and the combined device thermal efficiency increases with solar radiation. As solar irradiation increases, the combined cycle's exergy and thermal efficiency increases because of the effective use of solar irradiation in the solar receiver [1, 2]. As solar irradiation rises from 500 to 950 W/m², thermal and exergy efficiency grow from 35.69 to 46.60% and 48.67 to 68.25%, respectively. Nevertheless, with solar irradiation, the overall exergy loss rate of the combined cycle decreases. As solar irradiation rises from 500 to 950 W/m², it declines from 7589.46 to 3005.45 kW. It is visible on the right axis of Figure 2.

![Figure 2](image)

*Figure 2. Effect of solar irradiation on combined system performance.*

4.2 Performance evaluation with turbine inlet temperature
With the TIT, the thermal and exergy efficiency of combined cycle is increased. However, the rate of exergy destruction correspondingly decreases as shown in Figure 3. The explanation for improving the system's thermal and exergy efficiency is discussed as temperature improvement increases the exergy inflow which leads to increase turbine output, which improves combined cycle's thermal output improves. Alternatively, as the TIT rises, the difference in temperature increases and the heat rejection increases, this effect improves the performance of the combined cycle [42]. TIT increases from 350 to 470 °C, thermal and exergy efficiency increases from 44.96 to 45.61% and from 65.91 to 66.86% respectively. However, rate of exergy destruction decreases from 3308.09 to 3215.9 kW as shown in Figure 3.
The TIT also affects the cooling and heating performance of the system. COP heating and cooling decreases continuously with the TIT as shown in Figure 4. The explanation is that as TIT rises, the performance of the combined cycle improves, however the rate of heat in evaporator-2 decreases slightly, consequently cooling rate decreases. Cooling COP and heating COP decreases 0.4452 to 0.3353 and 1.4456 to 1.3355 respectively as TIT increases from 350 to 470 ℃.

4.3 Evaluation of system performance with turbine inlet pressure

Figure 5 indicates that the thermodynamic efficiencies of the combined system improve with the inlet pressure of turbine (TIP). The explanation behind this is that as the TIP raises difference in enthalpy between the turbine and the turbine, there is also more work output from the turbine, which contributes to an improvement in thermodynamic efficiencies [1]. As TIP increases from 8 to 16 MPa, thermal and exergy efficiency improves from 42.88 to 43.57% and 62.86 to 63.87% respectively, however rate of exergy destruction decreases from 4934.48 to 3371.16 kW respectively as shown in figure 5.
TIP also affects the COP of the system. COP of heating and cooling increases with the TIP as shown in figure 6. As TIP increases from 8 to 16 MPa, heating and cooling increases from 1.3353 to 1.4452 and 0.3357 to 0.4459 respectively.

There is also need to discussed the impact of bottoming cycle parameters on system performance such as generator temperature, condenser and absorber temperature.

4.4 Evaluation of system performance with generator temperature

Figure 7 shows that the system's exergy and thermal efficiency decreases marginally with the generator temperature. The efficiency of thermal and exergy decreases from 44.54% to 43.89% and 65.29% to 64.34% respectively however the exergy destruction rate increases accordingly with the generator temperature from 3371.16 to 4934.48kW, as temperature of generator increases from 90 to 130°C.

From Figure 8, the COP for heating and cooling is also shown to decrease with the temperature of the generator. As the temperature of the generator rises from 90 to 130 °C, COP of heating and cooling declines from 1.4452 to 1.3353 and 0.4497 to 0.3294, respectively. The temperature of the flowing refrigerant leaves the generator rises as the generator temperature rises, which may raise condenser and
absorber’s average temperature in the VAR cycle, resulting in further heat transfer losses or destruction of exergy at higher generator temperatures [31].

![Figure 7](image1.png)

Figure 7. System performance variation with generator temperature

![Figure 8](image2.png)

Figure 8. COPs variation with generator temperature

4.5 System performance evaluation with condenser and absorber temperature

Finally, the impact of condenser and absorber temperature on system performance was also investigated in this study. Both efficiencies of combined cycle decrease with the temperature of the condenser and absorber. The efficiency of thermal and exergy decreases by 1.42 and 2.12% respectively, as the temperature of the condenser and absorber increases from 40 to 70 °C. Exergy destruction rate, however, increases with condenser and generator temperature from 3371.16 to 4934.48 kW, as shown in figure 9. The system's heating and cooling COP also lowers with the temperature of condenser and absorber. The COP for heating and cooling decreases by 8.16 and 24.68 %, as the temperature increases from 40 to 70 °C, as shown in Figure 10. The generator pressure increases accordingly as the temperature rises, causing low amount of vapor of ammonia discharges from the generator [33]. As a result, exergy destruction will increase consequently system performance will decrease.
5. Conclusions

Following conclusions were made from results:

- Combined cycle’s thermal and exergy efficiency were increased with solar irradiation continuously however exergy destruction decreased.
- Maximum thermal and exergy efficiency were obtained 46.60% and 68.25% respectively at 950 W/m² while maximum exergy destruction was obtained 7589.46 kW at 500W/m².
- Exergy and thermal efficiency of the combined system also improved with TIT. However, COP for cooling and heating decreased simultaneously.
- Combined power production, heating and cooling performance improved with the TIP. Maximum thermal, exergy efficiency and COP for heating and cooling were obtained 42.88%, 63.87%, 1.4452 and 0.4448 respectively at 16MPa.
- Combined cycle system performance affected with bottom cycle parameters. Thermal and exergy efficiency, COP for heating and cooling decreased with the generator, condenser and absorber temperature.
- It was also concluded from the results that SORC and VAR system combined suitable for CSP applications for combined power generation, heating and cooling.
Thermo-economic analysis of the current study is required in future research.

**Nomenclature**

- $A_h$: Single heliostat area (m$^2$)
- $\dot{E}$: Rate of exergy (kW)
- $C_p$: Specific heat (kJ/kg-K)
- $f_{\text{view}}$: Receiver’s view factor
- $\text{DNI}$: Direct normal irradiation (W/m$^2$)
- $N_h$: Heliostat quantity
- $h_{\text{conv}}$: Coefficient for convective heat loss (W/m$^2$-K)
- $m$: Mass flow rate (kg/s)
- $h$: Specific enthalpy (kJ/kg)
- $Q$: Rate of heat (kW)
- $\dot{E}_{\text{Solar}}$: Solar exergy available at combined cycle inlet (kW)
- $Q_h$: Actual heat absorbed by heliostat field (kW)
- $Q_r$: Heat at central receiver (kW)
- $T_R$: Receiver surface temperature (K)
- $\dot{E}_{\text{Destruction}}$: Rate of exergy destruction (kW)
- $Q_{\text{Solar, Total}}$: Total heat absorbed by heliostat field from sun (kW)
- $s_{\text{CO}_2}$: Supercritical carbon dioxide
- $\dot{Q}_{\text{loss,r}}$: Receiver’s heat loss (kW)
- $T$: Temperature (K)
- $X$: Fraction of NH$_3$ in ammonia/water solution
- $\eta_h$: Heliostat efficiency
- $W$: Power (kW)
- $\eta_r$: Receiver thermal efficiency
- $s$: Specific entropy (kJ/kg-K)

**Abbreviations**

- CR: Concentration ratio
- Cond2: Condenser -2
- EV2: Evaporator -2
- ORC: Organic Rankine cycle
- Cond1: Condenser -1
- VAR: Vapor absorption refrigeration cycle
- EV1: Evaporator -1
- COP: Coefficient of performance

**Greek letters**

- $\alpha$: Solar absorbance
- $\epsilon$: Effectiveness
- $\beta$: Sun’s subtended cone half angle (rad)
- $\eta$: Efficiency
- $\sigma$: Stephen Boltzmann constant (W/m)
- $\zeta$: Thermal emittance

**Subscripts**

- $e$: Exit
- $i$: Inlet
- $r$: Receiver
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