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Thermodynamic Analysis of Solar-based Combined Power and Cooling Cogeneration Cycle

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Abstract

The performance of a solar-powered Rankine cycle that combines absorption refrigeration with solar thermal technology is examined in this communication. The research finds regions of irreversibility in the system and takes into account fluctuations in direct normal irradiance (DNI). In order to evaluate the cycle's performance at different DNI levels ranging from 600 to 1050 W/m², the research does a thermodynamic analysis of the cycle. It also assessed the irreversibility of each cycle's distinct parts, which resulted in potential performance-enhancing actions. Furthermore, the central receiver contributes 31.37% irreversibility to the cycle's exergy distribution, the heliostat contributes 27.4%, and other components like the HRSG, steam turbine, and VARs contribute the rest amounts. Only 25.43% of the cycle's energy production is useable.

Keywords

Heliostats, solar field, absorption refrigeration, evaporator, exergy destruction

1. Introduction

The world is experiencing a continuous increase in energy demand, primarily driven by rapid economic development. This surge in energy requirements poses challenges and opportunities for nations around the globe [1]. Renewable energy sources are recognized as a pivotal factor for the growth and prosperity of nations. Renewable energy sources offer a sustainable and environmentally friendly approach to meet energy demands [2, 3].

Solar energy is specifically highlighted as a promising and cost-effective option for energy production. It is abundant and naturally available, making it an attractive choice for generating valuable energy resources. Solar energy is not only promising but also economically and environmentally beneficial. Utilizing solar power can contribute to economic growth while mitigating environmental concerns associated with traditional energy sources. Moreover, it can be created as a viable alternative for nations like India where coal is the primary source of electricity [4, 5].

The simultaneous operating cycle for electric power generation and cooling purposes by a single source of energy is said to be cogeneration. Here, it means that the same energy source is used to provide both cooling and electricity. Cogeneration systems are designed to efficiently utilize waste or excess energy that would otherwise be lost. This often involves capturing the waste energy available from components like steam turbines or condensers [6 - 8].

According to the cogeneration system's energy analysis, a large portion of the energy released by it is discovered to be an irreversible source; thus, recovering this energy for use in the system is necessary to increase its effectiveness [8]. Chillers, refrigerators, and air conditioners are the largest consumers of electricity during the summer months at the same time solar energy is available maximum. When the cooling load is observed in large quantity then the absorption refrigeration machines will become a viable option and can replace the commercial power loads used inthe chillers and other cooling options. By the waste heat recovered from the cogeneration system and utilized to meet the cooling load of refrigerators and air conditioning, it is found that the system effectively reduced the grid peak load during the summertime. Cogeneration is the means by which primary energy is efficiently used. When different types of energy are needed, both residential and commercial applications can make use of the heat recovery and cogeneration system. It contributes to the nation's and the world's sustainable development and is a very effective energy-saving method [9, 10].

In this regard, a combination of the steam Rankine cycle and absorption refrigeration cycle was proposed by Parvez et al. [11] who noted that the cycle produced both the power generation and refrigeration effect from a solitary source of energy and found notable thermal efficiency. Dhahada et al. [12] studied a thermo-economics analysis of a cogeneration cycle in order to fulfill the demand for power and cooling. It used a combined cycle of Kalina and the absorption refrigeration cycle which is operated by a low-energy source. Mousaviet al. [13] investigated a solar-driven cogeneration energy system to produce cooling/heating in remote areas. It was used MATLAB, HYSYS, TRNSYS, and HOMER software for the simulation studies. Liu et al. [14] presented a study on waste heat-driven combined power and refrigeration energy systems in order to overcome not only the energy crises but also reduced CO2 emissions.

More recently, Zhang et al. [15] used the genetic algorithm approach to optimize the performance of a suggested cycle by parametric analysis. Their results showed that, compared to the cogeneration cycles and the Kalina cycle reported in the literature, the suggested cycle showed comprehensive power recovery efficiencies that were 58.11%, 23.85%, and 7% greater under the identical beginning circumstances.

The cogeneration cycle, which is powered by a solar tower, is the term for the energy and energy analysis of combined commercial power generation (Rankine cycle) and cooling that was evaluated in this study. The suggested cogeneration cycle is an attempt to use concentrated solar power (CSP) effectively to satisfy industry demands for cooling and electricity at the same time. It has been noted how certain parameters affect the exergy destruction and energy and energy efficiencies of the suggested cogeneration cycle.

2. The proposed system description and assumptions

As seen in Fig. 1, the investigated cogeneration cycle is made up of the absorption refrigeration cycle (ARC) and the steam Rankine cycle (SRC), both of which are powered by solar energy to create electricity and cooling at the same time. After entering the heat recovery steam generator (HRSG) and transferring heat to produce steam, the molten salt expands in the steam turbine. Additionally, once the turbine's exit steam has been condensed in condenser 1, it travels to the desorber to collect waste heat before entering the HRSG. The desorber's superheated water vapor was condensed in condenser 2 and transformed into a saturated liquid. Next, the high-pressure saturated liquid—available at condenser pressure—went through the throttle valve and produced the saturated liquid at lower pressure, or pressure inside the evaporator. The absorber has been receiving saturated vapour from the evaporator. After cooling in a heat exchanger, the LiBr-H2O mixed solution goes via a throttle valve to lower the pressure to the level of the absorber (Low Pressure). As seen in the illustration, two streams have been combined at the absorber to create a new combination. The new mixture has been passed through solution heat exchanger (SHX) and the pump, after SHX enters into desorber and the cycle is completed.

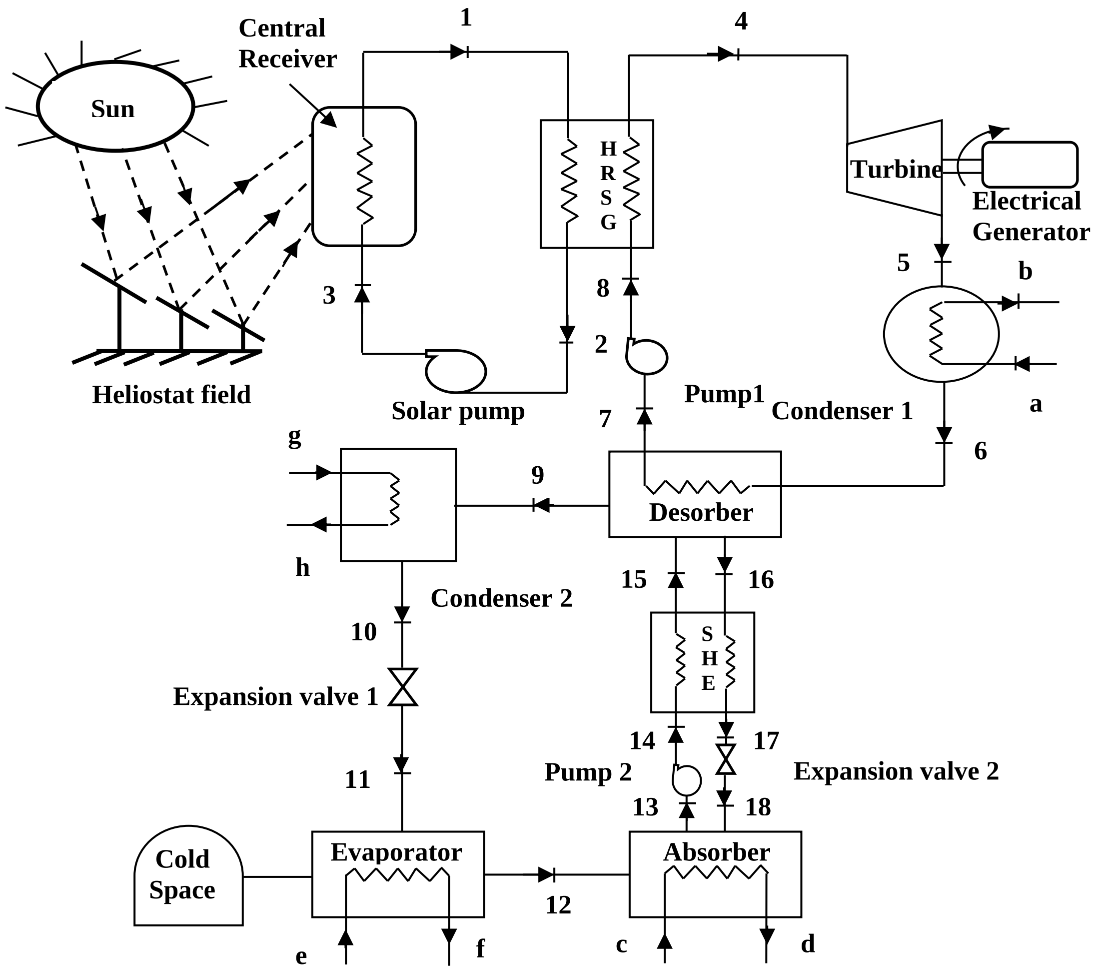

Figure1. Diagram of the suggested cycle for cogeneration of cooling and electricity

The suggested cycle has been investigated under the following assumptions:

- There may be pressure dips in the working pipes as the suggested cycle's components operate in a stable state with continuous solar radiation.
- The analysis ignores the heat losses to the atmosphere in the evaporator, condensers, steam turbine, and HRSG.
- It is discovered that the LiBr-H2O solution in the desorber and absorber is in an equilibrium state at the corresponding pressures and temperatures.
- Both the strong and the weak solutions exit the absorber and desorber, respectively.
- To prevent the solutions from crystallizing, the ARC requires that the solution entering the throttle valve be at least 70 to 80 degrees Celsius, which is above the temperature of crystallization.

3. Thermodynamic modeling of the proposed cogeneration cycle

The energy and exergy study considers both thermodynamics laws viz. first and second-law which can provide an opportunity to assess the analytical performance of the proposed cycle.

The rate of solar heat input $(\dot{Q}_{solar})$ to the heliostat field collectorcan be given as:

$$\dot{Q}_{solar} = A_{field}q \quad (1)$$

where, q is the solar radiation per unit area, which is direct normal irradiations (DNI), and Afield is the area of the heliostat field which depends on the aperture area (Aapp) and concentration ratio. Its concentration area can be calculated as:

$$C = \frac{A_{field}}{A_{app}} \quad (2)$$

$\dot{Q}_{solar}$ can be evaluated by:

$$\dot{Q}_{solar} = \dot{Q}_{CR} + \dot{Q}_{lost,heliostat} \quad (3)$$

where $\dot{Q}_{CR}$ is the solar radiations received by the central receiver (CR), and $\dot{Q}_{lost,heliostat}$ is designated by heat lost in the heliostat. The energy efficiency of heliostat can be defined as the ratio of output to the input energy and get as:

$$\eta_{energy,heliostat} = \frac{\dot{Q}_{CR}}{\dot{Q}_{solar}} \quad (4)$$

The overall energy efficiency of the cyclecan be expressed by:

$$\eta_{energy} = \frac{\dot{W}_{el}+\dot{Q}_{e}}{\dot{E}_{solar}} \quad (5)$$

where $\dot{W}_{el} = 0.95 \times \dot{W}_{T}$

$\dot{Q}_{e} = 0.8 \times \dot{Q}_{g}$ and $\dot{Q}_{g} = \dot{m}_{4}(h_{6} - h_{7})$

The exergy efficiency of the proposed cycle can be calculated by:

$$\eta_{exergy} = \frac{\dot{W}_{el}+\dot{E}_{e}}{\dot{E}_{solar}} \quad (6)$$

where $\dot{E}_{e} = \dot{Q}_{e} \times \left(\frac{T_0}{T_e} - 1\right)$

The energy and exergy equationsfor each component of the proposed cogeneration cycle is shown in Table 1.

4. Result and discussions

The EES program was used to do the thermodynamic simulation, and a parametric analysis was done to determine how DNI affected the solar-based cogeneration cycle's energy, mass flow rate of solution, COP, and energy efficiency. The Engineering Equation Solver (EES) software has been utilized to carry out thermodynamic simulation. A parametric analysis is incorporated into the simulation to look at how DNI affects several cogeneration cycle performance metrics that are fueled by solar energy. The analysis most likely looks at how variations in DNI impact the cogeneration cycle's energy, exergy efficiency, and mass flow rate of the solution COP. Understanding the system's flow and heat transfer processes depends on this parameter.

Exergy destruction within the various components of the proposed cycle, as shown in Fig. 2, is the focus of the analysis. A thermodynamic system's inefficiencies and losses are quantified by the concept of energy destruction. It is found that 31.37% and 27.04% of the irreversibility in the solar field, respectively, are caused by the central receiver (CR) and the heliostat. Furthermore, the HRSG exhibits a significant level of heat density, at around 9.07%. The energy destruction is determined to be less than 8% for each of the remaining cycle components. The cogeneration energy system shown in Fig. 3 has different energy and energy efficiency depending on the change in DNI. The DNI increase from (600 -1050 W/m2) is shown to provide a negligible improvement in energy efficiency. The surface temperature of the CR is directly impacted by the DNI, and following a significant increase in DNI, a little increase in receiver temperature is seen. It has been noted that as DNI rises, the cogeneration system's energy efficiency rises as well.

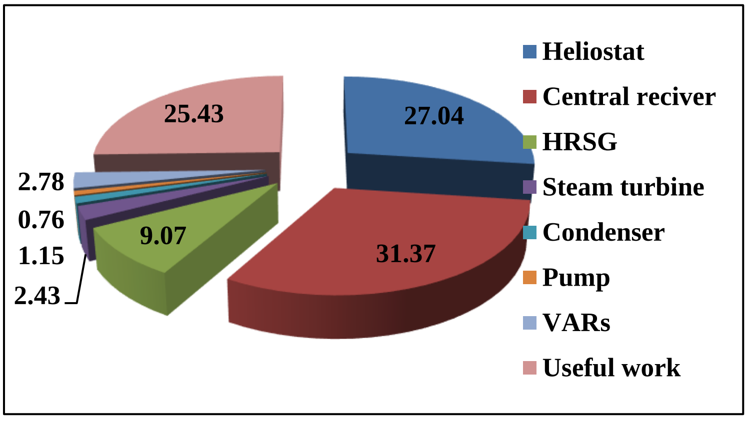

Figure 2. Proportion of energy dissipation at each phase of the cycle of combined power and cooling cogeneration

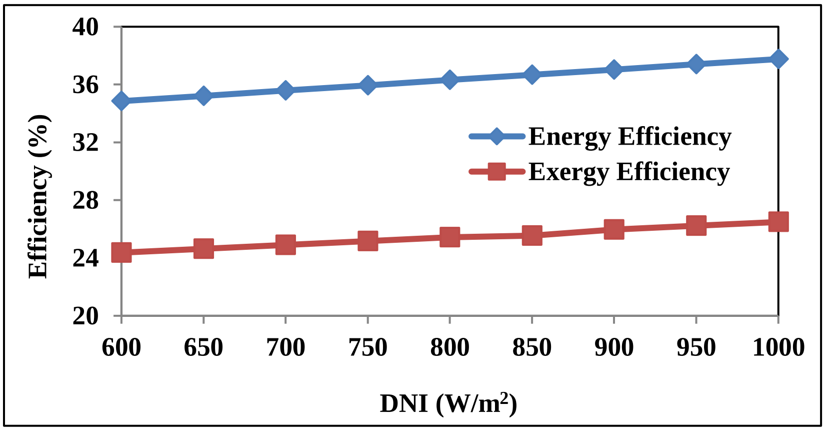

Figure 3. Energy and energy efficiency variations in response to changes in the combined power and cooling cogeneration cycle's DNI

The effect of DNI on the mass flow rates of a solution, COP, and circulation ratio of a cogeneration energy system is shown in Fig. 4 and Fig. 5. It is estimated that when the DNI is increasing of the cycle means the mass flow rates of solution decrease. Further, as the DNI increases the COP increases marginally, but CR decreases rapidly as presented in Fig. 5. The observed trend justifies that the evaporator temperature should be kept low for better cycle performance.

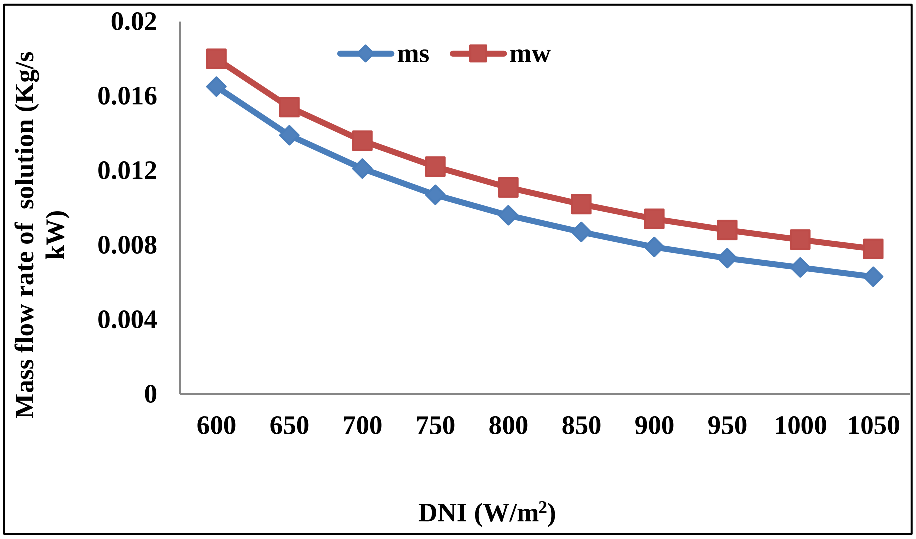

Figure 4. Effect of DNI on mass flow rates of solution

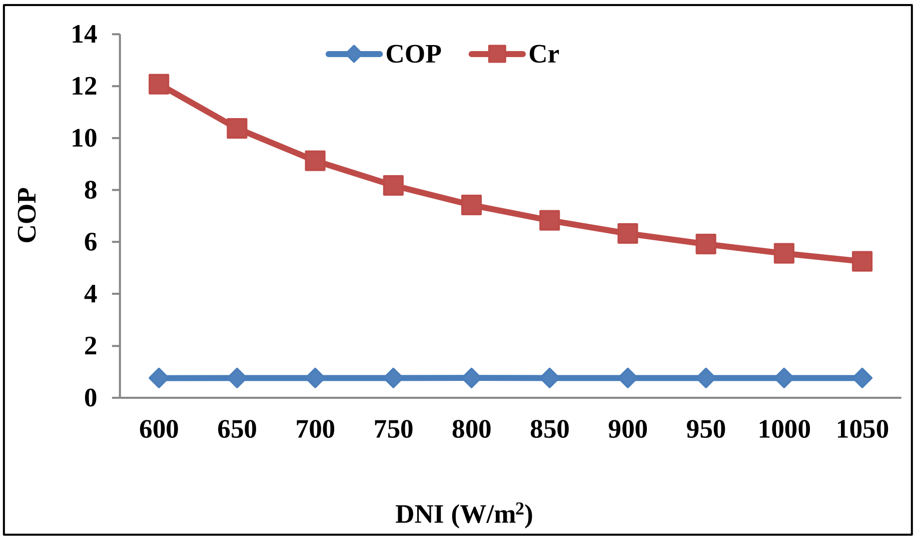

Figure 5. Effect of DNI on COP and circulation ratio

Figures 6 and 7 show how, while keeping absorber, generator, and condenser temperatures constant, changes in evaporator temperature (from 5°C to 15°C) impact mass flow rates of a solution, COP (Coefficient of Performance), and circulation ratio. The results show that the mass flow rates of the solution drop in proportion to an increase in the evaporator temperature. The justification for using a single-effect vapor absorption refrigeration cycle is strengthened by this result. Moreover, thetemperature increases will make the refrigeration cycle unviable. It justified the fact that the evaporator temperature should be kept low for better performance. Because the COP is inversely proportional to the temperature difference, it increases as the evaporator temperature rises. The results are in good agreement with [16].

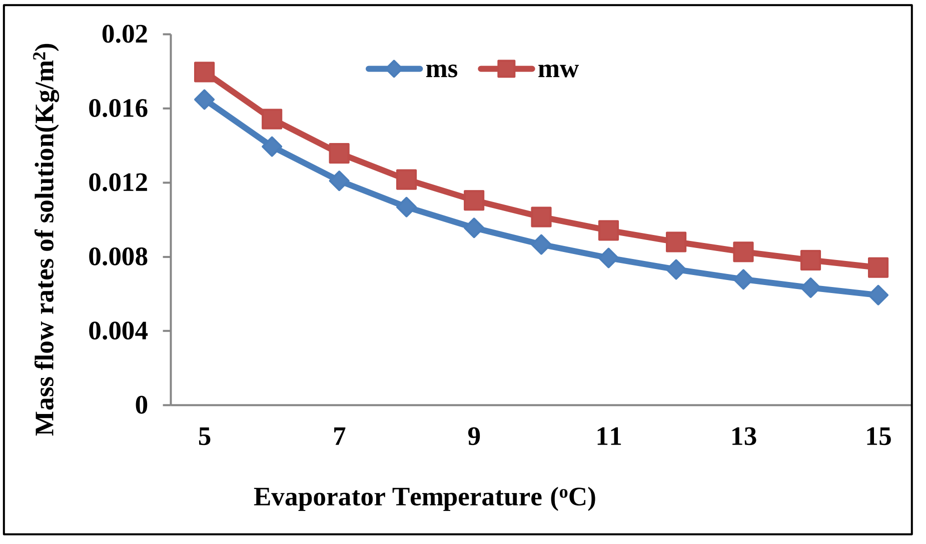

Figure 6. Variation of mass flow rates of solution with evaporator temperature

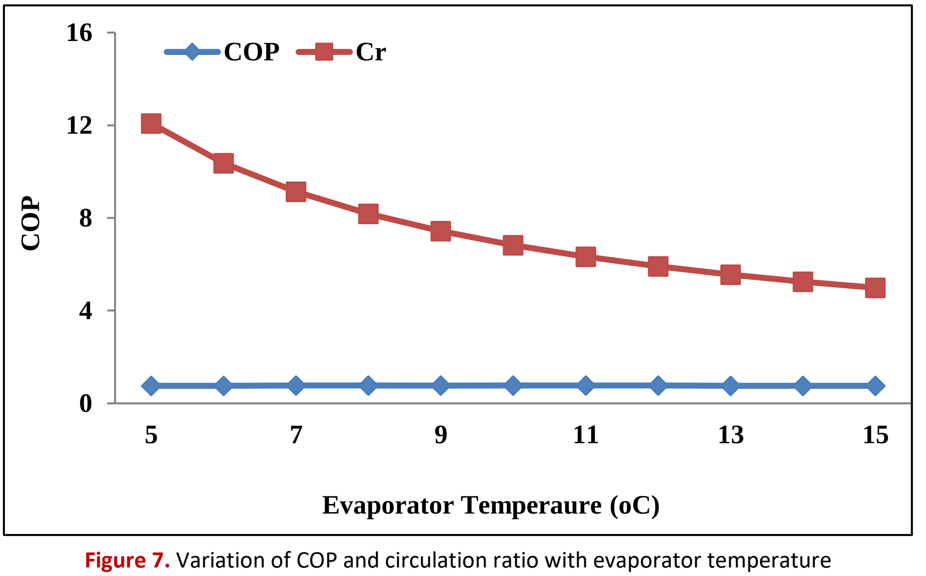

Figure 7. Variation of COP and circulation ratio with evaporator temperature

5. Conclusions

The effort made in thisstudy is to recover the large quantity of waste heat left at the exit of a condenser. This waste heat causes inefficiency in the Rankine cycle in most cases along with the electric power cooling in the range of AC. Therefore, the Rankine cycle meets the electricity demand and the absorption refrigeration cycle provides cooling in the AC ranges for space conditioning of a particular building. Theconclusions drawn from the current study are summarized as follows:

- The distribution of exergy within the cycle reveals that the central receiver is responsible for 31.37% of the total irreversibility in the system, the heliostat contributes 27.04%, and the remainder is attributed to various other components.
- With the increase in DNI (600-1050 W/m2), the energy efficiency increased from 34.85% to 37.76% while the exergy efficiency increased from 24.37% to 26.49%.
- COP of vapour absorption refrigeration cycle needs to be increased, which currently is below unity by trying different solutions/blends.m

6. Nomenclature

CO_2	Carbon dioxide
DNI	Direct normal irradiance (W/m^2)
$LiBr\text{-}H_2O$	Lithium bromide-water
SHE	Solution heat exchanger
T_0	Ambient temperature (°C)
A_{feild}	Area of heliostat field (m^2)
$\dot{E}_e$	Exergy of the evaporator (KW)
$\dot{Q}_{CR}$	Heat energy of central receiver (KW)
$\dot{Q}_e$	Energy of Evaporator (KW)
$\dot{Q}_g$	Heat energy of generator (KW)
$\dot{Q}_{lost,CR}$	Loss of heat energy from the central receiver (KW)
$\dot{Q}_{lost,heliostat}$	Loss of heat energy from heliostat (KW)
$\dot{m}$	Mass flow rate (Kg/sec)
$\dot{W}_T$	Turbine work (KW)
$\dot{W}_{el}$	Electrical power (KW)
η_{energy}	Energy efficiency (%)
η_{exergy}	Energy efficiency (%)
a–h	State points of cogeneration cycle
1–18	State points of cogeneration cycle

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Table 1. Energy and exergy balance equations of each component for the combined power and cooling cogeneration cycle

Component	Energy balance equations	Exergy balance equations
Heliostat field	$\dot{Q}_{solar} = I \times A$	$\dot{E}_{solar} = \dot{Q}_{solar}\left(1 - \frac{T_0}{T_{solar}}\right)$
Central receiver	$\dot{Q}_{(rcv)} + \dot{m}_3 h_3 = \dot{m}_1 h_1$	$\Psi_{(rec)} + \dot{m}_3 s_3 = \dot{m}_1 s_1$
HRSG	$\dot{m}_1 h_1 + \dot{m}_8 h_8 = \dot{m}_4 h_4 + \dot{m}_2 h_2$	$\dot{m}_1 s_1 + \dot{m}_8 s_8 = \dot{m}_4 s_4 + \dot{m}_2 s_2$
Steam turbine	$\dot{m}_4 h_4 = \dot{m}_5 h_5 + \dot{W}_{ST}$	$\dot{m}_4 s_4 = \dot{m}_5 s_5 + \dot{W}_{ST}$
Condenser 1	$\dot{m}_5 h_5 + \dot{m}_a h_a = \dot{m}_6 h_6 + \dot{m}_b h_b$	$\dot{m}_5 s_5 + \dot{m}_a s_a = \dot{m}_6 s_6 + \dot{m}_b s_b$
Pump1	$\dot{m}_7 h_6 + \dot{W}_{P1} = \dot{m}_8 h_8$	$\dot{m}_7 s_6 + \dot{W}_{P1} = \dot{m}_8 s_8$
Solar pump	$\dot{m}_2 h_2 + \dot{W}_{SP} = \dot{m}_3 h_3$	$\dot{m}_2 s_2 + \dot{W}_{SP} = \dot{m}_3 s_3$
Desorber	$\dot{m}_6 h_6 + \dot{m}_{15} h_{15} = \dot{m}_7 h_7 + \dot{m}_9 h_9 + \dot{m}_{16} h_{16}$	$\dot{m}_6 s_6 + \dot{m}_{15} s_{15} = \dot{m}_7 s_7 + \dot{m}_9 s_9 + \dot{m}_{16} s_{16}$
Condenser 2	$\dot{m}_9 h_9 + \dot{m}_g h_g = \dot{m}_{10} h_{10} + \dot{m}_h h_h$	$\dot{m}_9 s_9 + \dot{m}_g s_g = \dot{m}_{10} s_{10} + \dot{m}_h s_h$
Expansion valve1	$\dot{m}_{10} h_{10} = \dot{m}_{11} h_{11}$	$\dot{m}_{10} s_{10} = \dot{m}_{11} s_{11}$
Evaporator	$\dot{m}_{11} h_{11} + \dot{m}_e h_e = \dot{m}_{12} h_{12} + \dot{m}_f h_f$	$\dot{m}_{11} s_{11} + \dot{m}_e s_e = \dot{m}_{12} s_{12} + \dot{m}_f s_f$
Absorber	$\dot{m}_{12} h_{12} + \dot{m}_{18} h_{18} + \dot{m}_c h_c = \dot{m}_{13} h_{13} + \dot{m}_d h_d$	$\dot{m}_{12} s_{12} + \dot{m}_{18} s_{18} + \dot{m}_c s_c = \dot{m}_{13} s_{13} + \dot{m}_d s_d$
Pump 2	$\dot{m}_{13} h_{13} + \dot{W}_{P2} = \dot{m}_{14} h_{14}$	$\dot{m}_{13} s_{13} + \dot{W}_{P2} = \dot{m}_{14} s_{14}$
Expansion valve 2	$\dot{m}_{17} h_{17} = \dot{m}_{18} h_{18}$	$\dot{m}_{17} s_{17} = \dot{m}_{18} s_{18}$
Heat exchanger	$\dot{m}_{16} h_{16} + \dot{m}_{14} h_{14} = \dot{m}_{17} h_{17} + \dot{m}_{15} h_{15}$	$\dot{m}_{16} s_{16} + \dot{m}_{14} s_{14} = \dot{m}_{17} s_{17} + \dot{m}_{15} s_{15}$