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Comparative assessment of solar energy-based transcritical CO_2 Rankine cycles for different layouts

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Abstract: In this study, different transcritical CO_2 Rankine (tCO_2-RC) cycles with vacuum tube solar collectors are examined. In this context, the transcritical CO_2 Rankine cycle in different configurations, such as simple, regenerator, reheat, regenerator, and reheat, which is widely used in the literature, has been chosen. First, the energy and exergy efficiencies of the cycles were founded by performing thermodynamic analyzes of four various tCO_2-RCs under certain operating parameters. Moreover, parametric studies were carried out according to the factors affecting the system performance, like turbine input temperature and input pressure of the turbine. Both analyzes and parametric studies were conducted utilizing the Engineering Equation Solver (EES) computer program. As outcomes of analyzes, the highest thermal efficiency was founded for the tCO_2-RC with reheat and regenerator by 11.8%, and the lowest energy efficiency was calculated for the simple tCO_2-RC and tCO_2-RC with reheat by approximately 6.6%. While all tCO_2-RCs' energy and exergy efficiencies increased with the rise of the turbine's input pressure, the energy and exergy efficiency of all tCO_2-RCs decreased with the rise of the pump's input pressure.

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Güneş enerjisi kaynaklı farklı transkritik CO_2 Rankine çevrimlerinin karşılaştırılması

Anahtar Kelimeler
Vakum tüplü U borulu güneş kollektörü
Transkritik CO_2 Rankine çevrimi
Enerji
Ekserji

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Özet: Bu çalışmada, vakum tüplü güneş kollektörlü farklı transkritik CO_2 Rankine çevrimleri incelenmiştir. Bu kapsamda literatürde yaygın olarak kullanılan basit, rejeneratör, yeniden ısıtmalı, rejeneratör ve yeniden ısıtmalı transkritik CO_2 Rankine çevrimleri seçilmiştir. İlk olarak, belirli çalışma parametreleri altında dört farklı transkritik CO_2 Rankine çevriminin termodinamik analizleri yapılarak sistemlerin enerji ve ekserji verimleri hesaplanmıştır. Ayrıca türbin giriş sıcaklığı ve türbin giriş basıncı gibi sistem performansını etkileyen faktörlere göre parametrik çalışmalar yapılmıştır. Hem analizler hem de parametrik çalışmalar EES bilgisayar programı kullanılarak yapılmıştır. Termodinamik analizler sonucunda en yüksek enerji verimi % 11.8 ile yeniden ısıtmalı ve rejeneratörlü transkritik CO_2 Rankine çevrimi için, en düşük enerji verimi ise % 6.6 ile basit transkritik CO_2 Rankine çevrimi ve yeniden ısıtmalı transkritik CO_2 Rankine çevrimi için hesaplanmıştır. Tüm transkritik CO_2 Rankine çevrimlerinin enerji ve ekserji verimleri türbin giriş basıncının artmasıyla artarken, tüm transkritik CO_2 Rankine çevrimlerinin enerji ve ekserji verimi pompa giriş basıncının artmasıyla azalmıştır.

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1. Introduction

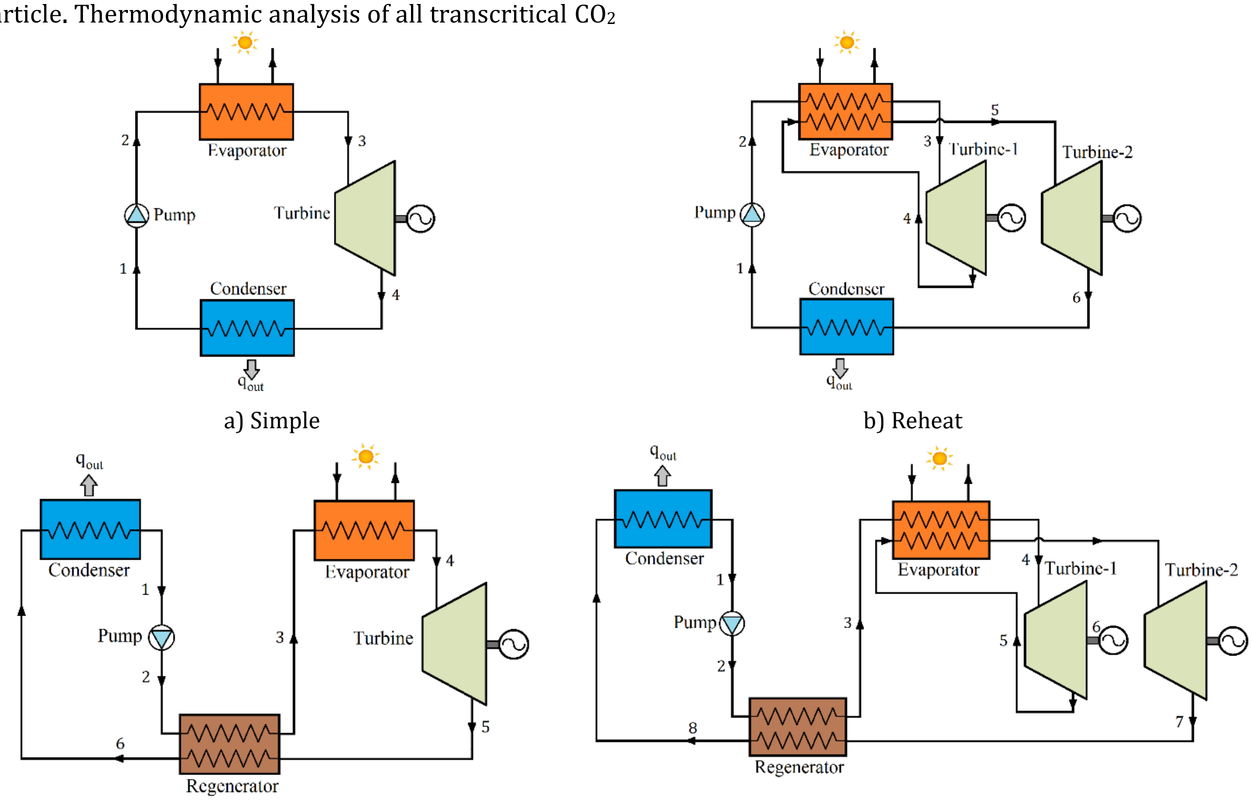
In recent years, rapid population growth, industrialization, and technological advances have raised the demand for energy consumed. Fossil fuels meet a significant portion of the growing energy demand. Critical environmental issues like greenhouse impacts, climate alteration, depletion of ozone, and global heating are brought on by fossil fuel consumption. These shortcomings of fossil fuels have recently increased interest in renewable energy resources. Solar energy has long been recognized as one of the most promising alternative energy resources. Moreover, to the utilization of alternative energy resources, the economical and effective use of low and high-temperature heat is more important [1]. Due to its simplicity, relatively low initial costs, and suitability for low/medium temperature heat resources, the organic Rankine cycle (ORC) is an effective and affordable method for generating electricity. In addition, these systems can be integrated into many systems, like solar energy, geothermal energy, and waste heat [2]. Many conventional ORCs use working fluids such as CFC, HCFC, and HFC. Unfortunately, these fluids have high global warming potential (GWP) and ozone depletion potential (ODP) [3]. Additionally, flammable organic liquids at high temperatures might cause major safety concerns [4]. In addition, in ORC, where organic fluids are used, a pinching point problem occurs between the heat source and the working fluid, and this causes an increase in irreversibility in the system. [5]. Therefore, the use of CO_2, which is both environmentally friendly and has good heat transfer properties, has increased in recent years. CO_2 is abundant, inexpensive, non-toxic, and beneficial to the environment. Additionally, CO_2 has good thermal compatibility with the heat source and enough thermal stability to tolerate its high temperatures [6]. CO_2, which is natural, inexpensive, and has a low critical temperature and pressure, has been investigated by many researchers as a supercritical agent fluid. However, the CO_2's low critical temperature characteristic has the disadvantage that in the tCO_2-RC, it is hard to condense turbine exhaust gas to liquid at subcritical pressure. The research group of Zhang et al. [7] has conducted significant research work to improve and test the solar energy-based CO_2 Rankine cycle. Al-Zahrani et al. [8] investigated the performance of a combined system based on geothermal energy for the production of electricity, hydrogen, and heat. The combined cycle consists of an ORC, an electrolyzer, and a tCO_2-RC. They calculated the tCO_2-RC's efficiency as 9.2% at a geothermal water temperature of 200 ˚C. They calculated the integrated cycle's energy efficiency as 13.67% and the exergy efficiency as 32.27%. Bamisile et al. [9] investigated the multigeneration's thermodynamic analysis made of a parabolic collector, a supercritical CO_2 Brayton cycle (sCO_2 BC), a tCO_2-RC, and a cascade cooling cycle. They calculated the tCO_2-RC's energy efficiency as 16.17% and the exergy efficiency as 5.53%, according to the results of the analysis. Cayer et al. [10] have done a detailed analysis of a tCO_2-RC using industrial low-temperature process heat as the heat resource. They found the efficiency of the tCO_2-RC with a regenerator as 8.5% and the efficiency of the tCO_2-RC without the regenerator as 7.3%, at a high pressure of 10 MPa. Pan et al. [11] established the tCO_2-RC using a rotary piston expander in vitro. They carried out experimental studies about operating parameters, electrical power, and efficiency. The experimental analysis revealed that when the high pressure is around 11 MPa and the low pressure is approximately 4.6 MPa, the constant net power production is approximately 1100W, and the system's thermal efficiency is 5.0%. Shu et al. [12] compared tCO_2-RC with a regenerator, reheat, and regenerator, and ORC with a regenerator, reheat, and regenerator for engine waste heat recovery. They used R123 as the working fluid in the ORC. They reported that by rising the turbine input temperature from 500 K to 1000 K, the thermal efficiency of the tCO_2-RC with reheat and regenerator increased 184% more than the tCO_2-RC with a regenerator. Additionally, they claimed that at maximum turbine input temperature, the tCO_2-RC with reheat and regenerator had a greater thermal efficiency than the ORC with reheat and regenerator. Yamaguchi et al. [13] examined the analysis of the solar energy-assisted tCO_2-RC. For the analysis, meteorological data from Japan, Kyoto, and typical summer and winter season days were used. While they determined that the solar energy-based tCO_2-RC had an efficiency of 3.4% for the winter and 5.78% for the summer, they found that the power generation was 0.118 kW for the winter and 0.177 kW for the summer. Using parabolic solar energy collectors, Sarmiento et al. [14] investigated the functionality of the solar energy-driven tCO_2-RC. The thermodynamic performance of the solar energy-aided tCO_2-RC with reheat was examined by Al-Zahrani and Dincer [15]. The investigated system was made of a parabolic solar collector, thermal energy storage, tCO_2-RC, and an absorption cooling system. The efficiency of the power cycle and the integrated system are both assessed in relation to variations in cycle temperature and pressure. They determined that the tCO_2-RC had an energy and exergy efficiency of 34% and 82%, respectively. The thermodynamic analysis of the solar energy-driven tCO_2-RC for electricity and heat production, as well as the optimization of the integrated system, was conducted by Kizilkan et al. [16]. They examined the performance of the integrated cycle monthly using the EES program. Following their investigation, they calculated the combined system's best energy and exergy efficiencies during the months of July and August. They found the maximum turbine output as 0.269 kW for the month of July. Liang et al.

[17] proposed a zero-emission fuel cell-based multigeneration system combined with the ORC and the tCO_2-RC. Shu et al. [4] compared four different transcritical CO_2 Rankine cycles: simple tCO_2-RC, tCO_2-RC with a regenerator, tCO_2-RC with reheat, tCO_2-RC with a regenerator, and reheat. They indicated that the performance of the tCO_2-RC with reheat and regenerator was the highest. Akbari [18] made comprehensive exergy and exergo-economic analysis of the integrated cycle made of the tCO_2-RC, the Stirling power system, and the liquefied natural gas process. Other theoretical work was done by Naseri et al. [19] concerning the renewable energy - driven tCO_2-RC for the production of hydrogen and hot water. Meng et al. [20] investigated the thermo-economic performances of geothermal energy based various tCO_2-RCs. They also compared the tCO_2-RC by the ORC and Kalina systems. They stated that the efficiency of the tCO_2-RC with reheat was higher than the other tCO_2-RCs. Moreover, they reported that while the net power generation of the tCO_2-RC with a reheat is much better than that of the Kalina system and ORC, the energy efficiency of the tCO_2-RC with a reheat is 36.9% lower than that of ORC.

In this paper, the performances of four various configurations of the tCO_2-RC with vacuum tube solar collector are compared. The lack of thermodynamic analysis and comparison of tCO_2-RCs with vacuum tube solar collectors is the reason for the development of this article. Thermodynamic analysis of all transcritical CO_2 Rankine cycles was carried out, taking into account the accepted system operating parameters. In addition, the evacuated U-tube solar collector's dynamic analysis was made. Using the Isparta meteorological data, the CO_2 outlet temperatures from the collector were calculated. Moreover, the effects of the pump input pressure, input pressure of the turbine, and the turbine's input temperature on cycle performance were investigated using parametric analyses. As a result of the analyses, the performances of the tCO_2-RCs and wherewith the design parameters affecting the cycle affect the performance were examined.

2. System Description

Schematic representations of four various tCO_2-RCs are shown in Figure 1; simple tCO_2-RC, tCO_2-RC with reheat, tCO_2-RC with a regenerator, and tCO_2-RC with reheat and regenerator. To make more use of energy, the tCO_2-RC with reheat cycle has been developed; in this cycle, the fluid exiting the first turbine then enters the evaporator again to be reheated. After reheating, it enters the second turbine (Figure 1b). Likewise, a heat exchanger has been added to the simple tCO_2-BC to improve its performance, as seen in (Figure 1c, in order to make use of the excess heat energy. The final cycle is a combination of second and third cycles, which combine reheat and regeneration processes.



a) Simple

b) Reheat

c) Regenerator

d) Regenerator and reheat

Figure 1. Different tCO_2-RC layouts

In the simple cycle, the heat needed for the tCO_2-RC is supplied from evacuated solar collectors. The 15 collector units that make up the solar collector system each have 13 U-type pipes. A U-tube system consists of two nested borosilicate glasses, an absorber surface, a fin, a U-shaped pipe, and a working fluid (Figure 2). The tCO_2-RC operates on the general operating principle described below: The agent fluid coming from the condenser output is pressurized via a pump. Then, the working fluid, CO_2, heats up and reaches the supercritical phase while directly the evacuated solar collectors. Work is produced by passing the fluid, which has high pressure and temperature, through the turbine. The condenser receives the low-pressure fluid that expands from the turbine. In the condenser, the fluid condenses before being returned to the pump. Thus, the cycle is completed.

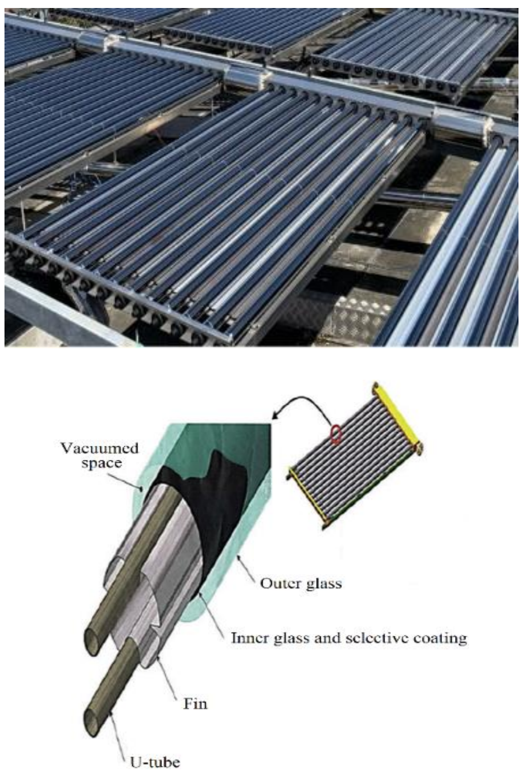


Figure 2. Evacuated U-tube solar collector's cross-section

The variation of the thermophysical characteristics of CO_2 with the temperature at 9000 kPa pressure is depicted in Figure 3. The thermophysical characteristics of CO_2 exhibit sudden changes close to the critical point, as seen in Figure 2. Due to its unique thermophysical properties, supercritical CO_2 has a greater heat transfer property than the liquid and gas phases.

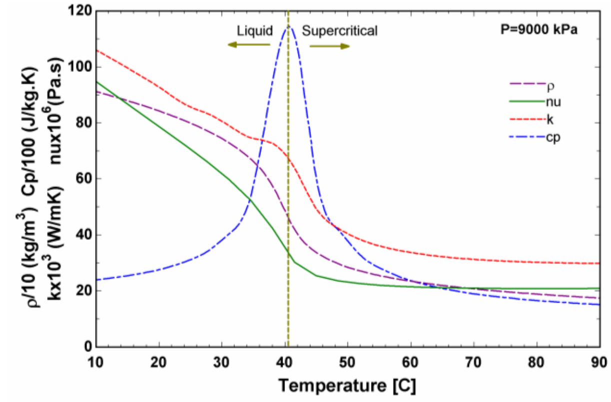


Figure 3. Thermophysical properties of supercritical CO_2 at 9 MPa

Table 1 provides the design parameters that were utilized to compare the performance of the evacuated tube solar collectors based tCO_2-BC.

Table 1. Operating parameters of transcritical CO_2 Rankine cycle

Parameter	**Value**
Input temperature of the turbine, °C	150
Outlet pressure of turbine, kPa	6500 [21]
Effectiveness of heat exchanger, %	98 [23]
Turbine's isentropic efficiency, %	90 [22]
Pump's isentropic efficiency, %	85 [15]
Pressure ratio	1.5 [21]

3. Thermodynamic Modelling

Various tCO_2-RCs' energy and exergy analyzes were performed using the EES program [24]. To simplify the computation and its complexity, some assumptions are made.

- Every component is taken into account as a steady-state system.
- It is assumed that there are no kinetic and potential energy changes in the system.
- Heat losses and pressure drops in the pipes are neglected.
- It is acknowledged that the condenser outlet has a saturated liquid.
- The reference state properties are 25°C and 101.325 kPa.

The Ref. [25] is primarily the basis for the design considerations of the solar collector system shown here. The following is a definition of the beneficial solar energy collected:

$$\dot{Q}_u = F_R\left[S\,A_{ap} - A_r U_L (T_{in} - T_a)\right] \quad (1)$$

Here F_R is the collector heat removal factor, S is the solar irradiation, A_{ap} is the collector aperture area, A_r is the receiver area, U_L is the entire heat loss coefficient of the collector between ambient and absorber surface, T_{in} is the agent fluid input temperature, and T_a is the ambient temperature.

The useable energy gathered by the evacuated U-tube solar collector can be expressed like the following to define the CO_2 temperature at the collector outlet :

$$\dot{Q}_u = \dot{m}C_p(T_{out} - T_{in}) \tag{2}$$

More details on finding useful heat from evacuated solar collectors can be found in Ref. [26].

The mass and energy balance equilibriums for systems with the continuous flow are written as follows in [27]:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{3}$$

$$\sum \dot{m}_{in}\left(h + \frac{v^2}{2} + gz\right)_{in} + \sum \dot{Q}_{in} + \sum \dot{W}_{in} = \sum \dot{m}_{out}\left(h + \frac{v^2}{2} + gz\right)_{out} + \sum \dot{Q}_{out} + \sum \dot{W}_{out} \tag{4}$$

here, $\dot{m}$ represents the fluid mass flow rate, and the circumstances at the input and output are denoted by the subscripts "in" and "out," respectively. Heat transfer rate $\dot{Q}$, power transfer rate $\dot{W}$, specific enthalpy h, speed v, altitude z, and gravitational acceleration g are all variables.

Entropy and exergy balance equality can be described as [28]:

$$\sum \dot{m}_{in}s_{in} + \sum \frac{\dot{Q}}{T} + \dot{S}_{gen} = \sum \dot{m}_{out}s_{out} \tag{5}$$

$$\sum \dot{m}_{in}\dot{ex}_{flow} + \sum \dot{Ex}^{Q}_{in} + \sum \dot{Ex}^{W}_{in} = \sum \dot{m}_{out}\dot{ex}_{flow} + \sum \dot{Ex}^{Q}_{out} + \sum \dot{Ex}^{W}_{out} + \dot{Ex}_{dest} \tag{6}$$

Here, $\dot{ex}_{flow}$ denotes current exergy, $\dot{Ex}^{Q}$ indicates heat exergy, $\dot{Ex}^{W}$ shows work exergy and $\dot{Ex}_{dest}$ represents exergy irreversibility. Each term given in the previous equation is described as follows:

$$\dot{ex}_{flow} = (h - h_0) - T_0(s - s_0) \tag{7}$$

$$\dot{Ex}^{W} = \dot{W} \tag{8}$$

$$\dot{Ex}^{Q} = \dot{Q}\left(\frac{T - T_0}{T}\right) \tag{9}$$

$$\dot{Ex}_{dest} = T_0\dot{S}_{gen} \tag{10}$$

The energy and exergy efficiency equilibrium for all the tCO_2-RC can be written as follows:

$$\eta_{energy} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{11}$$

$$\eta_{exergy} = \frac{\dot{Ex}_{\dot{W}_{net}}}{\dot{Ex}_{\dot{Q}_{in}}} \tag{12}$$

4. Validation

Al-Zahrani et al. [8]'s work in the literature was used to confirm the thermodynamic analysis of the studied tCO_2-RC with reheat and regenerator. The change in energy efficiency with regard to the turbine inlet pressure was investigated parametrically by accepting the design parameters of the reference research. A comparison of the reference work [8] and current study is given in Figure 4. The figure shows that the tCO_2-RC's calculated energy efficiency for the current investigation and the reference study are equivalent to one another.

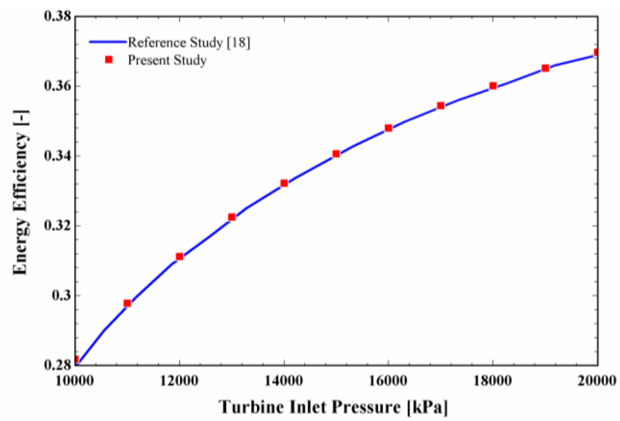


Figure 4. Validation of tCO_2-RC with regenerator and reheat

5. Results and Discussions

In this research, the performance of four various tCO_2-RCs integrated into an evacuated U-tube solar collector was investigated. In both the evacuated solar collector and the transcritical Rankine cycle, natural eco-friendly CO_2 was used. Finally, parametric studies were conducted for the factors affecting the system performance. Analyzes and parametric studies were made using the EES computer program. In Table 2, the values of pressure, temperature, mass flow rate, enthalpy, and entropy for the state points of the tCO_2 RC with reheat and regenerator are given.

Figure 5 displays the exit temperature of CO_2 from the collector for a typical summer day. Using the meteorological data of July 15, a typical summer day in Isparta city, Turkey, the CO_2 temperature from the evacuated U-tube solar collector was calculated. The figure shows that at noon, the CO_2 temperature soared to a maximum of 194°C.

Table 2. Thermodynamic properties of each point in the tCO_2-RC with regenerator and reheat

State	P [kPa]	T [°C]	h [kJ/kg]	s [kJ/kgK]	ṁ [kg/s]	e [kJ/kg]
1	6500	25.44	-230	-1.484	1.00	212.8
2	10000	33.14	-224.4	-1.482	1.00	217.6
3	10000	58.21	-87.4	-1.051	1.00	226.4
4	10000	150	65.17	-0.637	1.00	255.6
5	8062	131.1	52.18	-0.634	1.00	241.6
6	8062	150	74.86	-0.579	1.00	247.9
7	6500	131.5	61.46	-0.576	1.00	233.4
8	6500	35.11	-75.49	-0.968	1.00	213.5

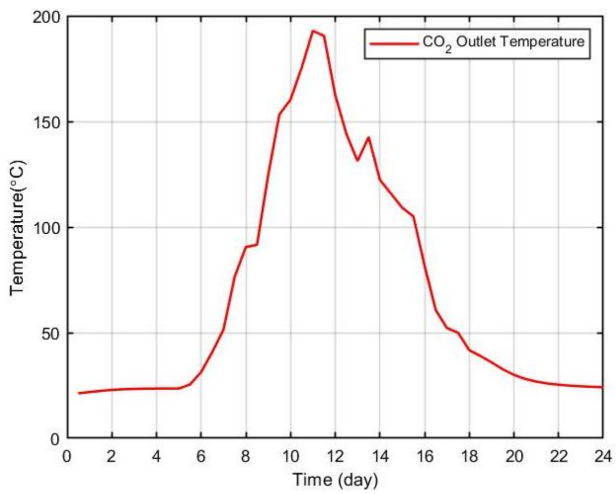


Figure 5. Variation of exit temperature of CO_2 from the collector

The system operating characteristics were taken into consideration when calculating the energy and exergy efficiency of various tCO_2-RCs. Figure 6 provided the energy and exergy efficiencies of all tCO_2-RCs. The highest efficiency was calculated for the transcritical Rankine cycle with regenerator and reheat under certain operating parameters. It was followed by the Rankine cycle with a regenerator, the simple tCO_2-RC, and the tCO_2-RCs with reheat.

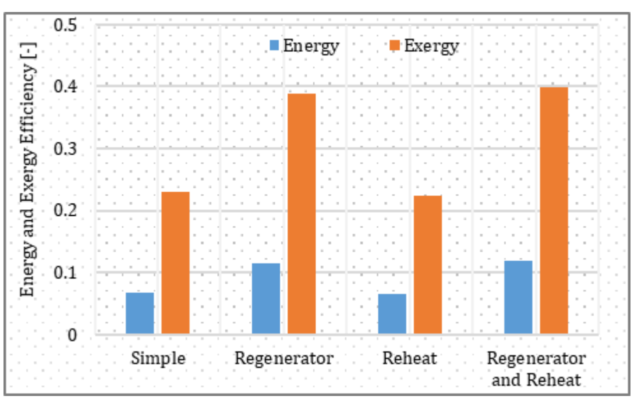


Figure 6. Energy and exergy efficiencies of various tCO_2-RCs

Figure 7 shows how the cycles' energy efficiency varies in regard to the temperature of the turbine inlet. The thermodynamic analysis revealed that the tCO_2-RC with regenerator and reheat had the best efficiency while the tCO_2-RC with reheat had the lowest efficiency. With the rise of turbine input temperature, the efficiency of the tCO_2-RC with regenerator, regenerator, and reheat raised at a higher rate, while the increment in the tCO_2-RC with reheat and simple was very small.

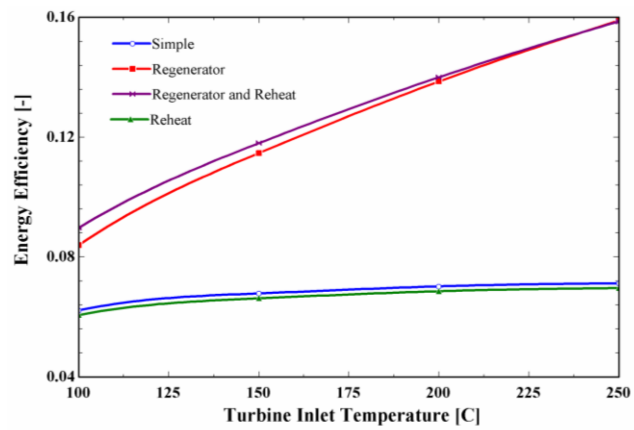


Figure 7. Effect of turbine inlet temperature on energy efficiency

Figure 8 depicts how the cycles' energy efficiency varies as a function of the turbine input pressure. The net power generated by the system will rise as the turbine input pressure rises, increasing the efficiency of all tCO_2-RCs. The highest efficiency is in the tCO_2-RC with regenerator and reheat, and the lowest efficiency is in the tCO_2-RC with reheat. All tCO_2-RCs have fairly similar energy efficiencies when the turbine input pressure is less than about 9500 kPa. However, the distinction among the efficiency of the tCO_2-RCs started to rise as the turbine inlet pressure increased above 9500 kPa.

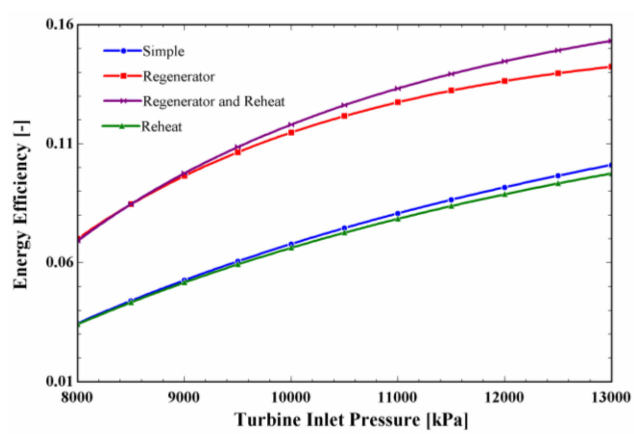


Figure 8. Effect of turbine inlet pressure on energy efficiency

The input pressure of the pump is another significant factor that The all tCO_2-RCs' energy efficiency decreased with the rise of pump input pressure. Figure

9 displays the impact of pump input pressure on energy efficiency. The quantity of work generated by the system will decrease as the pressure range in the turbine narrows with an increase in pump inlet pressure. This state will result in a decline in the energy efficiency of the tCO_2-RC.

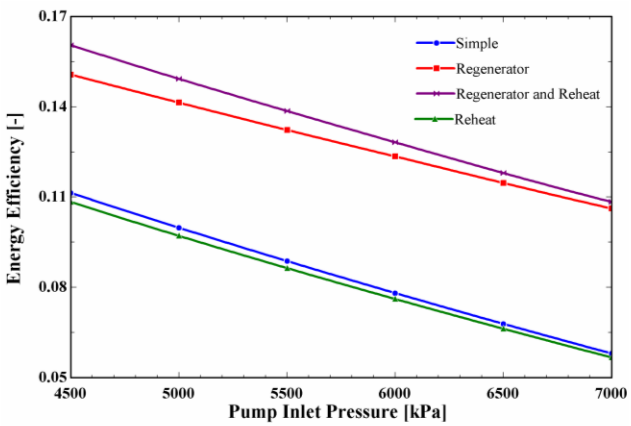


Figure 9. Effect of pump inlet pressure on energy efficiency

In Figure 10, the exergy changes of the cycles according to the turbine input temperature were given. As the turbine's input temperature raised from 100˚C to 250˚C, the network produced in all cycles, and the amount of heat entering the system raised. However, since the rising in the quantity of input heat is more than the network, the exergy efficiency of all cycles decreased according to the input temperature of the turbine.

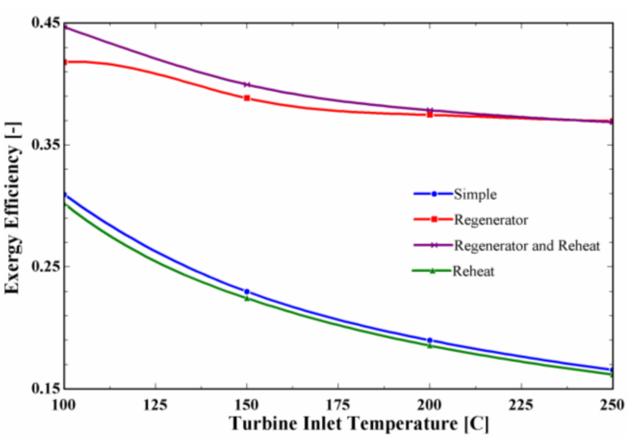


Figure 10. Effect of turbine inlet temperature on exergy efficiency

The effect of turbine inlet temperature on energy efficiency is depicted in Figure 11. As the turbine input temperature raised, both the energy efficiency and energy efficiency improved. As the turbine input pressure increased from 8000 kPa to 13000 kPa, the energy efficiency of the tCO_2-RC with regenerator and reheat, which has the maximum energy efficiency, improved from 23% to 51%. The highest increment in exergy efficiency with the rise of the high-pressure value was in the tCO_2-RC with reheat.

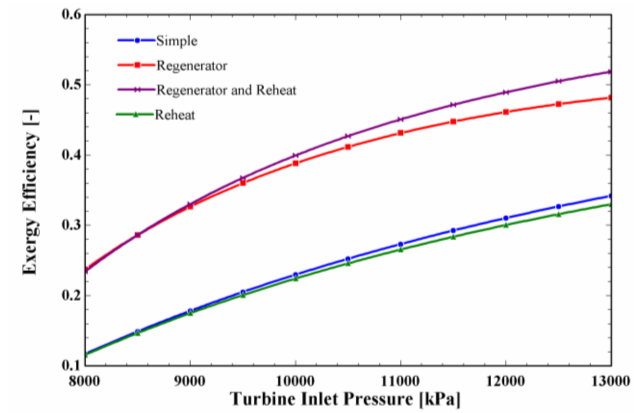


Figure 11. Effect of turbine inlet pressure on exergy efficiency

In Figure 12, the alteration in exergy efficiency with regard to the pump outlet pressure was given. Like the energy efficiency, the exergy efficiency of all tCO_2-RCs decreased as the network produced from the cycle would reduce with the rise of the pump input pressure. When the pump input pressure increased from 4500 kPa to 7000 kPa, the highest diminish of 48% occurred in the simple tCO_2-RC, and the lowest decrease of 31% take placed occurred in the tCO_2-RC with the regenerator.

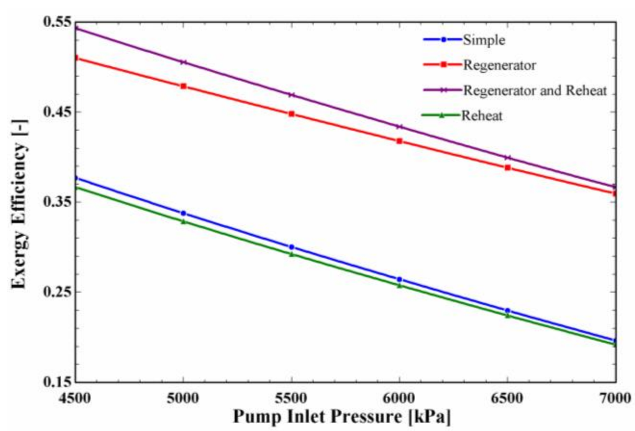


Figure 12. Effect of pump inlet pressure on exergy efficiency

6. Conclusions

Dynamic analysis of evacuated U-tube solar collector was made using Isparta meteorological data in this study. The CO_2 heated in the solar collector was directly used in four different tCO_2-RCs. The thermodynamic analysis of four various tCO_2-RCs with solar collectors was examined and the performances of the cycles were compared. The performance of the cycles was evaluated by considering the parameters affecting the operation of the system. Based on the analysis conducted, the following significant conclusions are drawn:

- Under the accepted design parameters, the highest energy efficiency was calculated in the tCO_2-RC with regenerator and reheat by 11.8 %, and the lowest energy efficiency was found in the tCO_2-RC with reheat by 6.6%.
- As the input temperature of turbine increased, the thermal efficiency of all tCO_2-RCs raised, while the exergy efficiency tended to decrease.
- All tCO_2-RCs' energy and exergy efficiencies raised when the turbine's input pressure increased from 8000 kPa to 13000 kPa. The tCO_2-RC with regenerator and reheat has the highest energy and exergy efficiency, and the tCO_2-RC with reheat has the lowest energy and exergy efficiency.
- The pump input pressure is one of the variables affecting the system's performance. As the pump's input pressure increased, all of the tCO_2-RCs' energy and exergy efficiency decreased. The highest diminish in energy efficiency is in the tCO_2-RC with reheat by 50%, and the greatest decrease in exergy efficiency is in the simple tCO_2-RC by 48%.
- High efficiency has been achieved by using the CO_2 heated in the evacuated U-tube solar collector directly in the tCO_2-RCs.
- It is thought that tCO_2-RCs should be used easily in solar energy applications, particularly in low and moderate-temperature practices.

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F_R	collector heat removal factor
S	solar radiation
A_{ap}	aperture area
A_r	receiver area
U_L	overall heat loss coefficient
T	temperature
$\dot{m}$	mass flow rate
h	enthalpy
v	speed
g	gravitational acceleration
z	altitude
$\dot{Q}$	heat transfer rate
$\dot{W}$	power transfer rate
s	entropy
$\dot{ex}_{flow}$	current exergy
$\dot{S}_{gen}$	entropy generation
$\dot{Ex}_{dest}$	exergy destruction
η	efficiency