

Integrated Photothermal-Thermodynamic Analysis of Direct-Heated S-CO₂ Tower Solar Thermal Power Systems: Postprint

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Abstract

This study integrates simple recuperated, pre-compression, recompression, partial cooling, and intercooled supercritical carbon dioxide (S-CO₂) Brayton cycles with solar power tower (SPT) systems, respectively, establishing an integrated optical-thermal-power model for direct S-CO₂ solar power tower systems. The thermodynamic performance of the entire SPT system under five S-CO₂ cycles is comparatively analyzed at different turbine inlet temperatures. The results demonstrate that as turbine inlet temperature increases, the efficiency of the entire SPT system reaches a maximum near 650 °C, indicating that higher operating temperatures are not necessarily superior for direct S-CO₂ solar power tower systems. Within the turbine inlet temperature range investigated in this study (500–800 °C), SPT systems based on intercooled and partial cooling S-CO₂ cycles achieve the highest efficiency, albeit with the most complex system configuration. The SPT system employing a recompression S-CO₂ cycle exhibits relatively high efficiency in the high-temperature range (650–800 °C) with a relatively simple system configuration, offering significant application potential.

Full Text

Preamble

Thermodynamic Analysis of Solar Thermal Power Tower Systems Integrated with the Direct-heated Supercritical CO₂ Brayton Cycles

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Abstract: In this paper, a complete mathematical model is developed for the solar power tower (SPT) system integrated with five different direct-heated supercritical CO₂ (S-CO₂) Brayton cycles (simple, pre-compression, recompression, partial-cooling, and intercooling) respectively, and the effect of turbine inlet temperature (TIT) on the thermodynamic performances of the integrated SPT systems is investigated and compared among these cycles. The results reveal that the overall efficiencies do not increase with TIT monotonically but show a parabolic tendency with maximum values around the TIT of 650 °C, which shows that it is not necessary to pursue excessively high TIT for the direct-heated integrated SPT systems. Furthermore, the intercooling and the partial-cooling S-CO₂ cycles achieve the highest overall efficiencies at the TIT of 500~800 °C, whereas the corresponding cycle configurations are more complicated. The recompression S-CO₂ cycle with relatively simple cycle configuration has higher overall efficiencies when the TIT is at 650~800 °C, making it become an attractive candidate for SPT system applications.

Keywords: direct system; S-CO₂ Brayton cycles; solar power tower; complete mathematical model; thermodynamic performances

As an effective approach for large-scale utilization of solar energy resources, solar power tower (SPT) systems have attracted increasing attention and development. However, compared with conventional fossil fuel power generation systems, their electricity cost remains relatively high. In the entire SPT system, the power cycle subsystem represents the final stage for converting solar energy into high-grade electrical energy. Adopting efficient and compact power cycle configurations is an effective way to improve overall SPT system efficiency and reduce power generation costs. The S-CO₂ Brayton cycle offers advantages over traditional steam Rankine cycles, including higher cycle efficiency, smaller equipment size, more compact system layout, and easier implementation of dry cooling [1], making the development of S-CO₂ Brayton cycles in solar thermal power systems a new research focus.

The application of S-CO₂ Brayton cycles in SPT systems mainly includes two types: indirect and direct. In indirect systems, S-CO₂ serves only as the working fluid in the power cycle subsystem, while other fluids such as molten salt act as heat transfer fluids in the receiver, with the receiver and power cycle subsystem connected through an intermediate heat exchanger. In direct systems, S-CO₂ functions as both the working fluid in the power cycle and the heat transfer fluid in the receiver. Since the allowable operating temperature of molten salt and other fluids is limited, restricting further improvement of SPT system efficiency, and S-CO₂ exhibits excellent stability at high temperatures, direct-heated S-CO₂ systems can achieve higher operating temperatures and further enhance system efficiency. Domestic and international scholars have begun researching direct-heated S-CO₂ tower solar thermal power systems. For example, Turchi et al. [2] applied several S-CO₂ Brayton cycles (with reheat) including recompression, partial-cooling, and intercooling to concentrating solar power systems, investigating their performance under dry-cooling conditions, with results show-

ing that partial-cooling and intercooling S-CO₂ cycles can still achieve thermal efficiencies above 50% under dry-cooling conditions. Chacartegui et al. [3] combined S-CO₂ Brayton cycles with organic Rankine cycles, studying the thermodynamic performance of this combined cycle when applied to tower solar thermal power systems, and found the combined cycle feasible but with limited efficiency improvement. Padilla et al. [4] integrated four S-CO₂ Brayton cycles with tower solar thermal power systems and conducted exergy analysis for each cycle with and without reheat, though in their study the receiver was treated merely as a general heat source providing heat to the S-CO₂ cycle without considering its specific characteristics.

Evidently, research on direct-heated S-CO₂ tower solar thermal power systems is limited and has focused only on analyzing the performance of the S-CO₂ Brayton power cycle itself, without investigating the thermodynamic performance of the entire direct-heated S-CO₂ solar thermal power system including the receiver. Therefore, this paper establishes a complete solar-thermal-work integrated model for direct-heated S-CO₂ tower solar thermal power systems (including the heliostat field, receiver, thermal storage subsystem, and power cycle subsystem). Based on this model, a comparative analysis of the thermodynamic performance of SPT systems under five S-CO₂ cycle configurations—simple recuperation, pre-compression, recompression, partial-cooling, and intercooling—is conducted.

1 System Description

The direct-heated S-CO₂ tower solar thermal power system studied in this paper mainly consists of four components: the heliostat field, S-CO₂ receiver, molten salt thermal storage subsystem, and S-CO₂ power cycle subsystem. The heliostat field adopts the layout form of the DAHAN power station, with specific parameters available in references [5, 6]. A cavity-type receiver is selected for the S-CO₂ receiver, which features relatively thick tube walls due to the high-temperature, high-pressure S-CO₂ serving as the heat transfer fluid. A LiCl-KCl (45%-55%) mixed salt [7] is chosen as the thermal storage medium for the thermal storage subsystem, with an allowable operating temperature range of 355~1100 °C. The thermal storage subsystem mainly comprises a high-temperature salt tank, low-temperature salt tank, and S-CO₂/molten salt heat exchanger. Five forms of S-CO₂ Brayton cycles [8]—simple recuperation, pre-compression, recompression, partial-cooling, and intercooling—are selected as the power cycle subsystem. The corresponding direct-heated S-CO₂ tower solar thermal power systems are shown in Figures 1 [Figure 1: see original paper] through 5.

The power cycle subsystem in the SPT system shown in Figure 1 is the simple recuperation S-CO₂ Brayton cycle, whose main components include the turbine, compressor, cooler, and recuperator. After expanding and doing work in the turbine, the high-temperature, high-pressure S-CO₂ preheats the S-CO₂ entering the receiver through the recuperator, then enters the compressor for pressure

increase after being cooled in the cooler. The high-pressure, low-temperature S-CO₂ enters the receiver after preheating in the recuperator and is heated into high-temperature, high-pressure S-CO₂, thus completing the entire cycle. Adding a recuperator can recover part of the heat and improve cycle efficiency.

The power cycle subsystem in the SPT system shown in Figure 2 [Figure 2: see original paper] is the pre-compression S-CO₂ Brayton cycle, which adds a low-temperature recuperator and pre-compressor to the simple recuperation S-CO₂ Brayton cycle. The process of the pre-compression S-CO₂ cycle is similar to the simple recuperation cycle, with the difference being that S-CO₂ is pre-compressed between the two recuperators to avoid the pinch-point problem [9] that occurs in the recuperator of the simple recuperation cycle, thereby improving recuperator efficiency. Meanwhile, the addition of the pre-compressor also makes the pressure at the compressor inlet independent of the turbine outlet pressure, ensuring that the main compressor can operate near the critical point of CO₂ and reducing compressor work.

The power cycle subsystem in the SPT system shown in Figure 3 [Figure 3: see original paper] is the recompression S-CO₂ Brayton cycle, which has exactly the same components as the pre-compression S-CO₂ cycle but with a different system layout. In the recompression cycle, the S-CO₂ at the low-temperature recuperator outlet is split into two streams: one passes through the cooler, main compressor, and low-temperature recuperator, while the other is directly pressurized to the cycle's maximum pressure through the recompressor, with the two streams converging before the high-temperature recuperator inlet. The split flow makes the heat capacity rates of S-CO₂ on both sides of the low-temperature recuperator more similar, thus preventing the pinch-point problem in the low-temperature recuperator and improving recuperator efficiency. Since the mass flow rate of S-CO₂ diverted to the main compressor is reduced compared to before splitting, it can be completely cooled, reducing the work consumption of the main compressor. Therefore, the recompression S-CO₂ cycle with relatively simple system layout can achieve relatively high efficiency.

The power cycle subsystem in the SPT system shown in Figure 4 [Figure 4: see original paper] is the partial-cooling S-CO₂ Brayton cycle, which is a combination of recompression and pre-compression S-CO₂ cycles. The partial-cooling S-CO₂ cycle adopts both the split-flow method to solve the pinch-point problem that easily occurs in recuperators and utilizes the pre-compressor to make the pressures at the compressor inlet and turbine outlet independent of each other. The difference from the recompression cycle is that S-CO₂ is split after being cooled and pre-compressed, with the rest of the cycle process being exactly the same as the recompression cycle. The partial-cooling S-CO₂ cycle has a larger heat absorption temperature difference, which is beneficial for the design of the S-CO₂ receiver and also facilitates integration with sensible heat storage systems [10], making it suitable for direct-heated S-CO₂ concentrating solar power systems.

The power cycle subsystem in the SPT system shown in Figure 5 [Figure 5:

see original paper] is the intercooling S-CO₂ Brayton cycle, an improved form based on the recompression S-CO₂ Brayton cycle. In the intercooling S-CO₂ cycle, the main compressor is divided into two stages, with S-CO₂ being cooled by an intercooler between the two stages. Intercooling can further reduce compressor work consumption and improve cycle efficiency, though the cycle layout is relatively more complex.

In the direct-heated S-CO₂ tower solar thermal power system, sunlight is reflected by the heliostat field and focused onto the receiver surface to heat S-CO₂ to the turbine inlet temperature. When solar energy is sufficient, part of the heated S-CO₂ directly enters the power cycle subsystem to generate electricity, while the other part transfers heat to the molten salt thermal storage subsystem through the S-CO₂/molten salt heat exchanger. If solar energy is insufficient, the molten salt thermal storage subsystem provides heat to the S-CO₂ power cycle subsystem through the heat exchanger to maintain rated power output. Adding a molten salt thermal storage subsystem can reduce the impact of solar fluctuations on the power cycle subsystem and ensure safe and stable system operation.

2 System Modeling

To conduct a complete thermodynamic performance analysis of the direct-heated S-CO₂ tower solar thermal power system, this section establishes a solar-thermal-work integrated model for the entire SPT system (including the heliostat field, S-CO₂ receiver, molten salt thermal storage subsystem, and power cycle subsystem). The following assumptions are made during modeling: (1) solar radiation at a given moment is considered constant, and all processes in the system reach steady state; (2) pressure and heat losses in pipelines are neglected; (3) kinetic and potential energies of fluids in each component are neglected; (4) heat exchange between the thermal storage subsystem and power cycle subsystem components and the environment is neglected.

2.1 Heliostat Field

The optical simulation of the heliostat field adopts the HFLCAL model [11, 12], assuming that the energy flux focused on the receiver surface by each mirror at any moment follows a two-dimensional Gaussian distribution. The total energy transferred from the heliostat field to the receiver surface at a given moment, $F(t)$, is:

$$F(t) = \frac{I(t) A}{\pi \sigma^2} \int_0^{\sigma} \int_0^{\sigma} \exp\left(-\frac{x^2 + y^2}{\sigma^2}\right) dx dy$$

where $I(t)$ is the direct solar radiation intensity; A is the area of a single heliostat; refl is the effective mirror reflectivity; $\text{atmo}(x,y)$ is the atmospheric attenuation coefficient; $\text{cos}(x,y,t)$ is the cosine efficiency; HF is the standard deviation of the normal distribution, which is affected by solar shape, heliostat tracking error, surface shape error, and astigmatism, with specific calculation methods available in reference [13].

2.2 S-CO₂ Receiver

The design dimensions and parameters of the S-CO₂ cavity receiver are shown in Table 1. Table 1 Geometric parameters of the S-CO₂ receiver.

Most of the energy projected onto the receiver surface by the heliostat field is absorbed by the S-CO₂ inside the receiver, while the other part is lost to the environment through convection, radiation, reflection, and conduction. The energy balance equation of the receiver is:

$$Q_{rec,totloss}$$

where Q_{rec} is the solar radiation energy reaching the receiver; $Q_{rec,abs}$ is the heat absorbed by S-CO₂; $Q_{rec,totloss}$ is the total heat loss of the receiver; $\dot{m}_{c,rec}$ is the mass flow rate of S-CO₂ in the receiver; $h_{ms,out}$ is the enthalpy of S-CO₂ at the receiver outlet; $h_{ms,in}$ is the enthalpy of S-CO₂ at the receiver inlet; Q_{conv} is the convective heat loss; Q_{em} is the radiative heat loss; Q_{ref} is the reflective heat loss; Q_{cond} is the conductive heat loss; calculation methods for each heat loss term can be found in reference [15].

The final thermal equilibrium state of the receiver is calculated through an iterative method. The thermal efficiency $\eta_{t,rec}$ and exergy efficiency $\eta_{e,rec}$ of the receiver are determined by equation (3):

$$\eta_{t,rec} = \frac{Q_{rec,abs}}{Q_{rec,totloss}} \quad \eta_{e,rec} = \frac{E_{rec,abs}}{E_{rec,totloss}}$$

where E_{rec} is the total exergy reaching the receiver; $E_{rec,abs}$ is the total exergy absorbed by S-CO₂.

2.3 Thermal Storage Subsystem

The thermal storage subsystem has two processes: charging and discharging. Neglecting heat exchange between the S-CO₂/molten salt heat exchanger and storage tanks with the outside world, the thermal efficiency of both charging and discharging processes is 100%. The heat exchange temperature difference inevitably causes exergy loss, and the exergy efficiency of the charging and discharging processes of the thermal storage system is calculated below.

When solar energy is sufficient, the mass flow rate of S-CO₂ in the receiver is greater than that required by the power cycle subsystem. The excess S-CO₂ transfers heat to the molten salt thermal storage subsystem through the S-CO₂/molten salt heat exchanger for storage, and the thermal storage subsystem undergoes a charging process. The inlet temperature of S-CO₂ in the S-CO₂/molten salt heat exchanger equals the turbine inlet temperature, and the outlet temperature of S-CO₂ is determined by the S-CO₂ power cycle subsystem. The temperature difference between S-CO₂ and molten salt at the high-temperature end of the heat exchanger is set to 20 °C. Based on the heat exchanger effectiveness (taken as 0.94), the thermal storage temperature of the low-temperature molten salt tank can be determined. The exergy efficiency $\eta_{e,tes}$ of the thermal storage subsystem charging process is:

out,c

where $m_{c,ex}$ is the mass flow rate of S-CO₂; $m_{s,ex}$ is the mass flow rate of molten salt; $h_{in,c}$ and $h_{out,c}$ are the specific enthalpies of S-CO₂ at the inlet and outlet, respectively; $h_{lt,s}$ and $h_{ht,s}$ are the specific enthalpies of molten salt at the inlet and outlet, respectively; $T_{ht,s}$ and $T_{lt,s}$ are the temperatures of high- and low-temperature molten salt, respectively; T_0 is the ambient temperature; $s_{in,c}$ and $s_{out,c}$ are the specific entropies of S-CO₂ at the inlet and outlet, respectively; c_p is the specific heat at constant pressure of molten salt.

The thermal storage subsystem uses the same heat exchanger to transfer heat to the S-CO₂ working fluid during discharging, with the flow directions of S-CO₂ and molten salt in the heat exchanger being opposite to those during charging. The exergy efficiency $\eta_{e,ter}$ of the thermal storage system discharging process is:

$$\eta_{e,ter} = \frac{c_{p,s} m_{s,ex,rel} (T_{out,c,rel} - T_{in,c,rel})}{c_{p,c} m_{c,ex,rel} (T_{out,c,rel} - T_{in,c,rel})}$$

where $m_{c,ex,rel}$ is the mass flow rate of S-CO₂ in the heat exchanger during discharging; $m_{s,ex,rel}$ is the mass flow rate of molten salt in the heat exchanger during discharging; $h_{out,c,rel}$ is the specific enthalpy of S-CO₂ at the outlet; $h_{in,c,rel}$ is the specific enthalpy of S-CO₂ at the inlet; $s_{out,c,rel}$ is the specific entropy of S-CO₂ at the outlet; $s_{in,c,rel}$ is the specific entropy of S-CO₂ at the inlet.

2.4 Power Cycle Subsystem

The S-CO₂ power cycle subsystem is the heat-to-electricity conversion stage of the direct-heated S-CO₂ tower solar thermal power system. Among the five S-CO₂ Brayton cycles, the intercooling S-CO₂ cycle has the most complex system structure, while the other four cycles can be obtained by simplifying it. Due to space limitations, this section only describes the thermodynamic model of the intercooling S-CO₂ cycle. The parameters of the S-CO₂ power cycle subsystem are shown in Table 2.

Figure 6 [Figure 6: see original paper] shows the T-s diagram of the intercooling S-CO₂ Brayton cycle at a turbine inlet temperature of 700 °C. State point 1 is the turbine inlet, corresponding to the cycle's highest temperature and pressure. State point 5 is the inlet of compressor C1, corresponding to the cycle's lowest temperature and pressure. State point 7 is the inlet of compressor C2, corresponding to the cycle's lowest temperature and intermediate pressure. The cycle's intermediate pressure is determined by the parameter RPR (ratio of pressure ratios), and the parameter SR (split ratio) defines the mass flow rate of S-CO₂ flowing to the main compressor. Process 1-2 is the expansion work process of S-CO₂ in the turbine. Processes 4-10, 5-6, and 7-8 are S-CO₂ compression processes in compressors. Processes 4-5 and 6-7 are S-CO₂ cooling processes. Processes 2-3 and 11-12 are heat exchange processes in the high-temperature recuperator. Processes 3-4 and 8-9 are heat exchange processes in

the low-temperature recuperator. Process 12-1 is the heat absorption process of S-CO₂ in the receiver.

where m is the mass flow rate of S-CO₂ in the cycle; T_0 is the ambient temperature; h_1 is the enthalpy of S-CO₂ at point 1; h_{12} is the enthalpy of S-CO₂ at point 12; s_1 is the entropy of S-CO₂ at point 1; s_{12} is the entropy of S-CO₂ at point 12.

Table 3 Energy balance equations of each component in the intercooling S-CO₂ Brayton cycle

The thermal efficiency η_t and exergy efficiency η_e of the intercooling S-CO₂ power cycle are calculated as follows [17]:

where the total exergy E_{total} absorbed by the S-CO₂ cycle is:

total total

Based on the above energy relationships and known cycle parameters, the parameters of each state point in the S-CO₂ cycle are obtained using an iterative method to determine the cycle's thermal efficiency and exergy efficiency. To compare the thermodynamic performance of the five S-CO₂ Brayton cycles, the RPR of the pre-compression cycle, SR of the recompression cycle, SR and RPR of the partial-cooling cycle, and SR and RPR of the intercooling cycle are all optimized for maximum cycle thermal efficiency at each turbine inlet temperature [18]. The solution results of the S-CO₂ cycle model are compared with data from reference [2], as shown in Table 4. The good agreement verifies the correctness of the S-CO₂ power cycle subsystem model.

Table 4 The optimal operating conditions and efficiency of S-CO₂ Brayton cycles at TIT of 700 °C

3 Results and Discussion

This paper selects noon on the summer solstice in Changji City, Xinjiang Province (88.6°E, 44.1°N) as the system operating condition to calculate and analyze the thermodynamic performance of the entire SPT system (including the heliostat field, receiver, thermal storage subsystem, and power cycle subsystem). The heat stored in the thermal storage subsystem is converted into useful work of the corresponding S-CO₂ power cycle subsystem through the discharging process to determine the overall system thermal efficiency $\eta_{t,sys}$ and exergy efficiency $\eta_{e,sys}$. Since only one calculation moment is selected in this paper, the thermal efficiency and exergy efficiency of the heliostat field remain constant and are therefore not analyzed.

3.1 Receiver Performance Analysis

Figures 7 [Figure 7: see original paper] and 8 [Figure 8: see original paper] show the variations of thermal efficiency and exergy efficiency of the receiver

in the entire SPT system with turbine inlet temperature under the five S-CO₂ cycles. Figure 7 indicates that the thermal efficiency of the receiver gradually decreases with increasing turbine inlet temperature because higher turbine inlet temperature raises the average temperature of the S-CO₂ receiver's heat absorption surface, leading to intensified heat loss and reduced receiver thermal efficiency. At the same turbine inlet temperature, the simple recuperation S-CO₂ cycle has the largest heat absorption temperature difference, resulting in the lowest average temperature on the receiver surface and minimal heat loss, thus achieving the highest efficiency. The thermal efficiencies of receivers under partial-cooling, intercooling, recompression, and pre-compression cycles decrease sequentially. Figure 8 shows that the exergy efficiency of the receiver increases with turbine inlet temperature; however, when the temperature becomes too high, the average temperature of the receiver's heat absorption surface increases, causing severe heat loss and significant exergy loss, leading to a decline in exergy efficiency. Before the peak point, the receiver exergy efficiency under the simple recuperation cycle is the lowest, followed by partial-cooling, intercooling, pre-compression, and recompression S-CO₂ cycles in increasing order. After the peak point, the situation is reversed, which is caused by the different heat absorption temperature differences among the five S-CO₂ cycles.

3.2 Thermal Storage Subsystem Performance Analysis

Figures 9 [Figure 9: see original paper] and 10 [Figure 10: see original paper] show the variations of exergy efficiency during charging and discharging processes of the thermal storage subsystem with turbine inlet temperature. Both increase gradually with turbine inlet temperature. Since the pre-compression S-CO₂ cycle has the smallest heat absorption temperature difference, the thermal storage temperature difference of the thermal storage subsystem is the smallest under the same heat exchanger effectiveness, resulting in the smallest average heat exchange temperature difference between S-CO₂ and molten salt in the heat exchanger and thus the smallest irreversible exergy loss caused by heat exchange temperature difference, giving the highest exergy efficiency during charging. The simple recuperation S-CO₂ cycle shows the opposite situation. During discharging, the intermediate heat exchanger is fixed, and due to the energy quality degradation during the charging process, the heat exchanger effectiveness during discharging will be lower than 0.94. Therefore, limited by heat exchanger effectiveness, the exergy efficiencies of the thermal storage subsystem discharging process under the five S-CO₂ cycles are relatively close.

3.3 Power Cycle Subsystem Performance Analysis

Figures 11 [Figure 11: see original paper] and 12 [Figure 12: see original paper] show the variations of thermal efficiency η and exergy efficiency η_e of the five S-CO₂ cycles with turbine inlet temperature. As turbine inlet temperature increases, both thermal efficiency and exergy efficiency of the five S-CO₂ cycles increase. At each temperature, the intercooling S-CO₂ cycle has the highest

thermal and exergy efficiencies, while the simple recuperation cycle has the lowest. The thermal efficiency of the pre-compression S-CO₂ cycle is slightly higher than that of the partial-cooling cycle. The thermal and exergy efficiencies of the recompression S-CO₂ cycle are relatively sensitive to turbine inlet temperature changes; when the temperature exceeds 650 °C, both its thermal and exergy efficiencies surpass those of the pre-compression cycle. The exergy efficiency of the partial-cooling S-CO₂ cycle is less sensitive to turbine inlet temperature changes, showing small variations across the entire temperature range. When the temperature is above 700 °C, the exergy efficiencies of pre-compression and partial-cooling S-CO₂ cycles are very close.

3.4 Overall SPT System Performance Analysis

The performance of the receiver, thermal storage subsystem, and S-CO₂ power cycle subsystem directly determines the thermodynamic performance of the entire SPT system. The combined effect of these three components causes the thermal efficiency and exergy efficiency of the entire SPT system under the five S-CO₂ cycles to vary with turbine inlet temperature as shown in Figures 13 [Figure 13: see original paper] and 14 [Figure 14: see original paper]. Both thermal efficiency and exergy efficiency show a trend of first increasing and then decreasing with turbine inlet temperature. This is because at lower temperatures, the influence of power cycle subsystem efficiency dominates, and the overall system efficiency increases with turbine inlet temperature. However, at excessively high temperatures, heat loss from the receiver intensifies, and the influence of receiver efficiency dominates, causing the overall system efficiency to decrease. At each temperature, the SPT systems under intercooling and partial-cooling S-CO₂ cycles have relatively high efficiencies. Although the efficiency of the partial-cooling S-CO₂ cycle itself is lower than that of the recompression S-CO₂ cycle at high temperature ranges, its corresponding overall SPT system efficiency is always higher than that of the recompression S-CO₂ cycle due to its larger heat absorption temperature difference and higher receiver efficiency. At temperatures below 650 °C, the overall system efficiency under the pre-compression S-CO₂ cycle is higher than that under the recompression S-CO₂ cycle, while the opposite is true above 650 °C. The overall SPT system efficiency under the simple recuperation cycle is the lowest, about 4% lower than that under complex cycles.

Through the analysis of the thermodynamic performance of tower solar thermal power systems under the five S-CO₂ cycles, it can be found that each S-CO₂ cycle form has its own advantages and characteristics. The thermodynamic performance of the entire solar thermal power system is not only affected by the power cycle subsystem but also constrained by the receiver and thermal storage subsystem. Therefore, the thermodynamic performance of the S-CO₂ power cycle itself cannot be the sole focus; the optimal cycle form should be selected based on the specific characteristics and actual needs of the solar power plant.

4 Conclusions

This paper establishes a complete solar-thermal-work integrated model for direct-heated S-CO₂ tower solar thermal power systems (including the heliostat field, receiver, thermal storage subsystem, and power cycle subsystem). Based on this model, a comparative analysis of the thermodynamic performance of SPT systems under five S-CO₂ cycle forms—simple recuperation, pre-compression, recompression, partial-cooling, and intercooling—is conducted. The effects of turbine inlet temperature variation on the overall system thermal efficiency and exergy efficiency are mainly discussed. The conclusions are as follows:

- (1) As turbine inlet temperature increases, the thermal efficiency of the receiver gradually decreases, while its exergy efficiency first increases and then gradually decreases. The thermal and exergy efficiencies of the thermal storage subsystem and power cycle subsystem increase with turbine inlet temperature. The combined effect of these three components causes the thermal and exergy efficiencies of the entire SPT system not to increase continuously with turbine inlet temperature but to have a maximum value around 650 °C, indicating that higher operating temperature is not always better for the entire system.
- (2) Within the turbine inlet temperature range studied in this paper (500~800 °C), the SPT system under the intercooling S-CO₂ cycle has the highest thermal and exergy efficiencies. Although the efficiency of the partial-cooling S-CO₂ cycle itself is lower than that of the recompression S-CO₂ cycle above 650 °C, the efficiency of its corresponding entire SPT system is second only to that of the intercooling S-CO₂ cycle. This is because the partial-cooling S-CO₂ cycle has the advantage of a larger heat absorption temperature difference and higher receiver thermal efficiency. While SPT systems under intercooling and partial-cooling S-CO₂ cycles have higher efficiencies, their system structures are also the most complex.
- (3) In the temperature range below 650 °C, the overall SPT system efficiency under the pre-compression S-CO₂ cycle is higher than that under the recompression S-CO₂ cycle, while the opposite is true above 650 °C, indicating that both cycles have their own advantages. The recompression S-CO₂ cycle has a relatively simple layout and its efficiency is comparable to that of the partial-cooling S-CO₂ cycle at high temperature ranges, thus possessing great application potential.

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Note: Figure translations are in progress. See original paper for figures.

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