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The energy-saving mechanism of coal-fired power plant with S—CO_2 cycle compared to steam-Rankine cycle

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ABSTRACT

S—CO_2 (Supercritical-CO_2) coal-fired power plant is a promising technology for efficient and clean utilization of coal for power generation. The comparative study between the S—CO_2 coal-fired power plant and the power plant with steam Rankine cycle from aspects of energy and exergy balances is conducted. The conversion and transfer of the energy and exergy in the power plants are revealed. With the main gas parameters of 32MPa/620 °C and double-reheat process, the power generation efficiencies of the S—CO_2 coal-fired power plant and the power plant with steam Rankine cycle are 49.06% and 48.12%, respectively. The corresponding exergy efficiencies are 48.02% and 47.10%, respectively. The energy-saving mechanism of the S—CO_2 coal-fired power plant is revealed: the smaller boiler efficiency and larger exergy efficiency of the boiler system in the S—CO_2 coal-fired power plant make the energy level of the energy being transferred to the S—CO_2 cycle is higher than that of the energy being transported to the Rankine cycle. The CO_2 absorbs the high-level energy and produces more mechanical power through the S—CO_2 cycle to obtain higher power efficiency.

1. Introduction

Coal is the second source of primary energy (approximately 30%), and contributes over 40% of the worldwide electricity production [1]. For China, coal is the most important primary energy. Raw coal production and coal consumption account for 73.6% of the total primary energy production and approximately 65.6% of the total energy consumption in 2014 based on coal equivalent method [2]. Serious haze, 90% of the SO_2 emissions, 70% of the dust emissions, and 67% of the NO_x emissions were produced during coal combustion processes [3]. It is urgent to make efforts to improve the efficiency and reduce the emissions of coal-fired power plants.

Coal-fired power plants based on steam-Rankine cycle have been widely utilized for a long history. The parameters have been extremely improved over the past few decades to increase the efficiency of coal-fired power plants. Many technologies are employed, among which the double reheat ultra-supercritical (USC) technology is the most remarkable method [4]. Double reheat can increase efficiency by means of raising the average temperature of the steam parameters at turbine inlet (i.e. the endothermic process). Thus, the potential to produce work in the turbine is greatly improved while the exhaust steam humidity requirement of the turbine is fulfilled [5]. The thermal efficiencies of typical double-reheat USC power plants include Kawagoe Power Plant in Japan, Mannheim Power Plant in Germany and Nordjylland Power Plant in Denmark can reach over 45% on a lower heating value (LHV) basis [6]. Until the end of 2016, there are 6 double-reheat USC power plants in China. The unit 3 and unit 4 of Guodian Taizhou power plant were successfully operated from September 2015 and January 2016, respectively. The boiler efficiencies of the two units reach 94.78% and 95.12%, and the power generation efficiencies reach 47.81% and 47.95%, respectively [7]. The pressure and temperature of the fresh steam, reheat temperatures of Taizhou Power Plant are 31MPa/600 °C/620 °C/620 °C. Further improvement on the steam parameter is limited by the material problem. Thus, further enhancement on power generation efficiency of coal-fired power plant comes to be a large challenge.

Supercritical CO_2 (S—CO_2) cycle has been paid much attention in recent years. Compared to USC steam Rankine cycle, the S—CO_2

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Nomenclature

A	Energy level
AP	Air preheater
ar	As-received
C	Carbon
CON	Condenser
C_p	Specific heat capacity
DEA	Deaerator
e	Exergy
EUD	Energy Utilization Diagram
$e_{f,ph}$	Physical exergy of different flows
$e_{f,ch}$	Chemical exergy of different flows
FC	Fixed carbon
H	Hydrogen
h	Enthalpy
HRH	High-pressure regenerative heater
HT	High-pressure turbine
HTR	High temperature recuperator
HV	Heating value
i	Unit *i*
j	Component *j*
k	Composition *k*
L	Latent heat of vaporization
LHV	Lower heating value
LRH	Low-pressure regenerative heater
LT	Low-pressure turbine
LTR	Low temperature recuperator
M	Moisture
m	Mass flow rate
MC	Main compressor
MT	Mid-pressure turbine
N	Nitrogen
O	Oxygen
Q	Heat
R	Universal gas constant
RC	Re-compressor
S	Sulfur
s	Entropy
$S{-}CO_2$	Supercritical carbon dioxide
T_0	Environment temperature
V	Volatile matter
VHT	Very high-pressure turbine
W	work
w	Mass fraction
x	Mole fraction

Greek symbols

Δe	Exergy change
ΔEXL	Exergy destruction
ΔH	Enthalpy change
η_b	Boiler efficiency
η_c	Cycle efficiency
η_{eb}	Exergy efficiency of the boiler system
η_{ec}	Exergy efficiency of the power cycle
η_{ex}	Exergy efficiency
η_p	Power generation efficiency

cycle has more compact turbine machinery, and is more effective and flexible [8,9]. A brief development history of $S{-}CO_2$ cycle is presented in Fig. 1. The researches began with a patent for partially condensing CO_2 Brayton cycle by Sulzer Bros in 1948 [10]. The $S{-}CO_2$ Brayton cycle was further developed by researchers such as Feher and Angelino until the late 1970s [11,12]. During the mid-1970s to late-1990s, the researches were aimed at solving the problems within the fluid machinery and heat exchange caused by high-temperature and -pressure characteristics of $S{-}CO_2$ cycle [8]. From the beginning of the 21century, the $S{-}CO_2$ cycle has been extensively studied in basic theoretical analysis [13], process innovation optimization [14], boiler design and optimization [15], technical and economic analyses [16], and demonstration of small power plants. In the aspect of power generation system integration, many works have been done on integrating $S{-}CO_2$ cycle with solar energy [17], nuclear energy [9], gas turbine [18], and coal-fired power plant [19].

The energy structure of China determines that developing $S{-}CO_2$ coal-fired power plants is necessary and promising. Mecheri and Moullec investigated the influences of heat exchange pinches, pressure losses, and cycle configurations on the thermodynamic performance of the $S{-}CO_2$ coal-fired power plant. The efficiency of the system reaches 48% under the condition of 30MPa/620 °C and a double reheat single recompression cycle [20]. Xu et al. emphasized the two key issues of large boiler pressure drops and residual flue gas heat extraction of $S{-}CO_2$ coal-fired power plant [21]. The 1/8

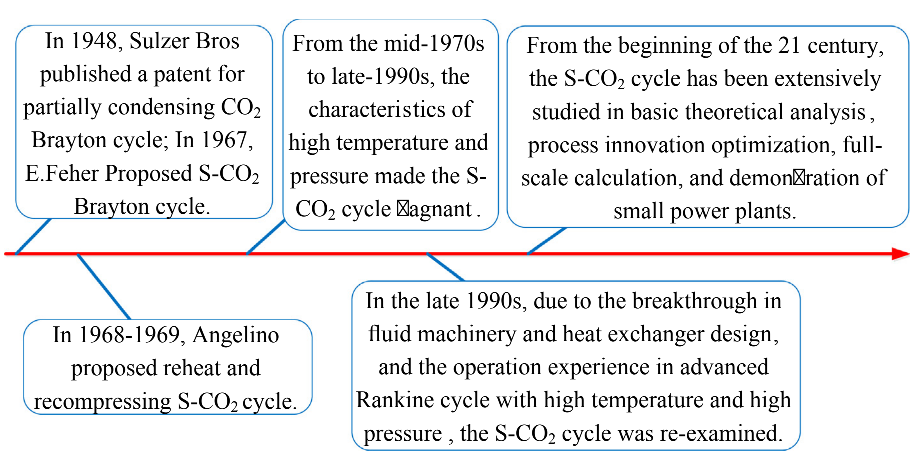

Fig. 1. The brief development history of $S{-}CO_2$ cycle.

principle to reduce the frictional pressure drop in the boiler and a flue gas cooler arrangement to extract the flue gas heat were proposed. The power generation efficiency of the power plant reaches 48.37% under the condition of 30MPa/620 °C and a double reheat single recompression cycle. Zhou et al. conducted the exergy analysis of a single reheat S–CO_2 Brayton cycle coal-fired power plant [22]. The results showed that the exergy loss ratios of the S–CO_2 boiler system and fuel combustion process are 82.2% and 53.5%, respectively. The former is close to that of the steam boiler, but the latter is about 5.3% higher than the traditional steam boiler. Ishiyama et al. compared the power generation systems with integrated steam, helium, S–CO_2 cycles and prototype nuclear fusion reactor [27]. The S–CO_2 Brayton cycle is recommended due to its efficiency and turbine compactness.

Viewing from the literature review, many works have been done in the aspects of energy and exergy analyses on the S–CO_2 coal-fired power plant. The comprehensive comparisons between the S–CO_2 Brayton cycle and the steam Rankine cycle were limited to constant temperature heat sources such as nuclear and solar energy. The comparison between the coal-fired power plant with S–CO_2 Brayton cycle and steam Rankine cycle is mainly from aspect of energy analyses. The energy analyses should be combined with exergy analyses to reveal the energy conversion and transfer routes in the power plants, and then obtain the energy-saving mechanism of S–CO_2 coal-fired power plant. Especially for China, the replacement of steam Rankine cycle with S–CO_2 cycle for the coal-fired power plant is a large project, and needs to be carefully expounded and proved.

This paper investigates the energy-saving mechanism of S–CO_2 coal-fired power plant against conventional coal-fired power plant with steam Rankine cycle. The comparison is carried out mainly from aspects of the first law and the second law of thermodynamics. The energy utilization diagram (EUD) methodology is implemented for key processes and unis to reveal the origins of exergy destructions.

2. Proposal of the technologies

S–CO_2 Brayton cycle technology is a power generation technology with supercritical CO_2 as the working medium of the power cycle. The operation parameters of CO_2 are beyond its critical point (7.38MPa/30.98 °C). The compression processes should be conducted near the critical point due to the low work consumption here [12]. In the other hand, considering the utilization of the double-reheat technology in large-scale coal-fired power plant with steam Rankine cycle, the double-reheat technology is also adopted in the S–CO_2 coal-fired power plant. The processes simulations have been performed with Aspen Plus software. Among property methods such as PENG-ROB, PR-BM, RK-SOAVE, SRK, BWRS and LK-PLOCK, the LK-PLOCK property method was proved to exhibit satisfactory results and revealed the best trends near the critical point compared to REFPROP. LK-PLOCK is also more accurate at high pressure and temperature [20]. Thus, LK-PLOCK property method is selected to perform the simulations in this paper.

The flow sheet of S–CO_2 coal-fired power plant with double-reheat single recompression cycle is illustrated in Fig. 2. The S–CO_2 Brayton cycle is composed of:

- {1–6} turbine {total flow}
- {6–7} high temperature recuperator (HTR) hot side {total flow}
- {7–8} low temperature recuperator (LTR) hot side {total flow}
- {8–10} heat sink {part flow}
- {8–13} re-compressor {part flow}
- {10–11} main compressor (after the heat sink) {part flow}
- {11–12} low temperature recuperator (LTR) cold side {part flow}

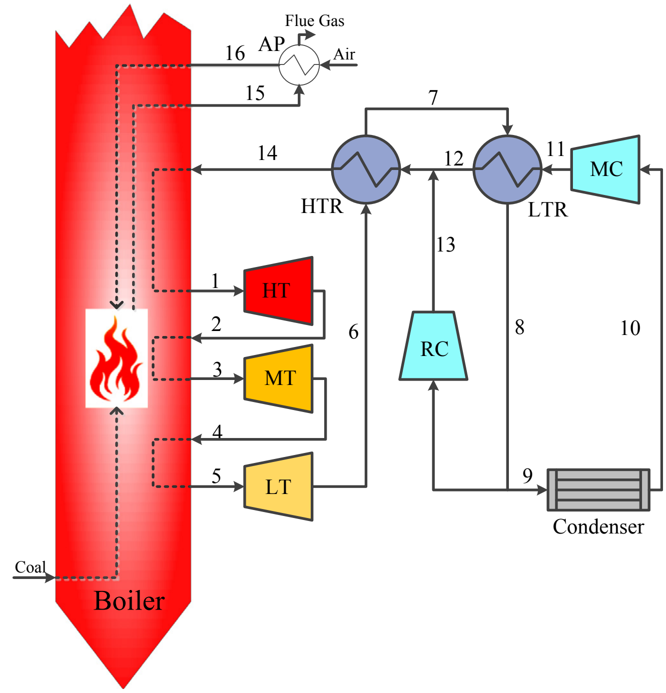

Fig. 2. The flow sheet of S–CO_2 coal-fired power plant with double-reheat single recompression cycle.

- {12–14} high temperature recuperator (HTR) cold side {total flow}

The fuel coal is combusted with the pre-heated air in the boiler. Most of the high-temperature heat is transferred to the cooling wall of the boiler by means of radiation and convection. The rest of the heat is partially absorbed by the air in the air-preheater (AP) and the remaining is discharged into the atmosphere with a temperature of 120 °C. S–CO_2 from the HTR absorbs the radiation and convection heat in the tube on the cooling wall of the boiler, and is heated to 620 °C. The high-temperature and –pressure CO_2 enters the high-pressure turbine (HT) to generate electricity. Then, the CO_2 goes through a double-reheat process, and generate electricity in the mid-pressure turbine (MT) and low-pressure turbine (LT), respectively. The parameters of CO_2 at the outlet of the LT are 7.9MPa/561.75 °C. The CO_2 flows into the HTR and LTR for two-stage heat recovery. After the heat recovery process, the CO_2 flow is split into two separate flows. The first flow (68.3% of the total flow) is cooled in the condenser and compressed to 33.45 MPa in the main compressor (MC). Then this flow is heated to 230 °C in the LTR by the recovered heat. The second flow (31.7% of the total flow) is compressed to 33.4 MPa in the re-compressor (RC), and mixes with the first flow at the outlet of the LTR. The mixed total flow is heated to 515.5 °C in the HTR by the recovered heat.

In this paper, the coal-fired power plant with integrated steam Rankine cycle and 10-stage heat recovery and double-reheat processes is chosen as the reference power plant. The flow sheet is shown in Fig. 3. The parameters are 32MPa/620 °C for the main steam, 620 °C/620 °C for the reheat steams. The turbines consist of one single-flow very high-pressure turbine (VHT), one double-flow high-pressure turbine (HT), one double-flow mid-pressure turbine (MT), and two double-flow low-pressure turbines (LT).

The steam outlets of the VHT and HT undergo two reheating processes and are transferred to the MT and LT for further expansion. The power plant adopts a 10-stage regenerative system with four high-pressure regenerative heaters (HRHs), one deaerator

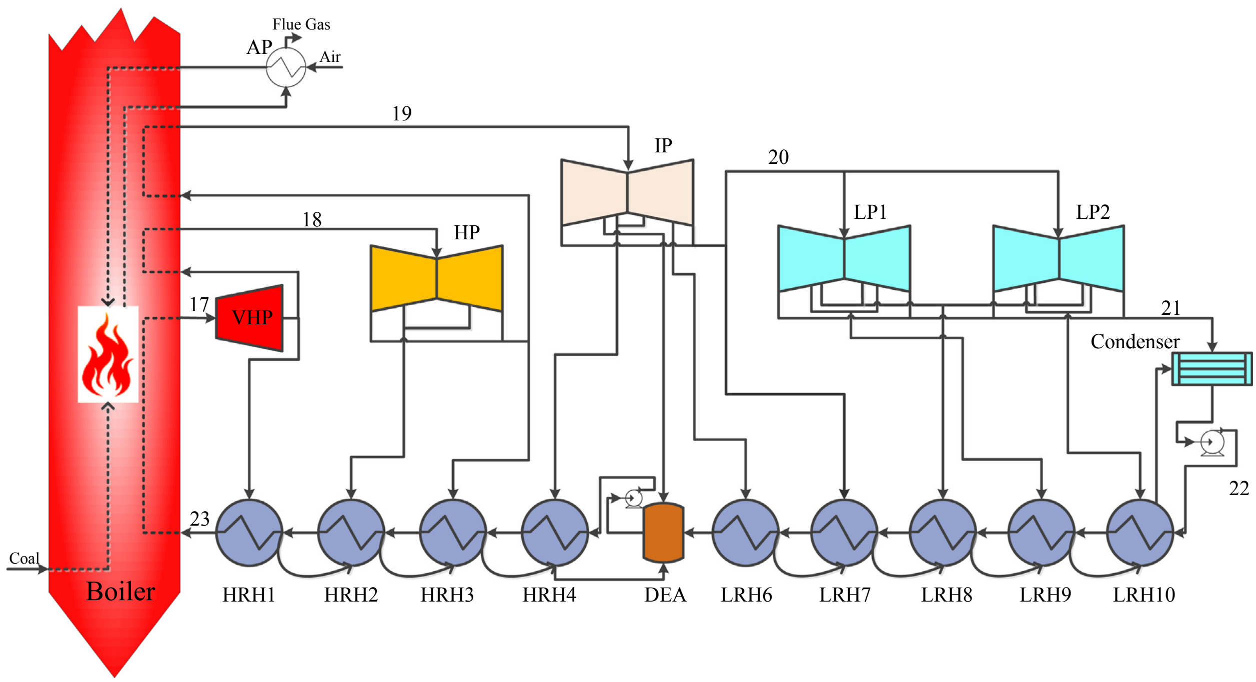

Fig. 3. The flow sheet of coal-fired power plant with steam Rankine cycle.

(DEA), and five low-pressure regenerative heaters (LRHs). The pressure of the exhaust steam from the LT into the condenser is set as 0.045 bar.

3. Methodology

The energy and exergy analyses are conducted out for comprehensive comparison of S–CO_2 and steam Rankine coal-fired power plants. The mass balance, energy balance, and exergy balance for certain unit i can be expressed as follows:

$$\sum_{in} \dot{m}_{ij} = \sum_{out} \dot{m}_{ij} \tag{1}$$

$$\sum_{in} m_{ij} h_{ij} - \sum_{out} m_{ij} h_{ij} = W_i + Q_i \tag{2}$$

$$\Delta EXL_i + W_i + e_{Q_i} = \sum_{in} m_{ij} e_{ij} - \sum_{out} m_{ij} e_{ij} \tag{3}$$

While $\dot{m}_{ij}$, h_{ij}, and e_{ij} are the mass flow rate, specific enthalpy, and specific exergy of component j at the inlet or outlet of unit i. W_i is the work output of unit i. W_i is positive for turbines, negative for compressors and pumps, and is zero for other units. Q_i is the heat released from unit i. Q_i is positive for condensers. ΔEXL_i is the exergy destruction of unit i; e_{Q_i} is the exergy of Q_i.

The energy balances of the power plants are based on the energy balances of each unit. Particularly, for the energy balance of the boiler, the enthalpy of the flue gas exhausted into the atmosphere should be calculated by the following equation because the lower heating value of the fuel coal is adopted:

$$h_{fluegas} = h^0_{fluegas} + (C_p)_{fluegas}\left(T_{fluegas} - T_0\right) - \left(\dot{m}_{H_2O} - \dot{m}_{coal} M_{ar}\right) L_{H_2O} + \mathrm{HV_{CO}} \tag{4}$$

The sensible heat of the flue gas is calculated by: $h^0_{fluegas} + (C_p)_{fluegas}(T_{fluegas} - T_0)$;

The H_2O in the flue gas comes from two aspects: the moisture exists in the coal, and the H_2O formed by the hydrogen combustion. Because the lower heating value of the coal is used in the calculation, the latent heat of the H_2O formed by the hydrogen combustion should not be contained in the enthalpy of the flue gas. Thus, the latent heat of the H_2O formed by the hydrogen combustion is calculated by: $(\dot{m}_{H_2O} - \dot{m}_{coal} M_{ar}) L_{H_2O}$;

The heating value of the CO in the flue gas is calculated by: $\mathrm{HV_{CO}}$.

Where $h_{fluegas}$ is the enthalpy of the flue gas; $h^0_{fluegas}$ is the enthalpy of the flue gas at standard status; $(C_p)_{fluegas}$ is the average specific heat capacity of the flue gas between T_0 to $T_{fluegas}$; $T_{fluegas}$ is the temperature of the flue gas; $\dot{m}_{H_2O}$ is the mass flow rate of H_2O in the flue gas; M_{ar} is the mass fraction of H_2O in the as-received fuel coal; L_{H2O} is the latent heat of vaporization for H_2O; HV_{CO} is the heating value of CO.

The exergy of the coal and particular flow is calculated by:

$$e_{coal} = LHV_{coal} \bullet \left(1.0064 + 0.1519\frac{w_H}{w_C} + 0.0616\frac{w_O}{w_C} + 0.0429\frac{w_N}{w_C}\right) \tag{5}$$

$$e_f = e_{f,ph} + e_{f,ch} = (h - h_0) - T_0(s - s_0) + \sum x_k e_{f,k} + RT_0 \sum x_k \ln x_k = \sum x_k[((h - h_0) - T_0(s - s_0))]_k + \sum x_k e_{f,k} + RT_0 \sum x_k \ln x_k \tag{6}$$

In the equations, w_H,w_O, w_N,w_C are the mass fractions of elements H, O, N, C in the coal; e_{coal} is the specific exergy of the fuel coal, kJ/kg; e_f is the specific exergy of different flows in the system; $e_{f,ph}$ and $e_{f,ch}$ are the physical exergy and chemical exergy of different flows in the system; $e_{f,k}$ and x_k are the standard chemical exergy and the mole fraction of the composition k in the flow.

In accordance with the energy balance and exergy balance equations, the power generation efficiency and exergy efficiency of the overall system can be expressed as follows:

$$\eta_p = \frac{W_{output}}{\dot{m}_{coal} \times LHV_{coal}} \quad (7)$$

$$\eta_{ex} = \frac{W_{output}}{e_{coal}} \quad (8)$$

The ultimate and proximate analyses of the fuel coal are listed in Table 1. The steams extraction parameters of the 10-stage heat recovery in the coal-fired power plant with steam Rankine cycle are illustrated in Table 2.

The following assumptions are made to simplify the simulation of the power plants:

- The minimum temperature approaches are set to 5 °C for recuperators and 30 °C (±1 °C) for air-gas heat exchangers [22].
- The pressure drops of the HTR and LTR are 0.05 MPa [25], and the pressure drops for the S–CO_2 boiler and the water boiler are 1.5 MPa and 5.9 MPa [26], respectively.
- Neglect the shaft seal loss, mechanical loss, the pressure loss of pipelines and separators [22].
- The isentropic efficiencies of the turbomachinery and compressors are set to 93% [24] and 90% [24,25], respectively. And the compressors are assumed to be driven by turbine shaft and not by electrical motors.

All other parameters are determined by process constraints after the main parameters are set. The main parameters used in the simulation are concluded in Table 3.

The parameters of key flows in the power plants (Figs. 2 and 3) are illustrated in Table 4. As can be seen in Table 4, in the S–CO_2 coal-fired power plant, the temperature of the exhaust CO_2 at the outlet of LT is as high as 561.75 °C due to the small expansion ratio of CO_2 through the turbines. Thus, the heat recovery processes should be implemented to maintain high energy efficiency.

The temperatures of pre-heated air enters the boilers are 506.53 °C and 263.64 °C, respectively. In current conditions, the high-temperature characteristic of the pre-heated air is a challenge faced by the S–CO_2 coal-fired power plant. In the other hand, under the condition of same boiler heat load, the mass flow rate of the CO_2 in the boiler is 39.565 kg/s, which is almost 10 times that of the H_2O. The larger mass flow rate would cause larger pressure drop when CO_2 goes through the boiler. According to the 1/8 principle proposed by Xu et al. [13], the pressure drop of the S–CO_2 in the boiler is set to be 1.5 MPa, which is nearly 1/4 that of the water in the boiler. The pressure drops of the CO_2 and H_2O in the boilers are shown in Table 3.

4. Results and discussions

4.1. Energy balances of the power plants

The energy balances of the power plants are based on the energy balances of each unit. The energy balance diagram of the S–CO_2 coal-fired power plant is illustrated in Fig. 4. In the figure, the widths of the blocks and lines represent the amount of energy. The only energy input of the power plant is the enthalpy of the fuel coal, 12.70 MW.

The electricity generated by the HT, MT and LT is 8.63 MW. However, the main compressor and the re-compressor consume 1.07 MW and 1.33 MW, respectively. Thus, the work output is 6.23 MW, leading to a net power generation efficiency of 49.06%. The exhaust energy of the boiler is 1.11 MW, including the enthalpy of the high-temperature ash and enthalpy of the flue gas. Thus, the boiler efficiency can be calculated by the ratio of heat absorbed by the working fluids in the power cycles and the enthalpy of the coal, which is 91.26% for the boiler in the S–CO_2 coal-fired power plant. The exhaust energy of the condenser in the S–CO_2 cycle is 5.36 MW. The heat exchange capacities of the HTR and LTR are 14.54 MW and 7.14 MW, respectively. The recuperators should be efficiently designed to satisfy the large-capacity heat exchanges.

The energy balance diagram of the coal-fired power plant with steam Rankine cycle is illustrated in Fig. 5. Same to the S–CO_2 coal-fired power plant, the energy input of the power plant is only the enthalpy of the fuel coal. The electricity generated by the VHT, HT, MT and LT is 6.41 MW. The pumps in the power plant consume electricity of 0.30 MW. Thus, the work output of the power plant is 6.11 MW, making a net power generation efficiency of 48.12%.

The exhaust energy of the boiler is 0.83 MW, therefore the boiler efficiency is (12.70–0.83)/12.70 = 93.46%. The exhaust energy in the condenser is 5.76 MW. The heat exchange capacities of the HRHs, DEA, and LRHs are 5.68 MW. The enthalpy of the feed water enters the boiler is 5.98 MW, with a temperature of 300 °C.

The heat exchange capacity distribution of the regenerative heaters is shown in Fig. 6. The total heat exchange capacity is 5.68 MW. The largest heat exchange capacity locates in the DEA, which accounts for approximately 22.31% of the total heat exchange capacity. The heat exchange capacity of the 5 LRHs is close to each other.

As can be seen in Table 5, the boiler efficiency of the S–CO_2 coal-fired power plant is lower than that of the power plant with steam Rankine cycle. In the other hand, the efficiency of the S–CO_2 Brayton cycle is higher than that of the steam Rankine cycle. As a result, the net power generation efficiency of the S–CO_2 coal-fired power plant is larger than that of the coal-fired power plant with steam Rankine cycle.

In our research, the boiler efficiency of the S–CO_2 coal-fired power plant is only 91.26%, which is smaller than that of the power plant with steam Rankine cycle (93.46%). This is mainly caused by the larger incomplete combustion loss of CO in the S–CO_2 coal-fired power plant, as can be seen in Fig. 7. Due to the higher temperatures of CO_2 entering the boiler (515.48 °C) and the pre-heated air (506.53 °C) in the S–CO_2 coal-fired power plant than that of the H_2O entering the boiler (300 °C) and the pre-heated air (263.64 °C), the average temperature in the S–CO_2 boiler is higher than that in the water steam boiler. According to the chemical reaction (R1),

Table 1
Ultimate analysis and proximate analysis of the fuel coal.

Ultimate analysis, wt%		Proximate analysis, wt%	
C_{ar}	68.55	M_{ar}	8.84
H_{ar}	3.96	Ash_{ar}	9.98
O_{ar}	6.85	V_{ar}	49.52
N_{ar}	0.74	FC_{ar}	31.66
S_{ar}	1.08	LHV_{coal}, MJ/kg	26.51

Table 2
The parameters of the steam extractions of the regenerative heaters.

	Temperature/°C	Pressure/bar	Mass flow/kg/s	Extraction ratio
HRH1	396.47	93	0.333	0.08
HRH2	574.46	70	0.257	0.067
HRH3	499.13	45	0.214	0.06
HRH4	531.06	25	0.161	0.048
DEA	450.55	15	0.096	0.03
HRH6	360.73	8	0.152	0.049
HRH7	266.72	3.8	0.144	0.049
HRH8	168.52	1.53	0.140	0.05
HRH9	89.31	0.6	0.133	0.05
HRH10	62.09	0.18	0.132	0.052

Table 3
The values of key parameters in the calculation.

Parameters	values	Parameters	values
Pressure/temperature of the fresh steam and S–CO_2/MPa/°C	32/620	Isentropic efficiency of turbines/%	93
Minimum temperature approaches of the boilers/°C	30	Double-reheat temperatures/°C	620
Minimum temperature approaches of the heat exchangers/°C	5	Isentropic efficiency of compressors/%	90
Excess air ratio in the boiler	1.3	Pump mechanical efficiency/%	99
Pressure drop in S–CO_2 boiler/MPa	1.5	Pressure drop in water boiler/MPa	5.9
Pressure drop in HTR and LTR/MPa	0.05		

Table 4
The key parameters in the power plants.

	Temperature/°C	Pressure/bar	Mass flow/kg/s
Coal	25.00	1.0	0.479
Air	25.00	1.0	5.534
1	620.00	320.0	39.565
2	550.77	192.3	39.565
3	620.00	192.3	39.565
4	560.58	123.3	39.565
5	620.00	123.3	39.565
6	561.75	79.0	39.565
7	235.00	79.0	39.565
8	86.58	79.0	39.565
9	86.58	79.0	27.023
10	32.50	79.0	27.023
11	80.74	334.5	27.023
12	230.00	334.0	27.023
13	229.66	334.0	12.542
14	515.48	333.5	39.565
15	543.48	1.0	5.965
16	506.53	1.0	5.534
17	620.00	320.0	4.171
18	620.00	90.0	3.837
19	620.00	42.0	3.365
20	266.72	3.8	2.814
21	36.12	0.045	2.408
22	24.72	373.0	3.112
23	300.00	373.0	4.171
24	263.64	1	5.534
Flue gas	120.00	1.0	5.965

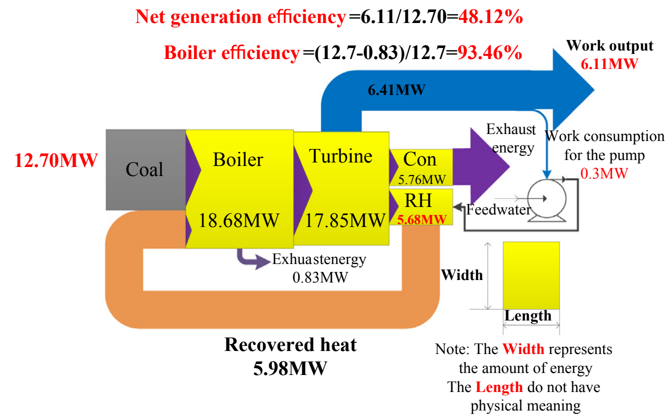

Fig. 5. The energy balance diagram of coal-fired power plant with steam Rankine cycle.

more CO would exist in the flue gas of the S–CO_2 boiler, which causes larger incomplete combustion loss. To avoid this and improve the S–CO_2 boiler efficiency, new boiler configuration design should be adopted.

This part presents the comparison on the energy balances of the power plants. Due to the larger incomplete combustion of CO caused by higher average temperature in the boiler, the boiler efficiency of S–CO_2 boiler is smaller than that of the water steam

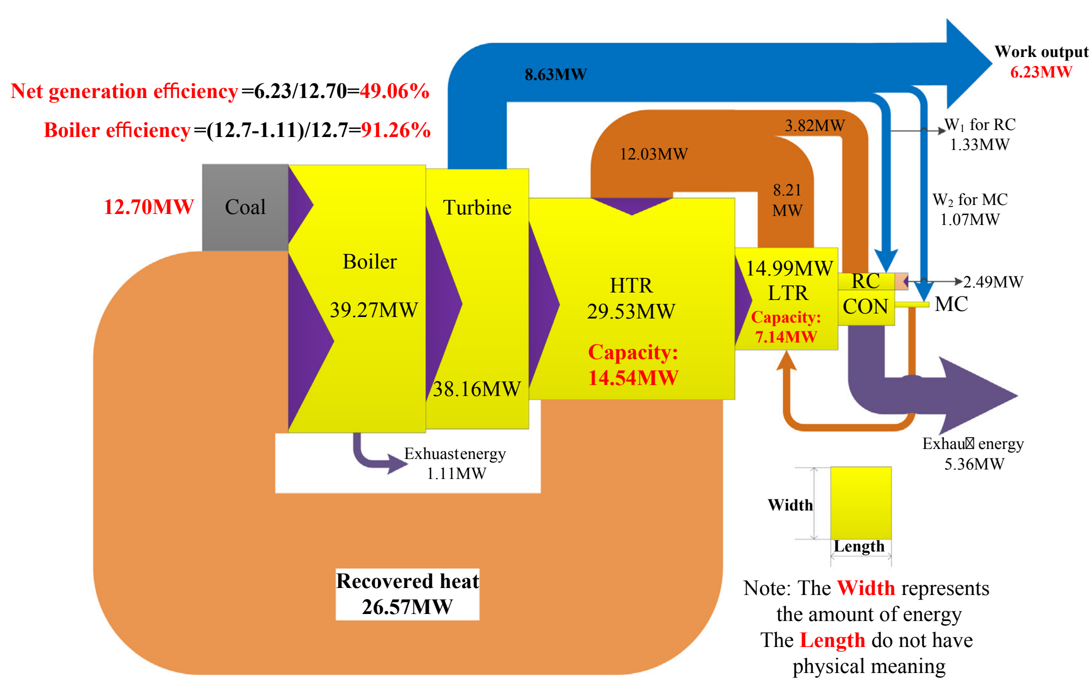

Fig. 4. The energy balance diagram of S–CO_2 coal-fired power plant.

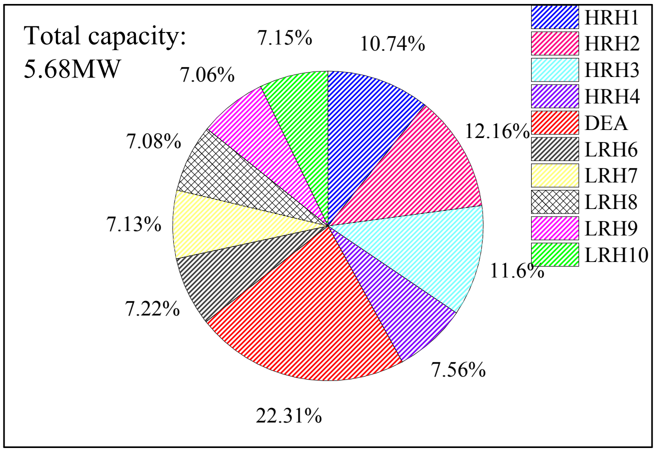

Fig. 6. The heat exchange capacity distribution of the regenerative heaters.

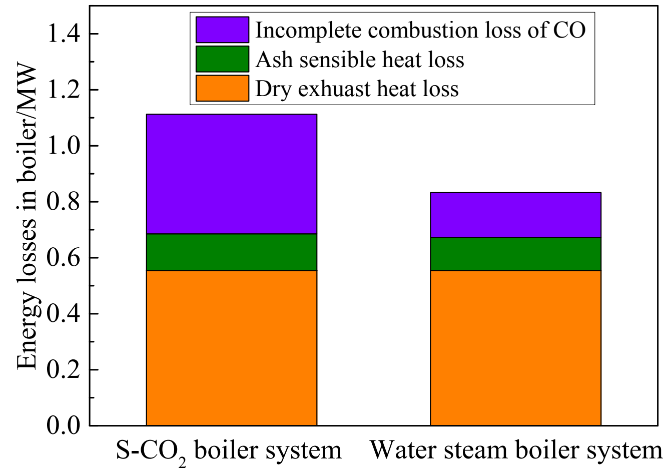

Fig. 7. The energy loss distribution in boiler of the power plants.

Table 5
The comparison on relative efficiencies of power plants with $S{-}CO_2$ and steam Rankine cycles.

	Power plant with $S{-}CO_2$ cycle	Power plant with Rankine cycle
Boiler efficiency/%	91.26	93.46
Cycle efficiency/%	53.76	51.49
Net power efficiency/%	49.06	48.12

boiler. However, benefiting from the higher cycle efficiency of $S{-}CO_2$ cycle than that of the steam Rankine cycle, the net power generation efficiency of the $S{-}CO_2$ coal-fired power plant reaches 49.06%, which is higher than that of the coal-fired power plant with steam Rankine cycle (48.12%).

4.2. *Exergy balances of the power plants*

The exergy of the flows in the flow sheets of the technologies, exergy destructions in different processes or units, and the exergy efficiencies of the power plants are calculated by equations (3), (5), (6) and (8). According to the equations, the exergy balances of the $S{-}CO_2$ coal-fired power plant and power plant with steam Rankine cycle are listed in Table 6.

The exergy inputs of the power plants are 12.975 MW. The exergy outputs are considered to only be the work output of the power plants. The exergy of the flue gas and ash is handled to be the exergy loss of the power plants. As can be seen in Table 6, the exergy outputs of the $S{-}CO_2$ coal-fired power plant and the power plant with steam Rankine cycle are 6.231 MW and 6.111 MW, respectively. Thus, the exergy efficiencies of the power plants are 48.02% and 47.10%, respectively. The largest exergy destructions exist in the boilers in both power plants, which account for approximately 34.39% and 40.86% of total exergy input for the $S{-}CO_2$ coal-fired power plant and the power plant with steam Rankine cycle, respectively.

The exergy destructions of the boiler systems (include the exergy destruction in the boiler, the air-preheater, and the exergy loss of the flue gas) of the $S{-}CO_2$ coal-fired power plant and the power plant with steam Rankine cycle are 5.582 MW and 6.121 MW, respectively. Thus, the exergy efficiencies of the boiler systems are 56.98% and 52.82%, respectively. As can be seen in Fig. 8, the exergy destruction distributions of the boiler systems are illustrated. The largest exergy destructions exist in the coal combustion processes, which are 2.653 and 3.109 MW for $S{-}CO_2$ boiler and water steam boiler systems, respectively. Due to the higher average combustion temperature in the $S{-}CO_2$ boiler, the energy level difference between the coal combustion reaction and the high-temperature flue gas is smaller than that in the water steam boiler. Thus, the exergy destruction of the coal combustion process in the $S{-}CO_2$ boiler is smaller. The exergy loss of the flue gas in the $S{-}CO_2$ boiler system is larger due to the existence of more CO in the flue gas. For the air-preheater, the heat exchange capacity of the AP in the $S{-}CO_2$ boiler system is much larger than that in the water steam boiler system, which leads to larger exergy destruction in the air-preheating process.

Combined with energy balance of the power plants, we can see that despite the boiler efficiency of the $S{-}CO_2$ boiler is smaller than that of the water steam boiler, the exergy efficiency of the $S{-}CO_2$ boiler system is higher than that of the water steam boiler. The exergy destructions of other units in the power plants are relative small. The exergy efficiencies of the $S{-}CO_2$ cycle and steam Rankine

Table 6
The exergy balances of the power plants.

Items	$S{-}CO_2$ power plant		Power plant with steam Rankine cycle	
	Values/MW	Proportion/%	Values/MW	Proportion/%
Exergy input				
Coal	12.975	100	12.975	100
Total	12.975	100	12.975	100
Exergy output				
Work output	6.231	48.02	6.111	47.10
Exergy Destruction				
Boiler	4.465	34.41	5.300	40.84
Air-preheater	0.223	1.72	0.173	1.33
VHT			0.051	0.39
HT	0.098	0.76	0.024	0.19
MT	0.074	0.57	0.078	0.60
LT	0.077	0.59	0.113	0.88
HTR	0.296	2.28		
LTR	0.088	0.68		
HRHs			0.103	0.79
LRHs			0.106	0.81
Condenser	0.356	2.74	0.204	1.57
MC	0.09	0.69		
RC	0.08	0.62		
Pumps			0.066	0.51
Flue gas	0.897	6.91	0.647	4.99
Total	6.744	51.98	6.864	52.90
Exergy efficiency/%	48.02		47.10	

cycle are 84.28% and 89.16%, respectively. For the condensers, although the energy losses of the S—CO_2 coal-fired power plant and the power plant with steam Rankine cycle are separately 5.36 MW and 5.76 MW, the exergy destructions are only 0.356 MW and 0.204 MW, respectively.

4.3. Energy-saving mechanism of S—CO_2 coal-fired power plant

The graphical exergy analyses (EUD methodology) method was firstly proposed by Ishida [23], and is implemented to analyze the origins of the exergy destructions. The energy level of a process is defined as:

$$A = \frac{\Delta e}{\Delta H} = 1 - T_0 \times \Delta S \big/ \Delta H. \quad (9)$$

where A is the energy level of the process; Δe, ΔS and ΔH are the exergy change, entropy change and enthalpy change during the process, respectively. T_0 is the environmental temperature. For an energy-transformation process, there exist an energy donor (A_{ed}) and an energy acceptor (A_{ea}):

$$A_{ed} = \frac{\Delta e_{ed}}{\Delta H_{ed}} \quad (10)$$

$$A_{ea} = \frac{\Delta e_{ea}}{\Delta H_{ea}} \quad (11)$$

And $\Delta H_{ea} + \Delta H_{ed} = 0$, (12)

Thus, the total exergy destruction during the energy-transformation process is:

$$\Delta e_{ea} + \Delta e_{ed} = A_{ed} \bullet \Delta H_{ed} + A_{ea} \bullet \Delta H_{ea} = (A_{ed} - A_{ea}) \bullet \Delta H_{ed} \quad (13)$$

For a continuous energy-transformation process, the exergy destruction can be obtained by the integral form:

$$\Delta e = \int (A_{ed} - A_{ea}) \bullet dH \quad (14)$$

In the energy-utilization diagram (EUD diagram), the x-coordinate is energy change, and the y-coordinate is energy level A, which is a dimensionless criterion. So the exergy destruction is illustrated by the shaded areas between the curves of the energy donor and energy acceptor. In the following part, the EUD methodology is implemented to analyze the energy-saving mechanism of S—CO_2 coal-fired power plant.

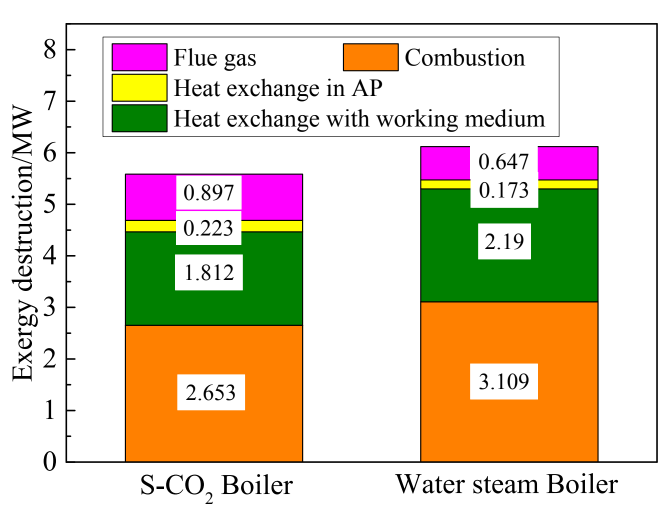

Fig. 8. The exergy destructions distribution in boiler systems.

According to the results about the energy and exergy balances of the S—CO_2 coal-fired power plant and the coal-fired power plant with steam Rankine cycle, the energy and exergy transfer routes in the power plants are illustrated in Fig. 9. The energy-saving mechanism of the coal-fired power plant with S—CO_2 cycle compared to steam-Rankine cycle can be revealed. The energy transfer in both power plants is divided into three processes: energy transfer in the boiler system, energy transfer in the power cycles, and energy transfer between the boiler system and the power cycle. Actually, the energy transfer between the boiler system and the power cycle is realized in the cooling wall, superheaters, and re-heaters throughout the boiler.

In Fig. 9, A_{coal} is the energy level of the fuel coal; $A_{com\ pro}$ is the energy level of the combustion product; A_t is the energy level of the energy being transferred between the boiler system and the power cycle; η_b, η_{eb}, η_c, and η_{ec} are the boiler efficiency, exergy efficiency of the boiler system, cycle efficiency and exergy efficiency of the power cycle, respectively. h_{coal} is the enthalpy of the fuel coal.

As can be seen in Fig. 9, the energy levels of the coal in both power plants are the same value (1.02). In the coal combustion process, the energy donor is the coal combustion reaction, and the energy acceptor is the combustion product. The energy level of the combustion product in the S—CO_2 boiler is 0.81, which is higher than that in the water steam boiler (0.78). This is mainly caused by the higher combustion temperature in the S—CO_2 boiler. In the other hand, the exergy destruction of the coal combustion process in the S—CO_2 boiler is smaller due to the higher energy level of the combustion product. After the energy is transferred from the coal combustion reaction to the combustion product, most of the energy ($\eta_b h_{coal}$) is absorbed by the work medium in the cooling wall, superheaters, and re-heaters. Although the amount of the energy being transferred between the boiler system and the power cycle in the S—CO_2 power plant is smaller than that in the power plant with Rankine cycle, the energy level is higher (0.64) than that in the power plant with Rankine cycle (0.58). In other words, more exergy exists in the energy being transported from the boiler system to the S—CO_2 than to the steam of the Rankine cycle. The energy with higher energy level in the S—CO_2 coal-fired power plant produces more electricity (6.231 MW) than that in the power plant with steam Rankine cycle (6.111 MW), which results in the higher entire power generation efficiency of the S—CO_2 coal-fired power plant (49.06%) than the power plant with steam Rankine cycle (48.12%).

In conclusion, the exergy efficiency of the S—CO_2 boiler system (57.0%) is higher than that of the water steam boiler system (52.8%), but the boiler efficiency shows the opposite result (91.3% and 93.5% for S—CO_2 boiler system and water steam boiler, respectively). Thus, although the amount of energy being transferred from the boiler system to the work medium is smaller, more available energy (exergy) is transported to the work medium in the S—CO_2 coal-fired power plant. In other words, the energy level of the energy being transferred between the boiler system and the power cycle in the S—CO_2 coal-fired power plant is higher than that in the power plant with steam Rankine cycle. In the other hand, the cycle efficiency of the S—CO_2 cycle (53.8%) is higher than that of the Rankine cycle (51.5%), and the exergy efficiency denotes the adverse trend (84.3% and 89.2% for the S—CO_2 cycle and steam Rankine cycle, respectively). Thus, the energy with higher energy level being transferred from the boiler system to the power cycle produces more work in the S—CO_2 cycle, which contributes to higher power generation efficiency of the entire S—CO_2 coal-fired power plant.

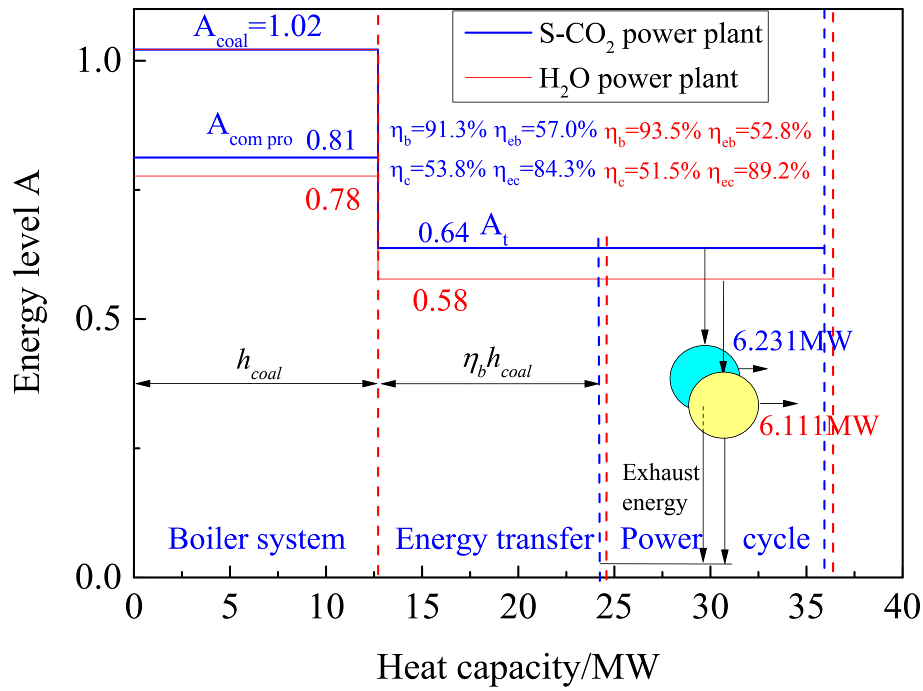

Fig. 9. The energy-saving mechanism diagram of S$-$$CO_2$ coal-fired power plant.

5. Conclusion

Adopting supercritical CO_2 cycle as the bottom cycle of coal-fired power plant is a promising technology for efficient and clean utilization of coal for power generation. The comparison between the S$-$$CO_2$ coal-fired power plant and the power plant with steam Rankine cycle from aspects of energy and exergy balances are made to reveal the energy-saving mechanism of S$-$$CO_2$ coal-fired power plant.

The S$-$$CO_2$ coal-fired power plant has advantages in power generation efficiency (49.06%) over the power plant with steam Rankine cycle (48.12%) under condition of 32MPa/620 °C/620 °C and double-reheat process. The corresponding exergy efficiencies are 48.02% and 47.10%, respectively.

The energy-saving mechanism of the S$-$$CO_2$ coal-fired power plant compared to the power plant with steam Rankine cycle is revealed. Due to the smaller boiler efficiency and larger exergy efficiency of the boiler system in the S$-$$CO_2$ coal-fired power plant, the energy level of the energy being transferred to the S$-$$CO_2$ cycle is higher than that of the energy being transported to the Rankine cycle. The energy absorbed by the CO_2 produces more mechanical power through the S$-$$CO_2$ cycle to obtain higher power generation efficiency.

Declaration of competing interest

None.

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