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Design and Performance Analysis of New Ultra-Supercritical Double Reheat Coal-Fired Power Generation Systems

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Abstract: In order to solve the existing problems of large mean heat transfer temperature differences of regenerative air heaters and high superheat degrees of regenerative extraction steam in double reheat coal-fired power generation systems, two new design schemes of ultra-supercritical double reheat cycles are proposed, which can realize the deep boiler-turbine coupling among the heat transfer processes of air, feeding water and regeneration extraction steam on the base of the principle of energy level matching. A typical 1000 MW ultra-supercritical double reheat cycle system is selected as the reference system and the overall system model is built by using the Ebsilon simulation software. The performances of two new systems are analyzed by using both the exergy method and energy equilibrium method. The results show that net output powers of both new systems 1 and 2 increase by 6.38 MW and 6.93 MW, respectively, and the standard coal consumptions of power generation decrease by 1.65 g/kWh and 1.79 g/kWh, respectively. The off-design performances of new systems and the reference system are analyzed, and the results show that performances of two new systems are better than that of the reference system. The system flow of the new system 2 is more complex compared with that of the new system 1. Generally speaking, the performance of new system 1 is better than that of new system 2. On the basis of new system 1, new system 3 is further optimized and its full operating condition performance characteristics are analyzed. The standard coal consumption rate of new system 3 is reduced about 1 g/kWh at higher load, and around 0.2 g/kWh at low load.

Keywords: ultra-supercritical double reheat cycle; boiler-turbine coupling; exergy method; thermal performance analysis; system optimization

1. Introduction

At present, coal is still the main energy source in many countries all over the world. However, in the next five years, the contribution of coal to the global energy structure will decline from 27% to 25%. China's economy is in the period of structural transformation, and coal demand will gradually decline. The International Energy Agency (IEA) predicts that China's coal consumption will show a structural decline of less than 1% per year on average [1]. The annual development report of China's power industry in 2019 pointed out that the installed capacity of non-fossil energy power generation in China accounted for 40.8% of the total installed capacity of power generation in the country, which was 2.1 percentage points higher than the previous year. Power generation from non-fossil energy sources has accelerated the growth, and the proportion of coal-fired power generation to the total power generation capacity is decreasing gradually [2]. In order to further develop the coal-fired power generation, it is imperative to improve the power generation efficiency and reduce the coal consumption.

In order to further improve the efficiency of power generation, many energy saving and consumption reduction measures have been proposed. Increasing steam parameters not only reduces the heat transfer loss of the boiler, but also increases the generating capacity of the unit, which can directly and effectively improve the efficiency of the unit. Nowadays, the main steam parameters of ultra-supercritical units have reached up to

600 °C/31 MPa, and the ultra-supercritical technology with higher parameters is also in progress. Now the development goal is to increase the initial steam parameters of coal-fired power plants to 700 °C/35 MPa, and its thermal efficiency can reach more than 50% [3].

The waste heat utilization of flue gas can effectively reduce the flue gas energy loss and improve the efficiency of thermal power plants [4–8]. Min Yan et al. [4] proposed a novel boiler cold-end optimization system, in which both bypass flue gas waste heat after the wet flue gas desulfurizer and steam extraction heat were used to preheat the air, resulting more flue gas flowing into the bypass flue and high-pressure steam extraction reduced. Compared with the bypass flue waste heat recovery system, Yu Han et al. [5] added a steam-air heater between the air preheater and the pre-positioned air preheater, resulting in better energy-saving effects. Based on the data from an existing 1000 MW typical power generation unit in China, Xu et al. [6] analyzed four typical flue gas heat recovery schemes quantitatively from the thermodynamics perspective. Wang et al. [7] proposed that both the flue gas waste heat recovery exchanger bundled by the polytetra-fluoroethylene tube (FGC1) and the dewatering heat exchanger (FGC2) could recycle heat and water respectively. Four integration schemes of FGC1 and FGC2 with the regenerative system were proposed and analyzed. Multi-pressure condenser could effectively improve the thermal efficiency of the cycle. Xu et al. [8] applied the thermodynamic law to obtain the guiding principle for the optimum design of multi-back pressure condenser. The above schemes improved the overall system efficiency by reducing the heat loss to the environment.

Improving the fuel quality and utilizing other heat sources can effectively reduce the coal consumption in power plants. Xu et al. [9] improved the thermal efficiency of coal-fired power plants by removing the water content in some coal types. The results showed that coal's mass flow rate would increase by 0.6–1.5% as 0.1 kg moisture was removed per kg raw coal. The net efficiency of the power plant could increase in the range of 0.6–0.9%. Wang et al. [10] proposed a general system integration optimization method (GIOM) considering various possible integration schemes. The results showed that at the relatively higher heliostat field power (HFP) work conditions, the heat energy should be integrated into the highest pressure feedwater heater (FHs) as a priority. While at the relatively lower HFP and higher turbine power work conditions, the heat should be distributed into two highest pressure FHs in a certain proportion instead of fully distributed into the highest pressure FHs. The performances of both the receiver and the power block were improved.

According to the Second Law of Thermodynamics, reducing the temperature difference of heat transfer can effectively reduce the heat transfer exergy loss. Zhou et al. [11,12] proposed a steam cycle optimization process with 10 regenerative heaters, which further improved the thermal performance of the double reheat power generation plant. To solve the problem of too high superheat degree of the extraction steam in 1000 MW ultra-supercritical (USC) units, the energy-saving effects of two systems with one or two outside steam coolers (OSC) respectively were analyzed and compared with that of the small turbine regenerative system in the cases of 85% and 90% internal efficiency of the small steam turbine. Meanwhile, the energy-saving effects were analyzed for various utilization modes of the superheat degree under different loading conditions.

Breaking the boundary between the steam turbine system and the boiler system, further reducing the heat transfer temperature difference of the working medium, improving the heat transfer characteristics of different working mediums on both sides of the steam turbine system and the boiler system, can further reduce the heat transfer exergy loss. Yang et al. [13] put forward a heat integration system based on the boiler-turbine coupling method for large-scale coal-fired power plants. Low-pressure steam extraction at the turbine side is used to preheat the low temperature air, the absorbed heat of the air during the air preheating process is reduced, and the replaced part of high-temperature flue gas is used to heat the feed water and condensate water, and then, the heat transfer exergy losses are reduced obviously. Finally, the mass flow of the high-pressure steam extraction

is reduced and the output power of the overall power plant is increased. Fan et al. [14] put forward another scheme of boiler-turbine coupling. An additional economizer is used to replace the air preheater. The flue gas is used to heat part of the feed water, while the air is heated by the heat of the superheat of 9-stage extraction steams. The air preheating process reduces the superheat degrees of 9-stage extraction steams and the temperature differences of heat transfer in the thermal system, and finally reduces the coal consumption of the unit.

In order to further optimize the double reheat coal-fired power plant system, this paper takes a 1000 MW advanced ultra-supercritical double reheat coal-fired power generation unit as a reference system. Based on the principle of energy level matching and considering the heat transfer characteristics of steam, air and water, new schemes of boiler-turbine coupling are proposed. The thermodynamic model is established, and the energy consumption analysis is carried out to reveal the energy-saving potential and thermal performance of the new systems.

The new systems proposed in this paper have the following features:

(1) The low temperature feed water and the superheat heat of extraction steam are used to heat part of the air.

(2) The mass flow rate of the flue gas required for the air preheater is reduced, and the remaining high temperature flue gas replaces part of high temperature steam extraction to heat both the condensate water and feed water, so as to increase the output power of steam turbine.

(3) Compared with systems of other literatures, the superheat degrees of extraction steams and the temperature differences of the air heat transfer process are reduced together in the new system 1 and new system 2.

In addition, the performance of the optimized system (new system 3) is obtained based on the further optimization of the new system 1. The optimized system uses the feed water to preheat the air and then the heated air enters into the air preheater to be heated further, which will reduce the temperature differences of heat transfer in the thermal system.

2. Description of Reference Double Reheat Cycle and Heat Transfer Process Analysis

A reference USC double reheat power plant is selected for analysis in this paper. The elemental analysis of design coal of the power plant is shown in Table 1, the net calorific value of design coal is 23,440 kJ/kg. The reference unit flowchart is shown in Figure 1. The initial parameters of the power plant are 30.1 MPa/605 °C/613 °C/613 °C. Ten-stage regenerative heaters and two-stage outside steam coolers are adopted in the regenerative extraction steam system. The temperature of the feed water entering into the boiler is 314 °C. Exhaust flue gas temperature is 117 °C, and the boiler efficiency is 94.70%. The coal consumption rate of power generation is 257.38 g/kWh. The overall performance of the plant is relatively high, however, there is still room for further energy saving and energy consumption reduction.

Table 1. Elemental analysis of coal.

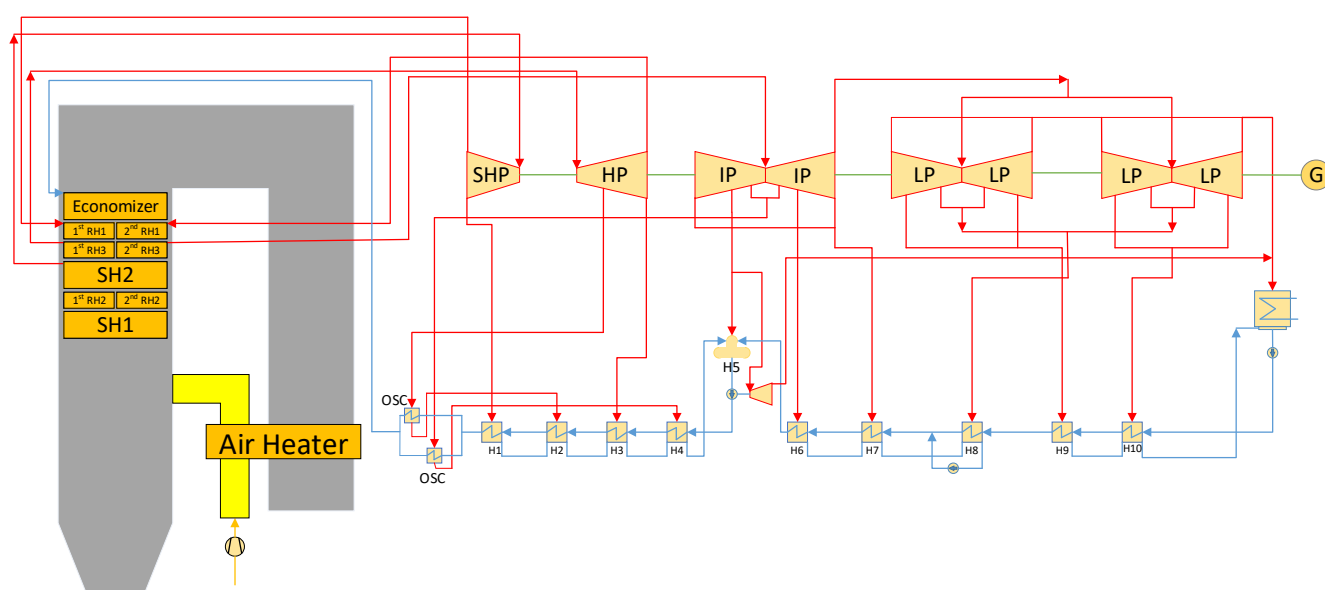|	Element	Mass Fraction/%
1	C_{ar}	61.7
2	H_{ar}	3.67
3	O_{ar}	8.56
4	N_{ar}	1.12
5	S_{ar}	0.6
6	A_{ar}	8.8
7	M_{ar}	15.55
8	V_{daf}	34.73

Figure 1. Flowchart of the reference double reheat coal-fired power generation system.

The heat transfer characteristics of both the air preheating system and the feeding water regenerative system are analyzed to reveal the potential of energy saving of the power plant.

In the air preheater, the mass flow rate of flue gas is 900 kg/s and the mass flow rate of air is 819 kg/s. The specific heat capacity of the flue gas is larger than that of air. As the heat transfer process proceeds, the heat transfer temperature difference between the flue gas and the air will increase. As shown in Figure 2, the high temperature terminal difference of the air preheater is 40 °C, but the low temperature terminal difference is around 87 °C, which makes the average heat transfer temperature differences of the whole air preheater reach up to 60 °C. The heat transfer exergy loss in the air preheater is large.
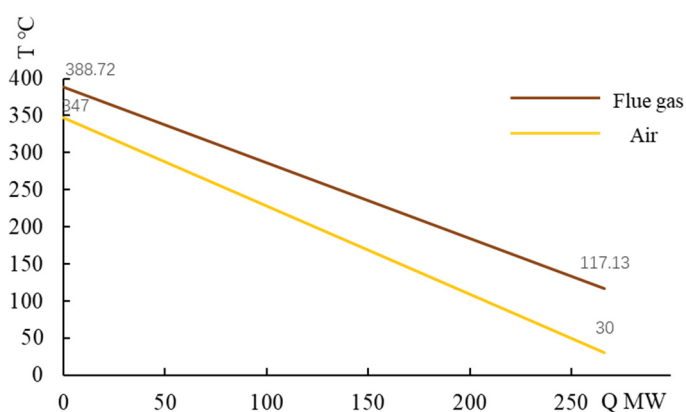

Figure 2. Heat transfer curve of air preheater.

As shown in Figure 1, 10-stage regenerative heaters and two-stage outside steam coolers are adopted to improve the temperature of the feed water entering into the boiler. Figure 3 shows the heat transfer process curve of the regenerative heating process. Applying two-stage outside steam coolers effectively reduces the superheat degrees of the second and fourth stage extraction steam. However, superheat degrees of other stages' extraction steam are still high at around 100–250 °C, which will inevitably result in the great exergy losses.
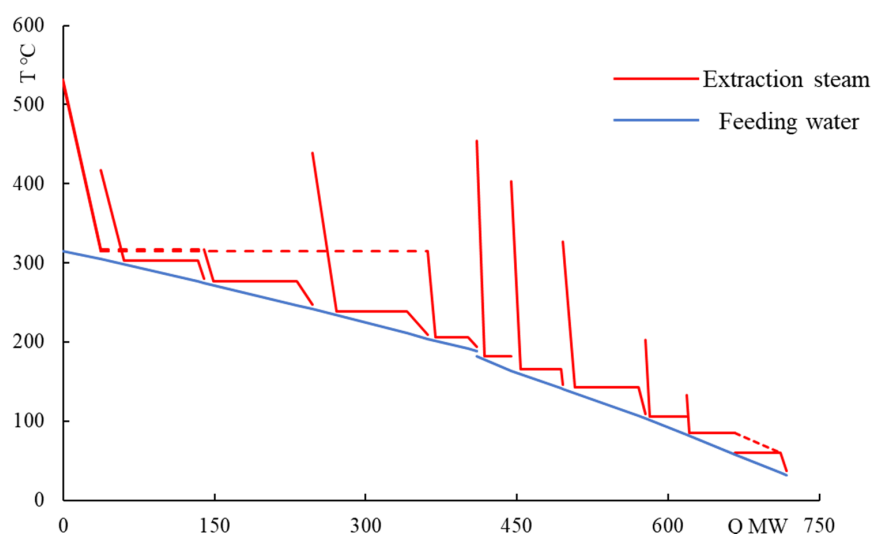
Figure 3. Heat transfer process curve of the regenerative heating process.

Generally, in the air preheater, the air temperature changes from 30 to 347 °C, and the feed water temperature changes from 30 to 314 °C. Their temperature change ranges are similar. The outlet temperature of the air is higher than that of the feed water, and the temperature difference between the air and the extraction steam is smaller. After heated by 9-, 10-stage steam extraction, the feed water temperature is about 80 °C. Comparing with the inlet air of the air preheater, the water heated by 9-, 10-stage steam extraction is more matched with the flue gas.

3. New Boiler-Turbine Coupling System

3.1. Optimizing Methods

Usually, the heat transfer processes are carried out independently in both steam turbine side and boiler system side in the conventional coal-fired power generation system (as shown in Figure 4), which results in great heat transfer exergy loss and low energy utilization efficiency. Yang et al. [13] proposed the boiler-turbine coupling system which broke the boundary of the unit and reduced the heat transfer exergy loss of the system. It reduced the temperature difference of the air heat transfer at the boiler side, however, it did not solve the problem of high superheat degree of steam extraction at the steam turbine side. Fan et al. [14] proposed another system that used 9-stage extraction superheat heat, not the flue gas heat, to heat the air, and used the flue gas heat to heat part of the feed water. The system effectively utilized the extraction steam superheat heat and reduced the heat transfer temperature difference of the air preheating process. However, it used part of the vaporization latent heat of the extraction steam to heat the air, which needed to change the internal structure of the regenerative heater. Due to a certain amount of steam extraction, the amount of heat to heat the air was constant. When the load changed, the excess air coefficient changed and the air temperature could not be adjusted. In order to solve this problem, different from the traditional heat exchange process between the boiler side and the turbine side, a new boiler-turbine coupling idea is proposed in this paper. As shown in Figure 5, the system uses both the low temperature feed water and the superheat heat of extraction steam to heat part of the air, and reduces the mass flow rate of the air passing through the air preheater. The mass flow rate of the flue gas required for the air preheater is reduced, and the remaining high temperature flue gas can be used to heat both the condensate water and the feed water, thus, the mass flow rate of the regenerative steam extraction can be reduced. Preheating the air with the low temperature feed water reduces the heat transfer temperature difference of the air heat transfer. Heating the air with the superheat heat of the extraction steam reduces the heat transfer exergy loss of the regenerative extraction steam system. The remaining high temperature flue gas replaces part of high temperature steam

extraction to heat both the condensate water and feed water, so as to increase the output power of steam turbine. By reducing the temperature difference of heat transfer in each heat transfer process, the cascade utilization of energy is realized, the exergy loss is reduced, and the power generation efficiency of the unit is improved.
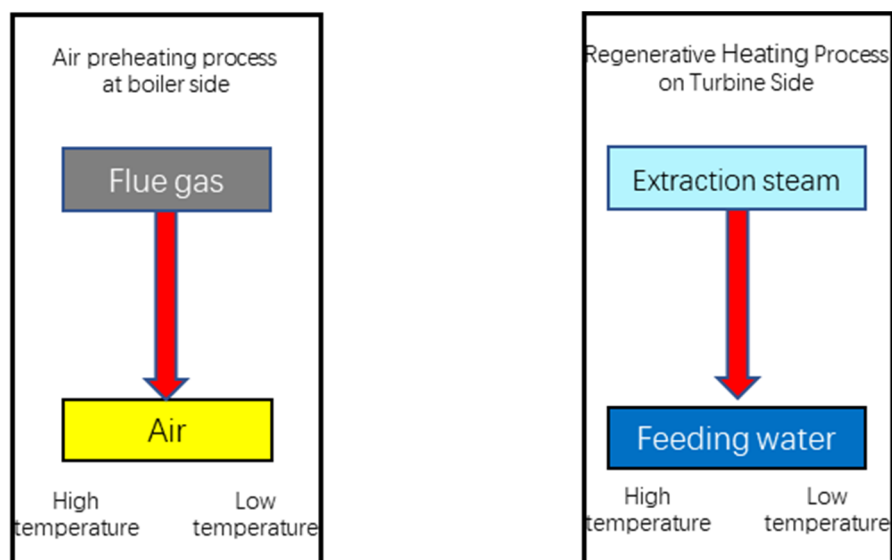
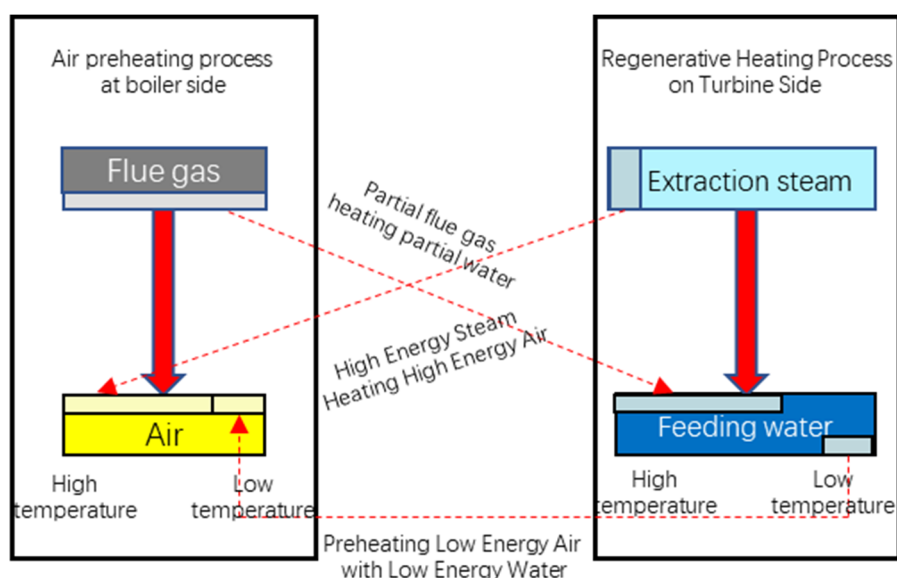**Figure 4.** Heat exchange process between traditional boiler side and steam turbine side.

Figure 5. Schematic diagram of a new boiler-turbine coupling.

3.2. New Boiler-Turbine Coupling System

Based on the integration idea of Section 3.1, two new boiler-turbine coupling systems are proposed in this paper. Figure 6 shows the flowchart of new system 1. As shown in Figure 6, two subsystems are integrated to new system 1 compared with the original system: the air heating subsystem for heating part of the air mainly is composed of one air-water heat exchanger (AH), and five gas-steam heat exchangers (AH1-AH5). This subsystem uses the steam superheat heat to heat the air. The flue gas bypass subsystem for heating the feed water with high temperature flue gas is composed of high and low temperature economizers (Eco1, Eco2). Figure 7 shows the flowchart of new system 2. New system 2 replaces two-stage outside steam coolers with the air heater on the basis of

new system 1. Since two-stage outside steam coolers have been cancelled, new system 2 needs to add a pre-economizer (Eco3) to ensure the inlet temperature of the feed water. New system 2 improves the volume flow rate of the air preheated by the extraction steam, however, heat transfer temperature differences of flue gas-steam heat exchangers at all levels are larger than those of new system 1.
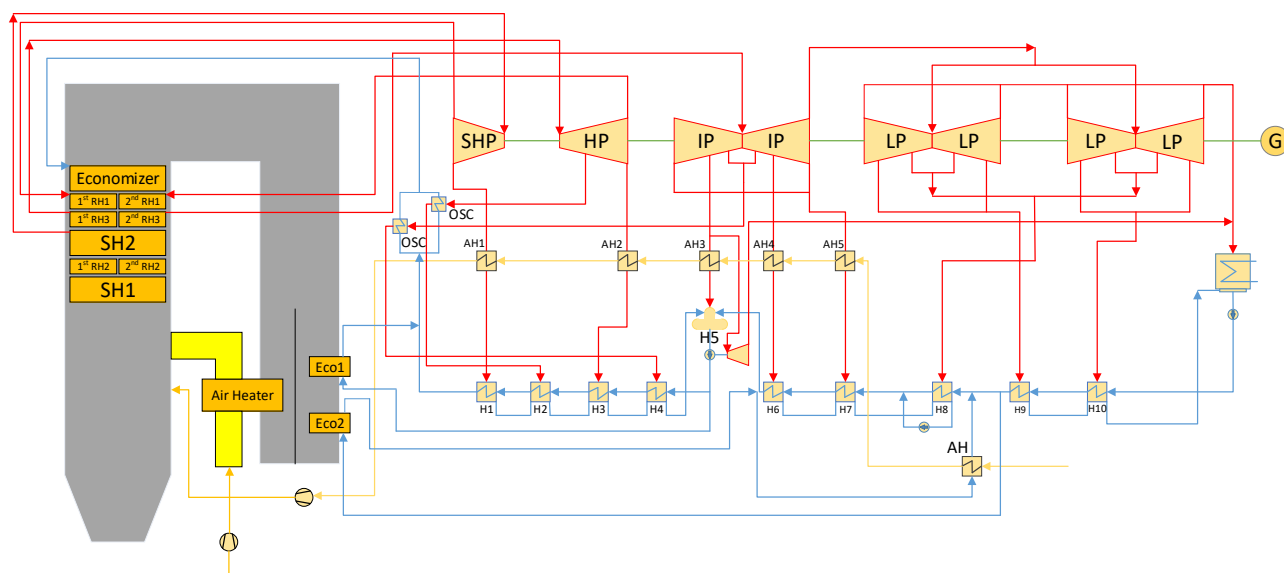
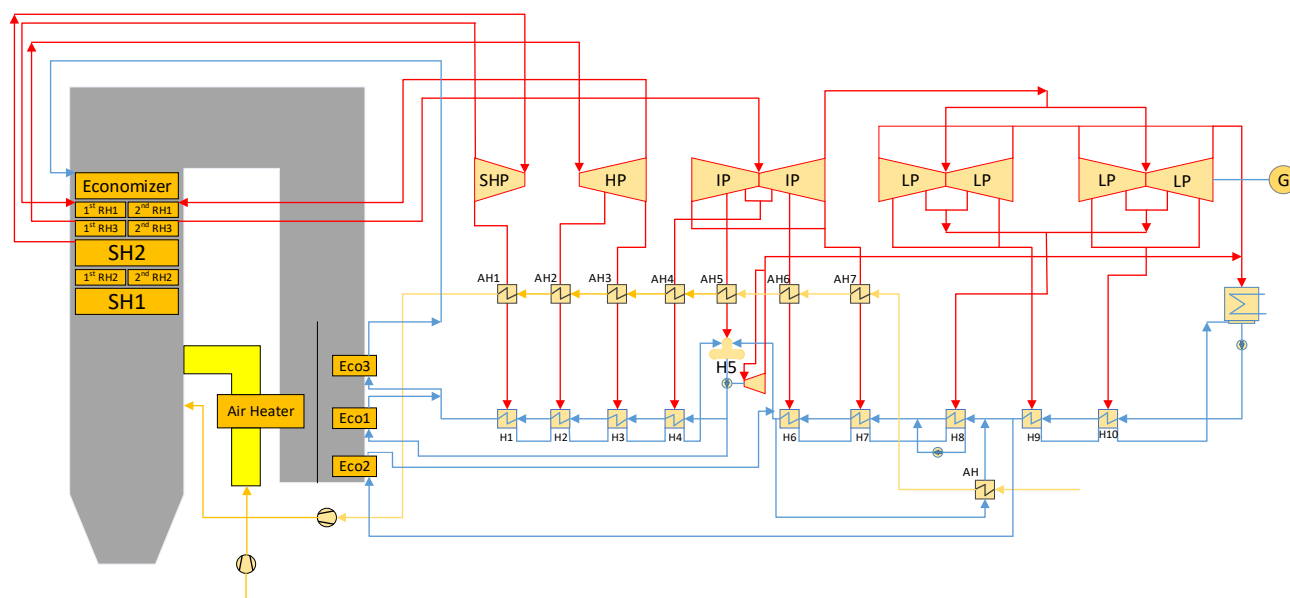**Figure 6.** Flowchart of new system 1.

Figure 7. Flowchart of new system 2.

Two new systems follow with the principle of energy cascade utilization through redistribution of heat. Before heating the feed water and the condensate water, the regenerative extraction steam heats the air because the air energy level is higher than the feed water energy level. Using the flue gas to heat part of the feed water and condensate water can reduce the mass flow of the regenerative steam extraction. Eventually, the regenerative extraction steam with high energy level is reduced, the unit power output is increased and the coal consumption of power generation is reduced.

4. Calculation and Analysis of Thermal Performance

4.1. Model Simulation Design and Assumptions

In order to evaluate the thermodynamic performance advantages of two novel systems, EBSILON Professional is used to build thermodynamic models. The model is based on the heat balance diagram of steam turbine and boiler manufacturer. The turbine heat acceptance (THA) load is selected as the design point of the system.

In the reference system model, the relevant definitions and assumptions for the model calculation are as follows:

(1)   The mass flow rate of live steam is 2533 t/h.
(2)   Regenerative steam extraction pressures and pipeline pressure drops at all levels are based on the thermal balance diagrams of the power plant.
(3)   The pressure drop coefficient of the 1st reheater and pipelines is 4.3%.
(4)   The pressure drop coefficient of the 2nd reheater and pipelines is 4.2%.
(5)   The absolute pressure drop of the boiler inlet feed water is 1.5 MPa.

For the adequate and comprehensive comparative analysis, the new system 1 model is improved as follows:

(1)   In order to compare the variations of extraction steam at all pressure levels, the main steam mass flow of new system 1 at the design point is kept the same as that of the reference system.
(2)   The initial parameters are 30.1 MPa/600 °C/610 °C.
(3)   The condenser pressure is 4.5 kPa.
(4)   The pressure drop of each air heater at the air side is 0.2 kPa.
(5)   The pressure drop of each air heater drop at the steam side is 2 kPa.
(6)   The heat loss of the air heater is ignored.
(7)   Air outlet temperature of AH is 150 °C.

4.2. Analysis of Heat Transfer Performance of Tail Flue Gas in New Boiler-Turbine Coupling System

Due to the reduction of high-pressure steam extraction, the mass flow rates of both the 1st reheat steam and 2nd reheat steam in two new systems increase. The increase of heat absorbed by the reheat steam in the boiler results in a lower flue gas inlet temperature of the air preheater than that of the reference system. As shown in Figure 8, the heat transfer temperature difference of the air preheater in new system 1 is 43 °C. In the flue gas bypass subsystem, the heat transfer temperature difference of the Eco1 is 30 °C, and that of Eco2 is 36 °C. As shown in Figure 9, the heat transfer temperature difference of the air preheater in new system 2 is 43 °C, the heat transfer temperature difference of Eco1 is 11 °C, that of Eco2 is 36 °C, and that of Eco3 is 26 °C. The heat transfer temperature differences of the tail flue gas in two new systems are much smaller than that of the air preheater in the reference system, which reduces the heat transfer exergy loss of flue gas and improves the thermal efficiency.
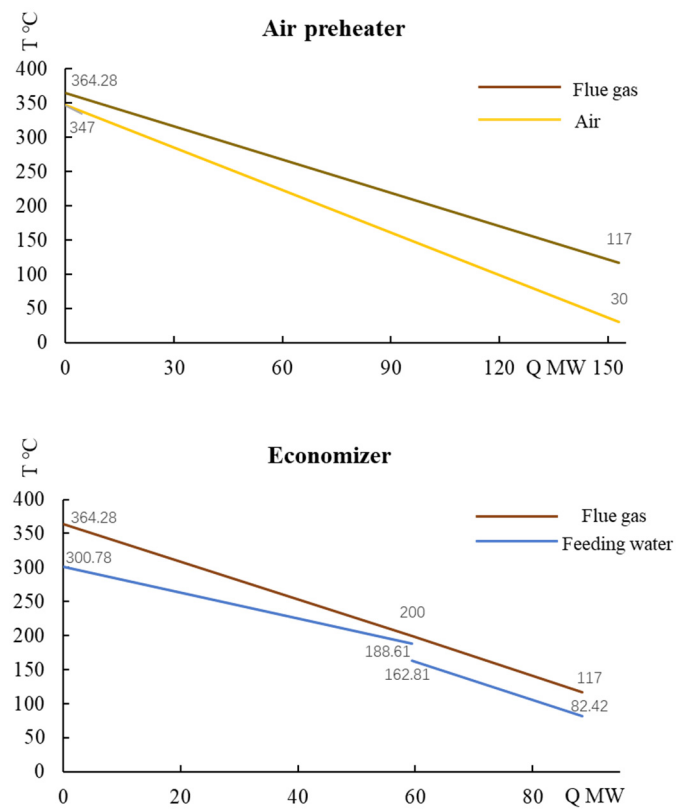
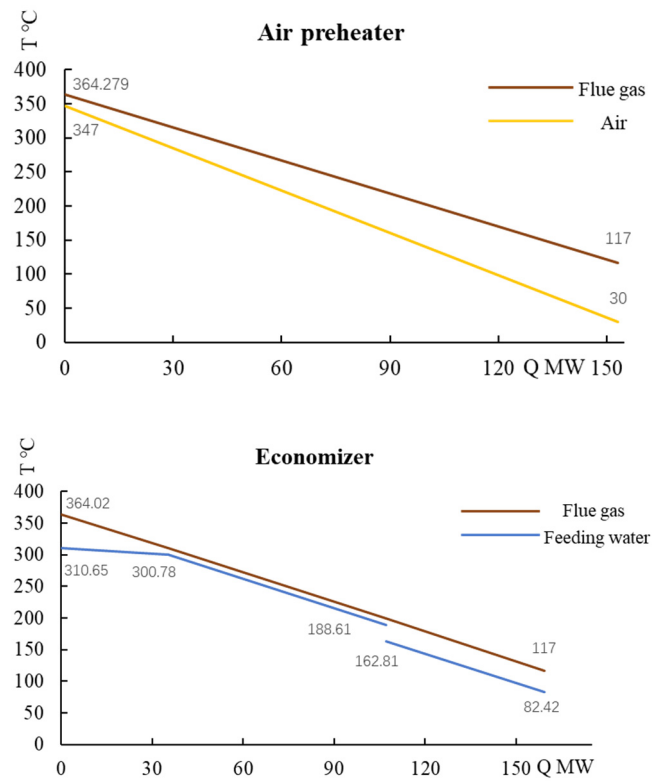**Figure 8.** Heat transfer analysis of the tail flue in new system 1.

Figure 9. Heat transfer analysis of the tail flue in new system 2.

4.3. Exergy Utilization Diagram of the Air Preheating Subsystem

In order to comprehensively analyze the energy utilization law of the air preheating subsystem and find out key influence factors of exergy loss in the process of energy transfer, the exergy utilization diagram (EUD) is adopted. The exergy loss reduced by the energy cascade utilization when the air is heated by the regenerative steam extraction is analyzed in depth, and its heat transfer characteristics are analyzed.

Different from the traditional exergy analysis method, the exergy utilization diagram can reveal the relationship between the energy change and energy grade change in thermal process in the form of diagram [15–17]. In EUD, ΔH is the energy change of the heat transfer process, which embodies the first law of thermodynamics, and A represents the energy level, which embodies the second law of thermodynamics. The area between curves of the exothermic side and the endothermic side is the exergy loss of the heat transfer process. Therefore, the image analysis can visually and clearly show the exergy loss in the process of energy transfer.

Energy-level "A" is a dimensionless quantity, which is the ratio of exergy change $\Delta \varepsilon$ to energy change ΔH in thermal process [17]. It is defined as follows:

$$A = \frac{\Delta \varepsilon}{\Delta H} = 1 - T_0 \frac{\Delta S}{\Delta H} \tag{1}$$

where, A is the energy level; ΔS is the entropy change, J/(kg·K); $\Delta \varepsilon$ is the exergy change, MW; ΔH is the energy change, MW; T_0 is the environment temperature, K.

In an ideal heat transfer process, A can be defined as follows:

$$A = 1 - \frac{T_0}{T} \tag{2}$$

where, T is the temperature of energy release side (heat source) or energy absorption side (cold source), K.

When analyzing the heat transfer process of air preheating subsystem, it is simplified as an ideal heat transfer process. Figure 10 shows the EUD of the heat transfer processes of seven-stage regenerative extraction steam in the reference system and new systems.

It can be seen that the energy level of regenerative extraction steam in the reference system is high, while the energy level of the feed water is low. The energy level difference between them is large, resulting in a great exergy loss in the heat transfer process. In new systems, the air temperature is higher than the feed water temperature, and the energy level of the air is higher than that of the feed water. The air heating subsystem can effectively reduce the heat transfer exergy loss. New system 2 has two more stages of steam-flue gas heat exchangers than new system 1, but the reduced exergy loss in each stage is less than that of new system 1.
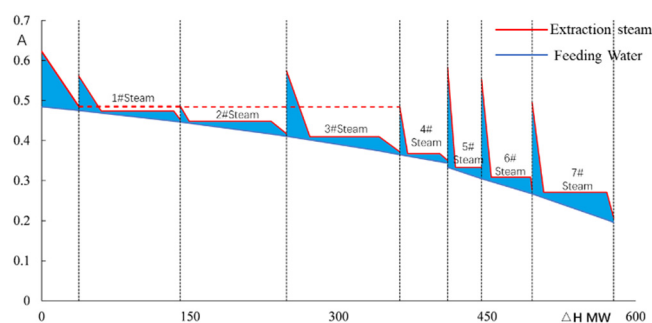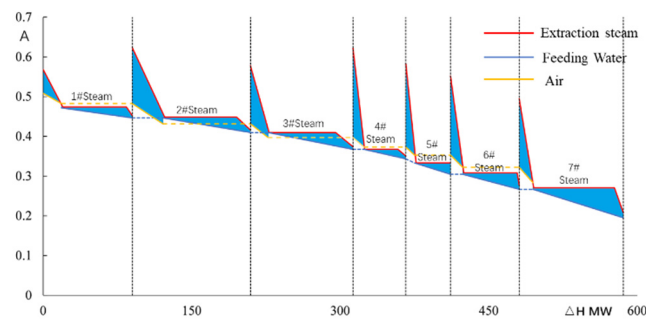

(**a**) Reference system

Figure 10. *Cont.*
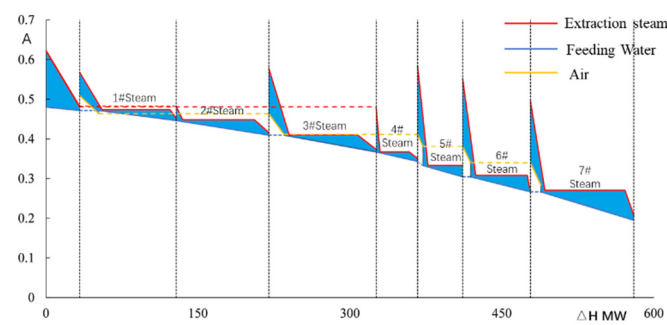
(**b**) New system 1

(**c**) New system 2

Figure 10. EUD diagrams of seven-stage regenerative steam extraction heat transfer processes of original system and new systems.

4.4. Analysis of Steam Extraction Mass Flow and Power Generation Capacity in Regenerative System

Heating the air with the superheat heat of extraction steam will increase the required mass flow of extraction steam at all pressure levels, while heating the feed water with the flue gas will reduce the mass flow of the regenerative steam extraction. Figure 11a shows the power generation capacity of each stage extraction steam. As can be seen from Figure 11b,c, in two new systems, the mass flow rates of 1–4 stages of extraction steam reduce. On the contrary, the mass flow rates of 5–8 stages of extraction steam increase. The mass flow rates of 9–10 stages are basically unchanged. Generally speaking, the total mass flow rate of steam extraction of new system 1 increases by 2.86 kg/s and that of new system 2 increases by 3.11 kg/s.
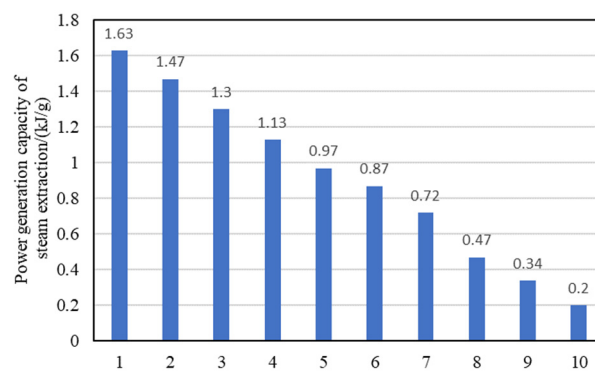
(**a**) Power generation capacity of different regenerative extraction steam.

Figure 11. *Cont.*
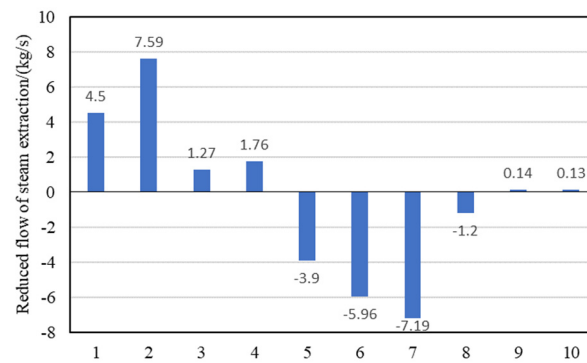
(**b**) Reduced mass flow rates of extraction steam at different stages in new system 1.
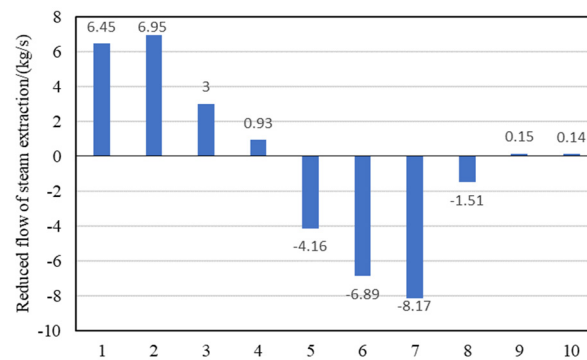
(**c**) Reduced mass flow rates of extraction steam at different stages in new system 2.
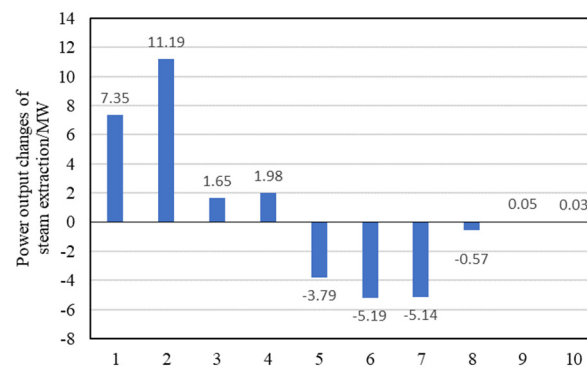
(**d**) Power output changes of steam extraction in new system 1.
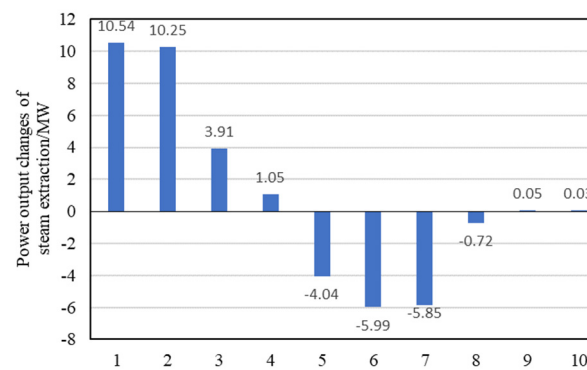
(**e**) Power output changes of steam extraction in new system 2.

Figure 11. Steam extraction mass flow and power generation capacity.

As shown in Figure 11d,e, there is a great difference in power output of different stages of steam extraction. Power generation capacity of high-pressure extraction steam is higher than that of low-pressure extraction steam. Reducing the mass flow rate of high-pressure extraction steam and increasing the mass flow rate of low-pressure extraction steam can increase the power output of the overall power plant. For the new system 1, the mass flow rate reduction of 1–4 stages of extraction steam makes the power output increase by 22.17 MW, while the mass flow rate increase of 5–8 stages of extraction steam results in the power output decrease of around 14.69 MW. Finally, the net power output of new system 1 increases by 7.56 MW. For the new system 2, the mass flow rate reduction of 1–4 stages of extraction steam makes the power output increase by 25.75 MW, while the mass flow rate increase of 5–8 stages of extraction steam results in the power output decrease of around 15.88 MW. Finally, the net power output of new system 1 increases by 9.23 MW.

4.5. Analysis of Steam Extraction Mass Flow and Power Generation Capacity in Regenerative System

Table 2 shows main performance data and calculation results of three systems.

Table 2. Main performance data and calculation results of three systems.

			Reference System	New System 1	New System 2
Main steam	Mass Flow	t/h	2533	2533	2533
	Press.	bar	301	301	301
	Temp.	°C	600	600	600
Reheat steam	Mass Flow	t/h	2342	2358	2365
	Press.	bar	10.1	10.1	10.1
	Temp.	°C	610	610	610
Double reheat steam	Mass Flow	t/h	2014	2057	2068
	Press.	bar	30.8	30.8	30.8
	Temp.	°C	610	610	610
Coal flow rate		t/h	319	319	319
Air mass flow		t/h	2948	2948	2948
Air heated by the steam extraction		t/h	-	1252	2035
Total power output		MW	992.56	999.92	1001.57
Power consumption of fans		MW	-	0.98	2.08
Net output power		MW	992.56	998.94	999.49
Standard coal consumption rate		g/kWh	257.38	255.73	255.59

The standard coal consumption rate (b_s) is calculated as follows:

$$b_s = bQ_{net,ar}/Q^s_{net,ar} \tag{3}$$

where, b is the coal consumption rate, g/kWh; $Q_{net,ar}$ refers to the net calorific value of design coal, kJ/kg; $Q^s_{net,ar}$ refers to the net calorific value of standard coal, kJ/kg.

$$b = 1000B/P_e \tag{4}$$

where, B is coal mass flow rate, t/h; P_e denotes to the net output power, MW.

Compared with the reference system, the total power output of new system 1 increases by 7.36 MW, and that of new system 2 increases by 9.15 MW. However, due to the pressure loss of air passing through multi-stage heaters in air preheating subsystem of two new systems, it is necessary to increase fans to overcome the air flow resistance. The power consumption of fans in new system 1 is about 0.98 MW, and that of new system 2 is about 2.08 MW. In summary, the net output power of new system 1 increases by 6.38 MW and the standard coal consumption decreases by 1.65 g/kWh. The net output power of new system 2 increases by 6.93 MW and the standard coal consumption rate decreases by 1.79 g/kWh. Replacing two-stage outside steam coolers with the air heater decreases the standard coal consumption under the design condition.

5. Off-Design Performance Analysis

In order to reveal the actual operation performances of two new systems. The off-design performances of two systems are deeply investigated. When analyzing the off-design performance, the model is set in the form of a fixed power output. Table 3 shows the data of three systems under off-design conditions.

Table 3. Data of three systems under off-design conditions.

Reference System		THA	75% THA	50% THA	40% THA	30% THA
Main steam flow	t/h	2533	1866.23	1211.74	971.68	738.1
Air flow	t/h	2947.82	2217.91	1608.96	1478.44	1197.33
Flue gas flow	t/h	3233.58	2436.85	1759.88	1602.73	1294.4
Exhaust flue gas temp.	°C	117.13	109.24	98.85	94.62	89.18
Hot air temp.	°C	347	334.78	311.23	304.05	288.13
Total power	MW	986.79	727.14	481.07	384.52	283.85
Coal flow	t/h	318.49	240.01	165.48	136.29	106.43
Standard coal consumption rate	g/kWh	258.47	264.33	275.47	283.85	300.27
New System 1		**THA**	**75% THA**	**50% THA**	**40% THA**	**30% THA**
Main steam flow	t/h	2516.39	1833.81	1189.19	959.68	731.11
Air flow	t/h	2927.67	2320	1718.39	1501.91	1189.88
Ratio of air heated by steam	%	42.63	42.65	37.71	37.29	38.66
Flue gas flow	t/h	3216.61	2537.57	1868.42	1625.44	1286.34
Bypass flue gas ratio	%	36.81	37.68	33.62	31.69	35.87
Exhaust flue gas temp.	°C	116.79	107.17	93.88	88.12	81.92
Hot air temp.	°C	346.95	333.19	305.37	294.29	276.13
Total power output	MW	986.78	727.14	481.07	384.52	283.85
Fans power consumption	MW	0.97	0.48	0.13	0.08	0.05
Coal flow	t/h	316.82	238.61	164.5	135.44	105.77
Standard coal consumption rate	g/kWh	257.37	262.96	273.92	282.14	298.45
Coal consumption reduction	g/kWh	1.1	1.37	1.55	1.71	1.82

Table 3. *Cont.*

New System 2		THA	75% THA	50% THA	40% THA	30% THA
Main steam flow	t/h	2510.41	1832.23	1190.27	960.47	726.96
Air flow	t/h	2924.15	2316.62	1718.79	1499.7	1190.28
ratio of air heated by steam	%	67.23	63.91	55.76	51.45	48.97
Flue gas flow	t/h	3212.74	2533.92	1868.85	1623.04	1286.78
Bypass flue gas ratio	%	65.78	63.1	57.12	52.09	55.35
Exhaust flue gas temp.	°C	116.86	106.12	90.44	85.33	81.29
Hot air temp.	°C	346.65	332.15	303.177	292.75	268.23
Total power output	MW	986.79	727.14	481.06	384.52	283.85
Fans power consumption	MW	2.03	0.87	0.25	0.12	0.07
Coal flow	t/h	316.44	238.26	164.54	135.24	105.81
Standard coal consumption rate	g/kWh	257.33	262.72	274.05	281.76	298.59
Coal consumption reduction	g/kWh	1.14	1.61	1.42	2.09	1.68

It can be seen that the proportion of air heated by the extraction steam varies under different load conditions in two new systems, and the mass flow of the flue gas displaced by the extraction steam will also change. Therefore, the boiler bypass flue gas mass flow needs to be adjusted with the change of working conditions. The boiler flue gas exhaust temperatures of new systems are low under the low load. The possible low temperature corrosion problem of tail heaters should be considered and the excessive low load operation should be avoided as far as possible.

Figure 12 shows the changes of the standard coal consumption rates under different working conditions of new systems and reference system. The results show that the performances of two new systems are superior to that of the reference system under different load conditions, and the standard coal consumption rates of new systems under different loads are 1.1–2.0 g/kWh lower than that of the reference system.
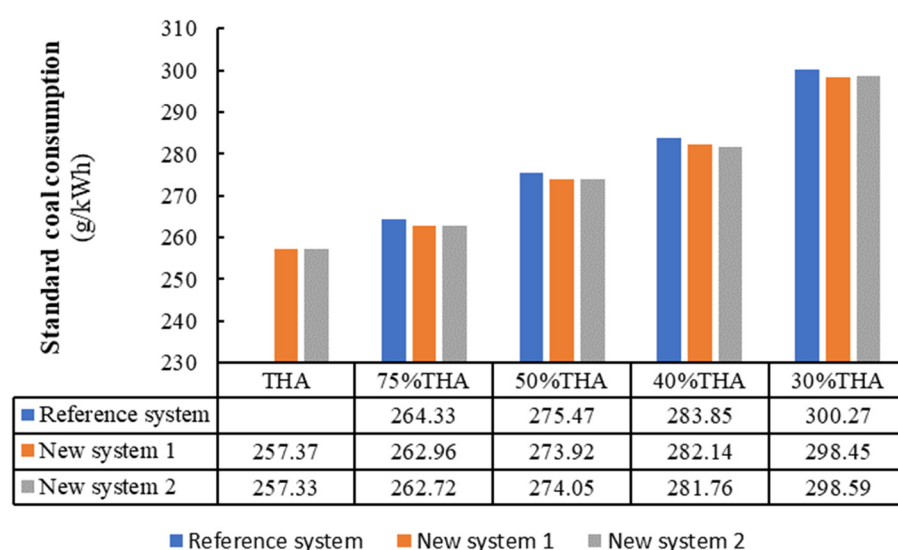

	THA	75%THA	50%THA	40%THA	30%THA
Reference system		264.33	275.47	283.85	300.27
New system 1	257.37	262.96	273.92	282.14	298.45
New system 2	257.33	262.72	274.05	281.76	298.59

■ Reference system ■ New system 1 ■ New system 2

Figure 12. Coal consumption rates of new systems and reference system.

Generally speaking, the reduced coal consumption rates of two new systems are approximate. However, the flowchart of new system 2 is more complex due to the substi-

tution of two-stage outside steam coolers with the air heater and the adding of one-stage pre-economizer. A large amount of air is heated by the extraction steam in new system 2. When the load changes, the wet steam is easily produced at the outlet of steam-air heat exchanger, which will affect the safety of unit. From this point, new system 1 is superior to new system 2.

6. Re-Optimization of New System 1

6.1. Energy-Saving Potential Analysis of New System 1

New system 1 uses the superheat heat of the extraction steam to heat part of air, which reduces the heat transfer exergy loss of extraction steam and part of air. As shown in Figure 13, in the air preheater of new system 1, the air inlet temperature is 30 °C, and the flue gas exhaust temperature is 117 °C. A large heat transfer temperature difference between them still results in a large heat transfer exergy loss of the air preheater. Air preheater in new system 1 still has the further energy-saving potential.
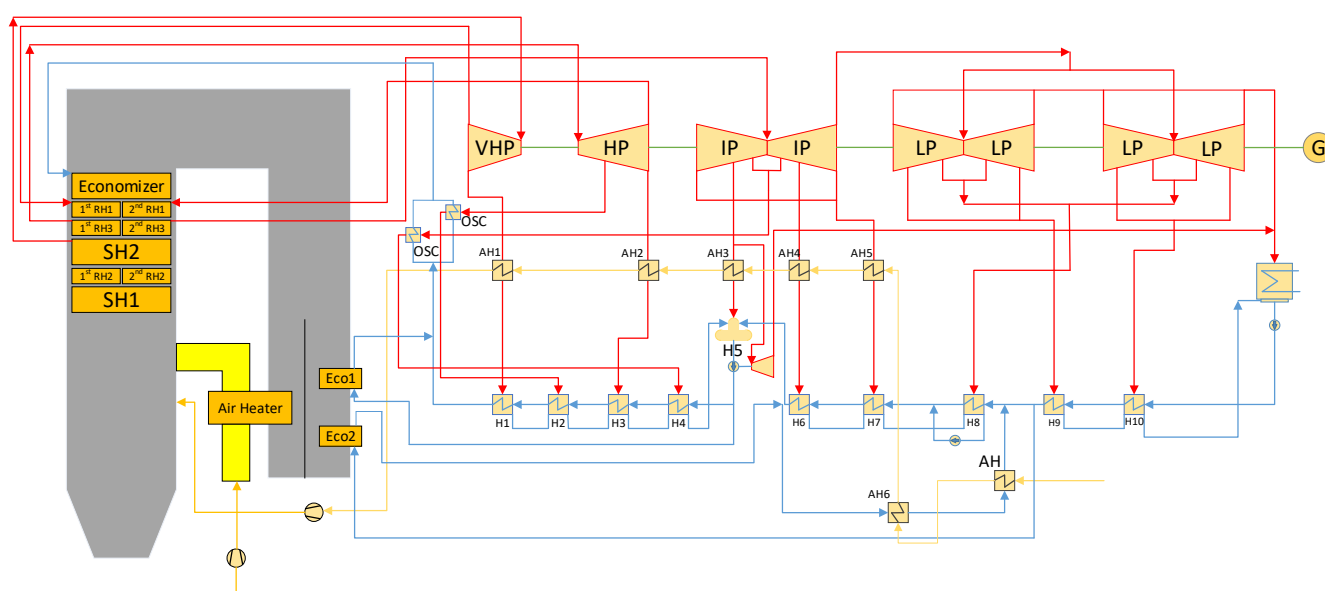

Figure 13. Flowchart of new system 3.

Therefore, new system 3 is proposed by optimizing the air preheater subsystem of new system 1. New system 3 uses the feed water to preheat the air and then the heated air enters into the air preheater to be heated further, which will reduce the heat transfer exergy loss of the air preheater. The flowchart of new system 3 is shown in Figure 13. After the air is heated to 100 °C by the air-feed water heat exchanger (AH), part of air enters into the air preheating subsystem and the remaining air enters into the air preheater, and finally, they jointly enter into the furnace.

6.2. Analysis of Optimization Results

In order to analyze optimization results, the thermodynamic model of new system 3 is also established by using the EBSILON Professional

Parameters of equipment added in the new system 3 are set as follows:

(1)   The outlet air temperature of AH is 100 °C.
(2)   Pressure drop values on both the water side and the air side of AH are 0.2 kPa.
(3)   The outlet air temperature of AH8 is 150 °C.

The heat transfer process curve of the air preheater in new system 3 is shown in Figure 14. Inlet and outlet temperatures of air and flue gas are very close. As part of the feed water is used to heat the air, the absorbed heat by air in the air preheater decreases and the mass flow rate of the bypass flue gas increases. With the mass flow increase of

the feeding water heated by the flue gas heater, the high-pressure steam extraction is reduced, resulting in the increase of the 1st reheat steam and 2nd reheat steam flow, and the increase of the absorbed heat by the steam. The temperature of flue gas entering into the air preheater is further reduced to 357.74 °C. The inlet temperature of air entering the air preheater rises from 30 °C to 100 °C. The heat transfer temperature difference of the air preheater decreases significantly and the heat transfer exergy loss decreases.
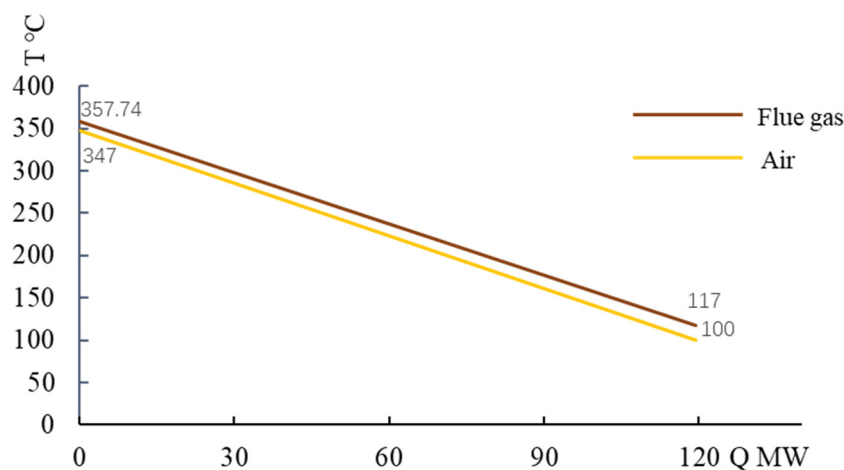

Figure 14. Heat transfer curve of air preheater.

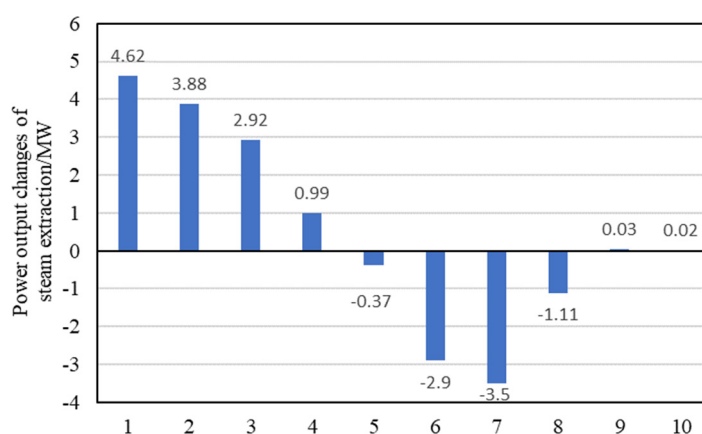### 6.3. Variation of Steam Extraction at Different Levels in New System 3

Due to the addition of an air-water heat exchanger (AH) in new system 3, the heat transfer amount between the air and the feed water increases, which results in the increase of steam extraction mass flow rates of 6–8 stage low-pressure heaters. At the same time, as the absorbed heat by air in the air preheater decreases, the mass flow of the bypass flue gas increases, and the steam extraction flow of high temperature economizer decreases.

As shown in Figure 15, compared with new system 1, the total mass flow rate of 1–4 stage extraction steam in new system 3 decreases by 8.58 kg/s and the power output increases by 12.41 MW. The total mass flow rate of 5–8 stages of extraction steam increases by 10.95 kg/s and the power output decreases by 7.88 MW. The mass flow rate of the low-pressure turbine exhaust steam increases by 2.167 kg/s, and the total unit power output increases by 4.58 MW.
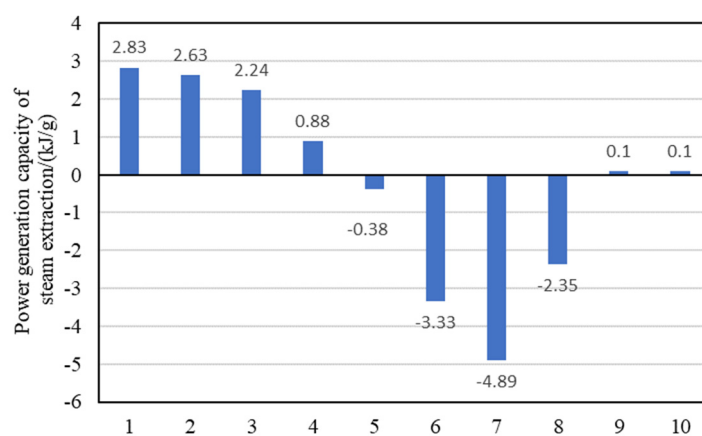
(**a**) Reduced flow rates of extraction steam at different stages.

Figure 15. *Cont.*

(**b**) Power output changes of extraction steam at different stages.

Figure 15. Variation of steam extraction at different levels.

6.4. Performance Analysis of New System 3

The performance parameters of new system 1 and new system 3 are compared as shown in Table 4.

Table 4. Comparison of main performance parameters of new system 1 and new system 3.

			New System 1	New System 3
Main steam	Mass Flow	t/h	2533	2533
	Press.	bar	301	301
	Temp.	°C	600	600
1st reheat steam	Mass Flow	t/h	2358	2368.57
	Press.	bar	10.1	10.1
	Temp.	°C	610	610
2nd reheat steam	Mass Flow	t/h	2057	2085.02
	Press.	bar	30.8	30.8
	Temp.	°C	610	610
Coal flow		t/h	319	319
Air flow		t/h	2948	2947.82
Flow of air heated by steam		t/h	1252	1259.67
Total power output		MW	999.92	1004.45
Power consumption of fans		MW	0.98	0.61
Net output power		MW	998.94	1003.84
Standard coal consumption rate		g/kWh	255.73	254.48

After the further optimization of new system 1, the heat transfer temperature difference of the air preheater is further reduced, and the proportion of bypass flue gas increases from 37% to 49%. The mass flow rates of 1–3 stage high-pressure extraction steam decrease, the 1st reheat steam flow rate increases by 2.83 kg/s and the 2nd reheat steam increases by 7.7 kg/s. The net power output increases by 5.5 MW and the standard coal consumption rate decreases by 1.25 g/kWh.

6.5. Off-Design Performance Analysis

A constant power model is set up to calculate the off-design performance of new system 3. Table 5 shows the performance data of new system 3 under variable operating conditions. As shown in Table 5, the bypass flue gas proportion of new system 3 is higher than that of new system 1 under all working conditions; the ratio of air heated by steam increases, and the power consumption of the fan increases. Flue gas exhaust temperatures of both new system 1 and new system 3 are similar.

Figure 16 shows the changes of the standard coal consumptions in both new systems 1 and 3 under off-design conditions. It can be seen that after the further optimization of new system 1, the coal consumption rate decreases by 1 g/kWh under both THA and 75% THA conditions. Under low load conditions, the reduction amount of coal consumption rate decreases. Especially under 30% THA condition, only 0.2 g/kWh is saved. New system 3 performs poorly at low load, which needs to be noticed and improved.

Table 5. Analysis of main parameters under variable parameters.

| New System 3		THA	75% THA	50% THA	40% THA	30% THA
Main steam flow	t/h	2507.06	1823.23	1182.16	953.66	725.37
Air flow	t/h	2918.34	2311.66	1715.18	1499.91	1189.28
Ratio of air heated by steam	%	42.93	44.38	49.55	52.97	48.77
Flue gas flow	t/h	3206.36	2528.49	1864.92	1623.27	1285.7
Bypass flue gas ratio	%	49.43	52.17	57.517	60.59	59.89
Exhaust flue gas temp.	°C	116.66	107.38	96.19	90.74	83.83
Hot air temp.	°C	346.61	328.98	297.83	285.73	262.45
Total power output	MW	990.96	731.1	483.95	387.96	287.41
Fans' power consumption	MW	0.6	0.33	0.18	0.15	0.05
Coal flow	t/h	315.81	237.75	164.19	135.26	105.72
Standard coal consumption rate	g/kWh	256.29	261.84	273.33	281.7	298.26
Coal consumption rate reduction	g/kWh	1.08	1.12	0.59	0.44	0.19
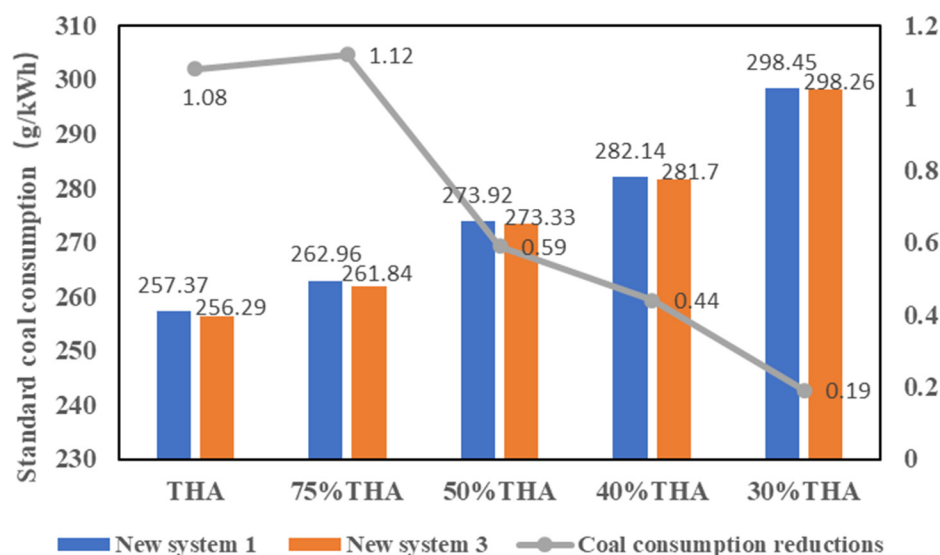
Figure 16. The standard coal consumption comparison of three systems under variable conditions.

7. Conclusions

This paper takes a 1000 MW ultra-supercritical double reheat coal-fired power generation system as a reference system. Based on the comprehensive consideration of the utilization of flue gas waste heat and the extraction steam superheat heat, different new heat integration systems based on the boiler-turbine coupling idea are proposed. The main features of new systems are as follows:

(1) The low temperature feed water and the superheat heat of extraction steam are used to heat part of the air in new system 1 and new system 2. The superheat degrees of

extraction steams and the temperature differences of the air heat transfer process are reduced together.

(2) New system 2 replaces two-stage outside steam coolers with the air heater on the basis of new system 1, and thermal performances of two systems are deeply compared.

(3) New system 3 uses the feed water to preheat the air on the basis of new system 1, and the heat transfer temperature difference of air preheater is further reduced.

Thermodynamic models are established and their thermodynamic performances under different operation conditions are investigated and compared. The results are as follows:

(1) The new systems follow with the energy level matching principle. Through the boiler-turbine coupling, the exergy losses of heat transfer process are obviously reduced, and the thermal efficiency of the overall system are improved. The results show that the power output of new system 1 increases by 6.38 MW and the standard coal consumption rate decreases by 1.65 g/kWh compared with the reference system. The power output of new system 2 increases by 6.93 MW and the standard coal consumption rate decreases by 1.79 g/kWh compared with the reference system.

(2) The performances of new system 1 and new system 2 are better than that of reference system under all working conditions. When the load decreases, the coal consumption rate of two new systems increases more slowly, and the inlet air temperature of boiler can be maintained to ensure the safe operation of the boiler. However, the exhaust temperatures of flue gas in two new systems are slightly lower than that of original system. When the load is too low, the exhaust flue gas temperature will be too low, and the tail heaters may be corroded at lower temperature. Therefore, two new systems should keep the load rate above 50% THA as far as possible.

(3) The standard coal consumption rates of new systems 1 and 2 are similar. However, new system 2 is more complex than new system 1. In addition, the wet steam may appear at the outlet of steam-air heat exchanger of new system 2 under variable operating conditions, which affects the safety of the system. Considered comprehensively, new system 1 is superior to new system 2.

(4) New system 3 further reduces the heat transfer temperature difference of air preheater, and improves the power generation efficiency of the overall system. The standard coal consumption rate is further reduced by 1 g/kWh under high load compared with new system 1. The energy-saving effect is obvious. However, the reduced coal consumption rate decreases at low load conditions. Especially under the 30% THA condition, the reduced standard coal consumption rate is only 0.2 g/kWh compared with new system 1.

In this paper, the waste heat utilization of flue gas is not considered. The waste heat of flue gas could be considered in future work. Additionally, the low reduced coal consumption rate at low load conditions in new system 3 is worth further studying. Moreover, the comprehensive economic performances of new systems should be further studied in the future.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclatures

Abbreviation

AH	Air heater
H	Regenerative heater
IEA	International energy agency
IP	Intermediate-pressure cylinder
LP	Low-pressure cylinder
OSC	Outside steam cooler
RH	Reheater
SH	Superheater
SHP	Super high-pressure cylinder
THA	Turbine heat acceptance condition
USC	Ultra-supercritical
HP	High-pressure cylinder
HRSG	Heat recovery steam generator
HT	High-pressure turbine

Symbols

b	Coal consumption, g/kWh
b_x	Standard coal consumption rate, g/kWh
A	Energy level
B	Coal flow rate, t/h
P_e	Net output power, MW
$Q_{net,ar}$	Net calorific value of design coal, kJ/kg
$Q^s{}_{net,ar}$	Net calorific value of standard coal, kJ/kg
T_0	Environment temperature, K
ΔS	Entropy change, J/(kg·K)
$\Delta \varepsilon$	Exergy change, MW
ΔH	Energy change, MW

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