A Novel 1000MW Double Reheat Ultra-Supercritical System with Turbine–Extraction-Heated Air Preheaters and Low Temperature Economizers

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**Abstract**

Herein, an optimization system of a 1000-MW double-reheat ultra-supercritical (USC) unit with turbine–extraction-heated air preheaters (EAPHs), which use turbine extractions as heat sources of air preheaters while economizer outlet flue gas’ heat absorbed by low-pressure economizers (LPEs), is designed based on a 1000-MW double-reheat USC unit (reference system). The energetic performance analyses of the USC system with EAPH and reference system are used to compare their major parameters. In addition, thermodynamic analyses under partial load operation conditions are presented. The results show that the proposed system reduced the exergy loss in the air-heating and flue gas-cooling processes. Furthermore, the optimization system increases the temperature of secondary air and reduces the overall superheat degree of several extractions. Theoretically, the proposed system reduces the standard coal equivalent (SCE) consumption by nearly 5.5g/kWh under THA load when the temperature of the flue gas entering the electrostatic precipitator is set to 95°C. Moreover, the proposed system can still reduce SCE consumption by 2.9 g/kWh under the condition of 50% THA load. Our findings indicate that system with APOB could improves the performance of the unit and may provide a theoretical basis for the optimization of double-reheat USC units.

**Key words**: Double reheat, ultra-supercritical power plant, Exergy analysis, System Optimization, Thermodynamic analyses

1. **Introduction**

Coal is still the main fossil fuel resources for electricity production in the world according to [1]. In China, coal –fired power generation accounts for more than70% of the total electricity generation and subsequently contribute almost 50%, 37%, 33% and 55% to the SOX, NOX, dust and CO2 emission volumes respectively [2-4]. Statics show that China has been the largest producer and consumer of energy all over the world since 2013 [5]. For 2016 as a whole, Chinese coal production fell by 7.9% and the price of steam coal increased by over 60%. Improve system efficiency and reducing coal consumption is still the main task of power plant design. Nowadays, ultra-supercritical (USC) power plants with large capacity and high parameter are considered to be feasible means to save energy and have rapidly developed word-wide. The double reheat USC unit is a new generation of USC unit which can improve the thermal performance compared with single reheat units [6]. According to Ref. [7], a double reheat unit with the inlet parameter 30.0MPa / 600 / 620 / 620 ℃ improves the heat efficiency by 2.4%-2.6%, compared with a common used USC unit with the inlet parameter 25.0MPa/600/600℃. The U.S. built the first double reheat unit with main parameter 34.4 MPa/649/566/566℃ in 1960s. Two 700 MW double reheat USC units in Japan were put into operation in 1990. The Taizhou power plant in China began to build double reheat USC units in 2012, and put it into operation in 2015. Over the past few decades, the double reheat USC units have received more and more attention for its rapid development all over the world. Zhao Z et al. [6] studied the exergy distribution system for a 1000MW double reheat USC power plant and provided the main reasons that led to exergy loss on the steam turbine. Rashidi et al. [8] investigated the thermodynamic analysis of a double reheat steam power plant. According to Ref. [9], component and process based exergy evaluation was performed on a coal-fired power plant in China, which provides guidance for energy-saving strategies. It pointed out that the exergy loss in the heat transfer process accounts for the largest proportion.

Though the high live steam pressure and temperature of USC unit improves its efficiency, there are still some imperfections which limit the improvement of its performance.

For example, the double reheat system causes great superheat degree of the first several extractions and the boiler exhaust temperature is extremely high, which leads to unreasonable energy-level matching and great exergy loss.

To reduce the superheat degree of the extractions, Liu et al [14] investigated the thermal performance of the steam and water cycle with single reheat after the installation of an additional outer steam cooler (AOC). Results show that the AOC is an effective method to reduce the superheat degree and improve the efficiency of the unit. Besides, Kjaer [15] proposed a regenerative steam turbine to utilize the superheat degree of the extractions. In this design, part of the extraction from the high pressure turbine enters the additional regenerative steam turbine not the regenerative heater. Extractions from the intermediate pressure turbine are replaced by those from the regenerative steam turbine. The superheat degree of the extraction is significantly reduced in this design, and the exergy destruction of regenerative heaters is reduced, which results in an overall improvement in efficiency.

To reuse the energy of the exhaust flue gas. Ref [10] proposed a low pressure economizer (LPE) based on the data of some 1000 MW typical power generation units in China and four possible arrangements of the LPE installation were proposed to compare its energy-saving effects. Results indicated that LPE connected with higher temperature section of the condensate line brings more reduction of standard coal equivalent (SCE). Wang et al. investigated the energy and water saving and the reduction of CO2 after the installation of LPE. Results show that the optimized measures can bring a reduction of SCE by 2-4g/ (kWh) [11]. Vladimir et al. [12] proposed an additional high pressure economizer installed at a long term running lignite-fired power plant. The results show that more than 30 MW of the flue gas waste heat is recovered, which brings an improvement in gross efficiency by 0.53 percentage points and 9.4 MW extra output power. In Ref. [13], the air preheater is divided into two stages to reduce the temperature difference in heat transfer process. Besides, a LPE is installed between the two air preheaters to obtain an appropriate flue gas temperature range. Thermodynamic and Technic-economic analysis are conducted to reveal the performance improvement. It was found that the SCE consumption can be reduced by 6.7(kWh).

Due to arrangement of boiler heating surface and design of regenerative system belong to deferent research area, researches concentrate more on the optimization of these two systems respectively, and ignored the joint optimization of boiler tail flue heating surface and regenerative system to achieve energy cascade utilization and improve system efficiency. To reduce both systems’ exergy loses, a theoretical optimization design of cascaded utilization of energy for a 1000 MW double reheat USC unit (optimization system) is proposed. Variations and energy saving effects of a double reheat ultra-supercritical thermal system (reference system) and the optimization system are analyzed to evaluate their performance.

1. **Reference system introduction and analysis**

**2.1 A double reheat USC power plant in operation (reference system)**

A typical coal-fired double reheat USC power plant in operation is chosen as the reference unit. The parameter settings that maximize continuous power are 310 bar / 600 ℃for the main steam, 610℃ for the reheat steam pressure, 33.5% for the proportion of the single reheat steam pressure to the main steam pressure, and 33% for the proportion of the double reheat steam pressure to the single reheat steam pressure[6]. The output power of the double reheat unit under the condition of THA load is 1000 MW. The unit consists of one super high pressure turbine (VHP), one high pressure turbine (HP), one intermediate pressure turbine (IP), and two low pressure turbines (LP). 10-stage regenerative system with four high pressure regenerative heaters (HRH), five low pressure regenerative heaters (LRH), and one deaerator (DEA) are adopted. Besides two additional outer steam coolers (AOC1, AOC2) are used to cool two extractions due to its high superheating degree. The exhaust steam pressure of the steam turbine is set 4.5kPa. The simplified schematic of the unit is presented in Fig 1.

从参考系统的锅炉虽然仍然采用Π型布置，但其内部受热面和传统一次再热锅炉有较大区别。

Furnace layout shown in Fig 2, the boiler furnace is composited by the membrane wall, along flue gas flow direction lays low-temperature superheater (Lts) screen tube, cold segment of high-temperature ﬁrst reheater (Csf), cold segment of high-temperature second reheater (Csf), High temperature super-heater (Hts), hot segment of high-temperature ﬁrst reheater (Hsf), hot segment of high-temperature second reheater (Hss), after which the flue gas channel is divided into the front flue and the back flue. The front duct arranges low-temperature first reheater (Ltfr) and the front-duct economizerz (Feco), the back duct arranges a low-temperature second reheater (Ltsr) and back-duct economizer (Beco). The rear flue is equipped with an air preheater(APH).

从锅炉受热面布置可以看出，锅炉主要受热面布置在炉膛之中，而水平烟道和竖井烟道中仅布置有空气预热设备。所以锅炉可以分成两部分:炉膛内完成水/蒸汽吸热的boiler，和尾部烟道内 完成空气预热的APH。



Fig 1 schematic of the double reheat unit on operation(reference system)



Fig 2 Boiler internal heat exchanger layout

**2.2 Thermal performance evaluation**

**为了能够对参考系统进行用分析，需要对其进行建模，利用ebsilon进行热力模型的简历需要输入一些主要参数**

Under design and off-design conditions some data need to be input as thermal calculation parameters. The main input parameters in the system simulation process are shown in Table 1，data such as extraction temperature/pressure, turbine back pressure and heat exchanger end differential are also needed.

In order to validate the correctness of the system, this paper selects turbine extraction mass flow as contrast value. The simulated and designed values of turbine extraction mass flow are shown in **错误!未找到引用源。** The error rate between the simulated and designed results is in the range of -2.91~2.6%, the design total steam extraction mass flow is 318.32kg/s，and the simulation value is 318.11, of which the error rate is 0.07%. The differences are in the allowed range, demonstrating the correctness and reliability of the simulation.

Table 1 The main input parameters during the simulation process on design and off-design situation（锅炉侧）

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Items | Parameters | Unit | THA | 75%THA | 50%THA | 40%THA |
| Superheat steam | Mass flow | t/h | 2533 | 1866 | 1212 | 972 |
| Pressure | bar | 311.4 | 231.2 | 153.2 | 123.6 |
| Temperature | ℃ | 605 | 605 | 605 | 605 |
| First reheater | Mass flow | t/h | 2318 | 1683 | 1112 | 898 |
| First reheater inlet | Pressure | bar | 105.3 | 77.1 | 51.2 | 41.3 |
|  | Temperature | ℃ | 427 | 428 | 433 | 435 |
| First reheater outlet | Pressure | bar | 103.2 | 75.5 | 50.1 | 40.4 |
|  | Temperature | ℃ | 613 | 613 | 613 | 613 |
| Second reheater | Mass flow | t/h | 2002 | 1481 | 997 | 811 |
| Second reheater inlet | Pressure | bar | 3.29 | 2.43 | 1.62 | 1.29 |
|  | Temperature | ℃ | 433 | 437 | 440 | 441 |
| Second reheater outlet | Pressure | bar | 3.05 | 2.24 | 1.48 | 1.18 |
|  | Temperature | ℃ | 613 | 613 | 613 | 605 |
| Feedwater | Temperature | ℃ | 314 | 302 | 276 | 263 |
| Economizer inlet | Pressure | bar | 346.4 | 250.2 | 161.2 | 128.8 |

Table 4 THA 工况下优化系统输入参数表（汽机侧）

|  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- |
| Items | | Parameters | Unit | THA | 75% THA | 50% THA | 40% THA |
| Turbine Exhaust parameters | | Pressure | kPa | 4.5 | 4.5 | 4.5 | 4.5 |
| Specific Enthalpy | kJ/kg | 2397.40 | 2425.3 | 2480.3 | 2510.9 |
| Extraction Parameters | 1th | Temperature | ℃ | 107.89 | 79.27 | 52.99 | 42.95 |
| Pressure | bar | 429.00 | 430.00 | 435.20 | 437.40 |
| 2th | Temperature | ℃ | 61.89 | 45.88 | 30.93 | 25.13 |
| Pressure bar | bar | 528.30 | 530.70 | 533.00 | 533.80 |
| 3th | Temperature | ℃ | 34.34 | 25.54 | 17.29 | 14.03 |
| Pressure bar | bar | 435.20 | 438.70 | 442.10 | 443.00 |
| 4th | Temperature | ℃ | 17.56 | 13.57 | 9.27 | 7.55 |
| Pressure bar | bar | 526.55 | 528.30 | 530.00 | 523.10 |
| 5th | Temperature | ℃ | 10.35 | 8.39 | 5.89 | 4.84 |
| Pressure bar | bar | 447.37 | 453.40 | 459.20 | 454.30 |
| 6th | Temperature | ℃ | 7.15 | 5.71 | 3.94 | 3.23 |
| Pressure bar | bar | 394.01 | 397.50 | 400.90 | 395.70 |
| 7th | Temperature | ℃ | 3.90 | 3.15 | 2.20 | 1.81 |
| Pressure bar | bar | 313.17 | 317.90 | 322.80 | 318.80 |
| 8th | Temperature | ℃ | 1.23 | 1.00 | 0.70 | 0.58 |
| Pressure bar | bar | 191.84 | 196.40 | 202.30 | 200.20 |
| 9th | Pressure | bar | 0.57 | 0.47 | 0.33 | 0.27 |
| Specific Enthalpy | kJ/kg | 2714.30 | 2724.20 | 2737.90 | 2736.60 |
| 10th | Pressure bar | bar | 0.21 | 0.18 | 0.13 | 0.11 |
| Specific Enthalpy | kJ/kg | 2573.40 | 2583.30 | 2597.50 | 2598.70 |

Table 2 Temperature profiles and exergy efficiency of heat exchangers gives the temperature profiles and exergy efficiency of regenerative heaters and the air preheater, revealing the energy match level of heat exchangers. For the air preheater, the temperature of flue gas at the inlet of the air preheater is 376℃ under full load, while the temperature of the air to be heated by the flue is 25℃, leading to the exergy loss of 27.17 MW. The temperature difference of the air preheater is calculated to be 72℃, and average temperature of the air is 177℃, causing the exergy efficiency of 77.16%.

Table 2 Temperature profiles and exergy efficiency of heat exchangers

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Components | Hot fluid  temperature | Cold fluid  temperature | Fluid temperature difference | Exergy efficiency(%) | |
| APH | 376．00 | 23.00 | 353.00 | 77.16 | |
| AOC1 | 526.56 | 304.51 | 222.05 | 87.86 | |
| AOC2 | 527.48 | 304.51 | 222.97 | 90.09 | |
| HRH1 | 417.07 | 273.62 | 143.45 | 96.18 | |
| HRH2 | 314.53 | 240.73 | 73.8 | 96.26 | |
| HRH3 | 434.54 | 205.89 | 228.65 | 93.67 | |
| HRH4 | 316.51 | 186.48 | 130.03 | 95.68 | |
| DEA | 447.37 | 163.63 | 283.74 | 91.81 | |
| LRH6 | 394.01 | 140.46 | 253.55 | 89.29 | |
| LRH7 | 313.17 | 104.49 | 208.68 | 86.83 | |
| LRH8 | 191.84 | 82.42 | 109.42 | 89.18 | |
| LRH9 | 116.61 | 59.25 | 57.36 | 81.40 | |
|  |  |  | | |  | |  |

Compared with the air preheater, the regenerative heaters have better exergetic performance. However, it is found that the temperature difference between the hot and cold stream at the inlet high temperature regenerative heaters is also relatively large, which causes irreversible loss. To reduce the superheat degree of the 2nd and 4th extractions and to heat feedwater, two additional outer steam coolers are installed. Results show that the inlet temperature difference of HRH2 and HRH4 is greatly reduced, and the exergy efficiency is improved correspondingly. However, the inlet temperature difference between the cold and hot stream of other high temperature regenerative heaters and DEA remains very high. Moreover, because of the material restriction of the water wall, the temperature of feedwater at the inlet of boiler is restricted to 315℃. In this case, the first extraction has to be throttled to ensure the temperature of feedwater not exceeding the restriction. The pressure of the first extraction is throttled from 106.70 bar to 88.56 bar, and the temperature is decreased from 425.60 ℃ to 413.77℃. The throttling of the first extraction will certainly cause extra irreversible loss.

1. Establishment and performance evaluation of optimization system

3.1 优化系统的建立

The layout of the optimization system is presented in Fig 4. In the optimization system, the air is not heated by flue gas in the conventional air preheater, and 8 new air preheaters (APH1~APH8) using several extractions to heat the air is adopted. The new air preheater is similar with regenerative heaters in function. As shown in Fig 3The extracted steam is sent to the regenerative heater to heat the condensate or feedwater, and then the drainage from the air preheater joins with that of corresponding regenerative heater. It should be mentioned that not all the extractions are available for air heating. For the last two stage extractions, their pressure is much lower than barometric pressure which makes it extremely difficult to maintain such low pressure environment inside the heat exchangers, and APH8 shall be shut down under low load for the same reason. Besides, to reduce the superheat degree of extractions entering the heat exchangers and to improve the air temperature, four additional outer steam coolers (AOC1- AOC4) are installed at the outlet of APH1. Since extractions from the 2nd to the 5th have superheat degree, they are sent to AOCs in the order of temperature level. For example, the 2nd extraction has the highest temperature, and is applied to heat the air in the 1st AOC, and then the cooled steam from AOC1 is sent to HRH2 and air APH2 to heat the water and air respectively.



Fig 3 Process ﬂow diagram of an RH and APH

As shown in Fig 4, the flue gas at the outlet of economizer is utilized to heat the condensate and feed water by four LPEs, which are paralleled to HRH2, HRH4, LRH6, and LRH8 respectively. Part of the condensate and feed water at the inlet of regenerative heater are sent to the LPE, then the heated water join with the main feedwater/condensate. Considering the acid dew point temperature, the flue gas temperature at the outlet of the last LPE is set 95℃[18]. The temperature of heated water from LPE is set little higher than the feedwater from the corresponding regenerative heater.



Fig schematic of the double reheat unit with steam/air preheater and LPE (optimization system)

3.2 **Model establishment and system analysis method**

2.1 System simulation method and main assumption

In this paper, the thermodynamic cycle of the thermal system under different loads are simulated with EBSILON ®Professional for its flexibility and level of detail. EBSILON is a power plant simulation tool which can calculate thermodynamic quantities. The simulation model gives the detailed data with high degree of reliability to calculate the thermodynamic state.

EBSIOLON's simulation and parameter calculation of the equipment is based on experimental characteristics of units during the process of modeling design and off-design conditions. However, only some characteristic curves of turbine can actually be obtained, while boiler and heat exchanger data cannot be obtained, which can only use the system built-in data as an alternative. It’s needed to assume that the difference between the built-in data and the actual data to meet the error requirements.

The following assumptions need to be made in order to be able to simulate correctly.

(1) The operation of the power plant is considered to be in a steady state.

(2) The thermal efficiency of the equipment is calculated from the thermal balance of the input and output parameters of the components, assuming that the calculation error requirement can be met.

(3) According to the heat map, set the upper and lower end difference of the heat exchanger as a fixed value, and do not change among different condition.

(4) Do not take the pips complexity and equipment’s installation difficulty of the optimization system into consideration.

2.2 System analysis method

为了能够对优化系统与参考系统进行评估，揭示系统优化的内在机理，本文引入热力学第二定律的分析方法即用分析方法。

Exergy analysis , is adopted to reveal the location, the magnitude and the sources of thermodynamic inefficiencies of the unit. Usually, exergy loss and exergy efficiency are chosen as evaluation indices of the thermal system or an individual component，The general exergy balance of component can be expressed in the following form:

(1)

Exergy efficiency can be calculated in the following form:

(2)

For different equipment under stable operating conditions, equation(1)、 (2) has different forms as shown in Table 5[?]

For flue gas、air、water and steam, the specific exergy can be calculated by equation 3.

(3)

Then the total exergy rate associated with a ﬂuid stream becomes:

the fuel speciﬁc exergy is calculated as [?]：

Where LHV refer to lower heating value of fuel，C、H、O、N refer to share of carbon, hydrogen, oxygen, nitrogen by element analysis.

Table 3 The exergy destruction rate and exergy efﬁciency equations for plant components

|  |  |  |
| --- | --- | --- |
|  | Exergy destruction rate | Exergy efficiency |
| Boiler |  |  |
| Pumps |  |  |
| Heaters |  |  |
| Turbine |  |  |
| Condenser |  | ?? |
| Cycle |  |  |

3.3 优化系统建模与热力学分析

Based on the analysis of the reference system, the temperature difference of the heat exchangers is very high and exergy efficiency is rather low. Using turbine extractions to heat air can realize the reduce the heat transfer temperature difference and improve efficiency. The economizer outlet flue gas is used to heat low-temperature economizer can also reduce the loss of regenerative system.

In order to complete the heat balance calculation of the thermodynamic system, the key parameters in the optimization system need to be set.

The optimization system’s parameters are based on reference system, so the main parameters must be the same. The main parameters settled are including: superheat steam’s mass flow, pressure and temperature; First/ second reheater’s inlet and outlet pressure and temperature; Turbine Exhaust’s pressure and temperature; turbine extraction’s pressure and temperature/ specific enthalpy etc. The main parameters settled under THA situation can be found in Table 1 and Table 4.

The results of exergy analysis of the optimization system are presented in Table 5. It is found that the optimization system has an improvement in exergetic performance. The total exergy loss of the cascaded utilization system is 1096.52 MW, which is 32.28 MW lower than the original. The new system also brings the improvement from 46.98% to 47.99%. Results show that the exergy loss of boiler is 855.72 MW, leading to the increment of exergy efficiency by 0.18%. In addition, there is almost no change of the exergy parameters of the steam turbines, generator and condenser. For the air heating system and regenerative system, great change happens compared with the formal system, and detailed analysis will be carried out next.

Fig 5 Optimization system exergetic efficiency analysis

Table 5 results of exergy analysis of the optimization system

|  |  |  |  |  |  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  |  | Boiler | Turbine | | | | APH | Regenerative System | | | | Other | TOTAL |
|  |  | VHP | HP | IP | LP | HRH | DEA | LRH | LPE |  |  |
| Exergy loss | MW | 872.22 | 9.95 | 7.55 | 12.44 | 20.29 | 16.45 | 4.00 | 0.57 | 6.26 | 10.73 | 142.42 | 1116.72 |
| Exergy efficiency | % | 58.53 | 95.15 | 96.6 | 96.25 | 89.71 | 88.73 | 96.14 | 95.47 | 86.8 | 91.62 | -- | 47.97 |

**4.4 two systems comparative analysis**

Through the analysis of the reference system and the optimization system, it can be seen that the exergy efficiency of the optimization system is improved and the exergy loss is reduced。Comparing the proportion of different components in the total exergy loss between the two systems, we can see that in addition to the turbine, the exergy loss of all components is reduced. Among them, the air preheating system efficiency is the most obviously improved，to 38.94%.

Table 6 Items exergy efficiency of optimization system and reference system

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Items | | Optimization  System(%) | Reference System(%) | Difference  (%) | Difference rate |
| Boiler（without AP） | | 40.99 | 41.09 | -0.10 | -0.25% |
| Turbine |  | 3.01 | 2.95 | 0.06 | 2.03% |
| Air Preheater | | 0.78 | 1.27 | -0.50 | -38.94% |
| Other |  | 6.69 | 7.13 | -0.44 | -6.18% |
| Regenerative system | | 1.01 | 1.05 | -0.04 | -3.35% |



Fig 6 variation of mass flow rate of extractions

The variation of mass flow rate of extractions is presented in Fig 6. Mass flow rate of the 1st, 3rd, 5th and 8th extractions of the optimization system is more than that of the base model, and the rest extractions except the last two, has the opposite variation. The impact of system optimization on extraction is twofold. Air is heated by extractions, which will cause an increment in mass flow. And the use of LPE can saves extractions.so the total extraction keeps unchanged. Turbine total extraction steam volume change affect its exergy efficiency by 2.03%.

The air is heated by the staged air preheaters and additional outer steam coolers in the cascaded utilization system. It can be seen from Table 5 that the total exergy loss of the air heating system is 16.35 MW, 10.82 MW lower than that of the original air preheater which is 27.17 MW. Fig 7 shows that the exergy efficiency of APHs and AOCs is higher than 82%, except APH8. For the air preheaters, the improvement in performance results from more reasonable energy utilization and the heat transform method. In the conventional air preheater, the air is heated by high temperature flue gas through convection. The working principle of the staged air preheaters in the optimization system is the same as regenerative heaters. The superheated vapor is firstly cooled to saturated steam through convection, then to saturated water through phase change heat transfer, and the latter process accounts for the main part. In this case, the temperature difference is greatly reduced, causing the improvement in performance.



Fig 7 exergy analysis of the air heating system

It is also found that the APH8 has the greatest exergy loss and lowest efficiency of all the air preheaters. According the exergy loss equation Eq.4, the great exergy loss of APH8 is mainly because the average temperature of the cold stream is extremely low, which is only 62 ℃, even more than 110 ℃ less than that of the conventional air preheater. The low temperature level of the air of APH8 also causes its inefficiency. Besides, compared with APHs, the change of heat transfer method of AOCs causes larger temperature difference between the hot and cold stream. However, the efficiency of AOCs is relatively high, owing to the high temperature level of the cold stream.

(4)

As can be seen from Fig 8 the logarithmic mean temperature difference of the air preheater is less than the 72℃ mentioned in section 4.2, with the temperature difference of AOCs being from 44 to 63 ℃ and the APHs being from 11 to 26 ℃, with the maximum temperature difference occurring at AOC4 is 62.75℃, the minimum temperature difference is APH5 12.00 ℃.



Fig 8 temperature difference of the air preheater

The total exergy loss of the LPEs is 10.73MW, and the overall exergy efficiency is 91.62%, much higher than that of air preheater. As shown in Fig.6, the temperature difference of the LPEs is between 23 ℃ to 42 ℃, less than that of the original air preheater. The LPE1 and LPE4 have the greatest exergy loss of all the LPEs, which mainly results from their great heat transfer rate. Besides the exergetic performance of the LPEs keeps falling along the flow direction of flue gas. This is because the higher temperature of the cold stream, the greater exergy efficiency, for the temperature difference changes little in heat transfer process.



Fig 9 exergy analysis of the LPEs

The layout of the optimization system also changes the exergetic performance of regenerative heaters, shown in Fig.7. The total exergy loss of regenerative heaters is 11.60 MW, which is greatly reduced compared with that of the base model. On the one hand, owing to the heat injected by flue gas, less extracted steam is needed for the HRH2, HRH4, LRH6, and LRH8 which are paralleled with LPEs, which causes less exergy loss of corresponding regenerative heater. On the other hand, the LPEs heat condensate and feedwater to higher temperature level than the regenerative heaters, leading to less heat transfer rate of HRH1, HRH3, DEA and LRH7, which will also reduce the exergy loss.



Fig 10 comparative exergy analysis of regenerative heaters

There are three factors causing the improvement in exergetic performance. Firstly, for HRH1, HRH3, DEA and LRH7, due to higher temperature level of the water from LPE, the exergy of the inlet feedwater increases, leading to the efficiency improvement. Secondly, the installation of AOCs causes the reduction of the superheat degree of the extracted steam entering the HRH3 and DEA, and reduce its exergy, namely . At last, compared the original regenerative heaters, an extral part of drainage from corresponding air preheater of optimization system is used to heat the feedwater. Compared with the extracted steam, the drainage from air preheaters has lower exergy, which reduce the irreversible loss to some extent.

Table 4 shows the output power and SCE consumption of the reference system and optimization system. It is found that the optimization system brings an improvement of output power by 11.02 MW, and a reduction of SCE consumption by 5.5 g/kWh. The finding indicates that the optimization system is more efficient, and the energy saving effect is evident.

Table 7 results of exergy saving effect

|  |  |  |
| --- | --- | --- |
| items | Output power (MW) | SCE consumption (g/kW h) |
| Reference system | 999.1 | 257.6 |
| Optimization system | 1011.19 | 252.10 |

**Performance evaluation at partial load**

Considering that large USC power plants always operate underpartial load conditions for peak regulation, it is necessary to study the thermal performance of double reheat USC power plants under.

Four typical operation conditions, namely, THA load, 75% THA load, 50% THA load, and 40% THA load conditions, were selected for thermodynamic analyses in the present study. Data of reference system and optimization system is taken from heat balance chart under variable conditions. Some major parameters can be found in Table 1.

As shown in Fig 11, Optimizing system efficiency does not always outweigh the reference system and even worse at low loads. Optimized system efficiency was 47.52 % for the THA case and 46.51 % for the reference system, increased by 1%. ，相当于煤耗降低了5.5g/kWh, However, under the 75% THA condition, the optimized system was 46.56% and the reference system was 45.90 %, a difference of 0.66 %. When the load is equal to 50% THA, the difference between the two systems is 0.43 %，相当于煤耗降低了2.9g/kWh. When the load continues to decrease, 优化系统和参考系统的用效率更加接近。



Fig 11 系统火用效率对比图

采用。THA工况下可以看到AP和Boiler子系统的ELDR都为负值说明优化系统用损失小于参考系统，而OTHER则相反为正，参考系统用损失较小，TURBINE和RS子系统则ELDR几乎相等，总的用损为优化系统小于参考系统。随着系统负荷减小到75%THA，优化系统和参考系统的Turbine和RS系统依旧几乎相等，但是AP和Boiler系统中优化系统的用损在接近参考系统，而Other中的用损大于参考系统更多，虽然依然总的用损为优化系统小于参考系统但优化系统的优势越来越不明显。随着负荷继续降低情况和THA到75%THA类似，



Fig 12 不同分系统变工况下单位发电量 相对用损变化

1. **Conclusion**

This study presents an optimization design of cascaded utilization of energy for a 1000 MW double reheat ultra-supercritical unit. Thermodynamic analysis is conducted to reveal the comprehensive effects of the optimization measures. The optimization system is found to be effective in overcoming the imperfections, and may provide an effective method for the optimization of a double reheat system in USC unit. The following conclusions are drawn from this study:

(1) The thermodynamic analysis indicates that the cascaded utilization system decreases the temperature difference of in air heating and flue gas cooling process which achieves the cascaded utilization of energy, and the superheat degree of the 3rd and 5th is reduced greatly. Besides, more exhaust heat of the flue gas is recovered since the LPE cools the flue gas to lower temperature.

(2) The layout in the cascaded utilization system changes the mass flow rates of extractions, which causes increment of output power of the generator and increment of the cold source loss.

(3) The optimization system improves the temperature of secondary air, which improves the fire condition in the combustion chamber. The temperature of secondary air under full load in the optimization model is 365 ℃, which is 34 ℃ higher than the design value.

(4) For the first extraction, throttling loss is not existed while the feedwater of boiler improves little.

(5) The optimization system theoretically brings remarkable energy saving effects. The simulation results show that the optimization system can reduce the SCE consumption by 5.6 g/kWh under full load, and can even bring a reduction of 2.9 g/kWh under 50% THA load.

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