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Exergy analysis of the turbine system in a 1000MW double reheat ultra-supercritical power plant

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Abstract

This study aims to achieve exergy distribution in a turbine system for a 1000 MW double reheat ultra-supercritical power plant. An exergy balance equation is used to identify the real points of energy loss in the turbine system, which can provide a reference for optimizing operation and conserving energy. Results show that irreversibilities yield the highest exergy loss in the turbine, particularly in the very high pressure (VHP) and low pressure (LP) cylinders. Compared with that in the turbine of a single reheat unit, exergy loss in the studied turbine is higher and exergy efficiency is lower. For the regenerative system, exergy loss is lower and exergy efficiency is higher in the double reheat unit than in the single reheat unit. Thus, the layout of the regenerative heaters in the double reheat unit is more reasonable. However, exergy loss in high pressure regenerative heater Nos. 3 and 6 and low pressure regenerative heater Nos. 7 and 10 in the heat recovery system is extraordinary because of different reasons. Exergy loss in the condenser of the double reheat unit is relatively smaller than that in the condenser of the single reheat unit.

Key words: double reheat ultra-supercritical power plant; turbine system; exergy balance equation; exergy loss; exergy efficiency

1. Introduction

At present, environmental problems are becoming increasingly serious along with the level of technology and development in the country. Developing large-capacity, high-parameter, low-pollution, and remarkably efficient coal-fired power-generation techniques are the most feasible means to save energy [1]. The high-parameter double reheat ultra-supercritical (USC) unit is a new generation of USC units that can be ranked between existing USC units and the European Union-financed AD700 project [2]. Double reheating not only can fulfill the exhaust steam humidity requirement of a low-pressure cylinder, but also can improve the heat efficiency of a unit. Compared with the frequently used parameter setting of 25.0 MPa/600/600 °C in a USC unit, setting the turbine inlet parameter of a double reheat unit to 30.0 MPa/600/620/620 °C can improve the heat efficiency of the unit by 2.4%~2.6% [3]. M. M. R et al. studied that the efficiency of double reheat power can be theoretically by 1.0%~2.0% higher than that of single reheat unit [4]. During the 1950s, the U.S. and several other countries began studying the USC unit. The U.S. built two 325 MW (34.4 MPa/649/566/566 °C) double reheat units during the 1960s, which became the first USC double reheat units in the world. Japan built two 700 MW (31 MPa/566/566/566 °C) double reheat USC liquefied natural gas units, which started operation in 1989 and 1990. In addition, a power plant in Denmark put into operation a 412 MW (28.4 MPa/580/580/580 °C) double reheat unit in 1997. After a year, a 420 MW unit with a steam setting of 29 MPa/582/580/580 °C was put into operation. This unit used a deep seawater cooling scheme, in which cooling water was 10°C, thereby achieving an efficiency of 47%. Building a double reheat USC unit is possible, based on the maturity of thermal power generation technology in China, the successful operation of supercritical and ultra-supercritical units, and the superiority demonstrated by these units. Since

2009, many research institutions and universities have delved into the design of the double reheat USC unit in China. The Taizhou Power Plant of China Guodian Corporation and the Huaneng Anyuan and Laiwu Power Generation Plant have begun building double reheat USC units in 2012. These units started operation in 2015. The Huadian Jurong Power Generation Plant and the Shenhua Guohua Beihai Power Plant are currently building units to add to the growing list of double reheat USC units in China.

With the rapid development of double reheat USC units both in China and other countries, the economic analysis and optimization of such units is becoming a popular area of research. The steam turbine is an important component of double reheat USC unit, and its performance directly affects the efficiency of the entire power plant. Sven K et al. presented a modified steam turbine design with a small separate turbine that drove a 100% boiler feed pump combined with a balancing motor, and this design solved the problem of superheated steam from intermediate pressure (IP) turbines and paved the way for an affordable double reheat technology[5]. Long S et al. investigated the causes and influence factors of jet deviation in a dual-circle tangential-firing single-furnace USC boiler [6]. Weiping Y et al. set up a 1350 MW secondary reheat unit principle thermal system. Calculation and analysis were performed for different configurations of the feedwater pump steam turbines by considering the influences of two versions of the feedwater pump steam turbines on reheat steam flow rate, unit heat rate, and reheater design [7]. Junjie Y et al. used theory of equivalent enthalpy drop as the basis to provide the principles for the local quantitative analysis of the economics of the thermal system of a supercritical unit with a double reheat cycle and developed a set of complete quantitative calculation methods [8]. Gang X et al. presented two superheat utilization schemes to extract steam from a double reheat USC power plant;

they then performed thermodynamic and techno-economic analyses to determine the energy-saving effects of the proposed schemes[9].

All the aforementioned works rely on heat balance analysis based on the first law of thermodynamics, namely energy-based methods. In this method, all forms of energy are considered to be equivalent. The loss of quality of energy is not taken into account and energy flows into and out of a system along paths of mass flow, heat transfer, and work [10]. The works based on the energy-based methods are easy to understand, but only the quantitative variation of energy is considered and its qualitative variation is disregarded. Therefore, the use of such a method for assessing the improvement possibilities of a process can lead to a distorted vision of studied system and cannot characterize the irreversibility of processes within the system [11, 12]. Also, these works overemphasize the thermodynamic cycle of the turbine and the loss of the condenser, but disregard energy-level matching and irreversible energy loss. In contrast, exergy analysis can characterize the work potential of a system. Exergy is defined as the maximum theoretical useful work obtained in relation to its environment [13-15]. Exergy analysis based on the second law of thermodynamics can accurately represent the maturity and utilization of a device [16]. Due to its ability to capture the variation of the energy grade, it has been used widely in recent years. Sandhya H et al. performed an exergy analysis on a USC power plant and built a model based on the concepts of coal combustion, energy balance, enthalpy balance, entropy changes, and heat transfer of the steam power cycle [17]. Isam H A identified and quantified the sites having largest energy and exergy losses of a power plant [18]. Sairam A et al. performed energy and exergy analyses on a 660 MW coal-fired supercritical thermal power plant at 10%, 80% and 60% normal continuous rating conditions under constant pressure and pure sliding pressure operation [19]. Lingnan W et al.

figured out and compared the exergetic performances of each component on a typical 600MW coal-fired power plant [20]. Their work focused on exergy loss distribution in the regenerative system of the power plant, and ignored exergy loss analysis on each cylinder of the turbine, regenerative heaters and the condenser system. Yongping. Y, Yuanyuan. L et al. analyzed the USC and double reheat USC unit based on the exergy analysis, and pointed out the exergy loss, but the reasons that led to the exergy loss were lost[21, 22]. Furthermore, their proposal required formulating a highly complicated and professional calculation equation for the unit. In the present study, exergy loss analysis on a 1000 MW double reheat USC power plant using Aspen plus was performed. The major components of the turbine system and the regenerative heaters that sustained heavy energy losses as well as the irreversibility degree of the system in double reheat USC were investigated, compared with a single reheat USC (Refs [23]). This study aims to construct a simple and effective analysis model for a double reheat USC unit as well as to provide remarkable insights into the operation and energy-saving optimization of such unit.

<i>Nomenclature</i>		<i>subscripts</i>	
TMCR	Turbine maximum continuous rating	Q	Parameters of heat
I	Exergy loss (W)	in	Parameters in the system
E_{in}	Exergy flowing into the system (W)	out	Parameters outside the system
E_{out}	Exergy flowing out of the system (W)	s	Drain parameter at the corresponding level
η	Exergy efficiency (%)	$i-1$	Drain of upper stage
HRH	High pressure regenerative heater	w	Parameters of feedwater
LRH	Low pressure regenerative heater	i	i-stage extraction
DEA	Deaerator	j	is set at 1 to 5, representing VHP, HP, IP, LP1, and LP2
D	Mass flow rate (kg/s)	c	Parameters of condenser
D_j	Mass flow rate of steam that enters the turbine (kg/s)	c_1	Parameters of drain from the H1
D_j'	Mass flow rate of steam that exits the turbine (kg/s)	c_2	Parameters of exhaust from LP1
e_j	Special exergy of steam that enters the turbine (J/kg)	c_3	Parameters of exhaust from LP2
e_j'	Special exergy of steam that exists the turbine (J/kg)	c_4	Parameters of exhaust from the external steam turbine
e	Specific exergy (J/kg)	$tur, loss$	The loss of turbine
W	Work (W)	$rh, loss$	The loss of the regenerative heaters
θ	Energy efficiency (%)	c, in and c, out	Parameters of cold side
h	Specific enthalpy (J/kg)	h, in and h, out	Parameters of hot side

2. System simulation

2.1 System description

The double reheat technique is adopted for the USC unit. The parameter settings that maximize continuous power are 31MPa/600°C for the main steam, 610°C for the reheat steam temperature, 33.5% for the proportion of the single reheat steam pressure to the main steam pressure, and 33.0% for the proportion of the double reheat steam pressure to the single reheat steam pressure. The plant consists of one single-flow super high pressure (VHP) turbine, one double-flow high pressure (HP) turbine, one double-flow intermediate pressure (IP) turbine, and two double-flow low pressure (LP)

turbines. The steam emission part of the VHP turbine enters the boiler reheater to undergo a reheating process and is emitted into the HP after fulfilling the steam parameter requirements. The steam emission part of the HP turbine performs double reheating and then enters the IP turbine to conduct further work. Compared with the traditional single reheat unit, the double reheat unit has lower pressure and significantly improves the specific volume of IP, with its volume flow being more than twice as high as the equal-capacity single reheat unit. However, the double reheat unit enables more extra heat to be absorbed by the reheater, which makes deploying the heating surface of the boiler more difficult [7]. Accordingly, the unit adopts a 10-stage regenerative system with four high pressure regenerative heaters (HRHs), five low pressure regenerative heaters (LRHs), and one deaerator (DEA). Simultaneously, the unit increases an extraction steam turbine that shares the same extraction pipe with the DEA, which directly drives the feed pump. The steam temperature and the superheat degree of the extractions of HRH2 and HRH4 locating after the first and second reheaters are extremely high. Directly entering the regenerative heater of the corresponding level will result in a high degree of superheating and under usage of energy levels for steam extraction. Therefore, the unit utilizes two additional outer steam coolers to take advantage of the superheat from the extractions of HRH2 and HRH4 (Fig. 1). The process flow diagram of the double reheat USC power plant is shown in Fig. 1. The exhaust steam pressure of the steam turbine is set as 0.045bar.

2.2 Model description and methodology

In this study, the inlet steam entering the turbine undergoes a series of isentropic expansion process, which are simulated by COMPR block [24, 25]. COMPR calculates the power plant required (or produced) given the pressure ratio, isentropic, polytropic and mechanical efficiencies.

The accuracy of the results depends on the efficiencies specified [26, 27]. COMPR block in isentropic mode calculates the net work output from the change in enthalpy for isentropic expansion, then translates this to net work output using isentropic and mechanical efficiencies: $W = \eta_s \eta_m \dot{m} \Delta h$. The heat exchanger is simulated by two heat exchanger blocks, which represent the cold and hot sides of the heat exchanger. In this manner, the amount of heat exchange can be obtained when modeling the heat exchange process. The fluid separation module FSPLIT is used to shunt steam, and one part of the stream flows into the regenerative heater whereas another part flows into a compressor block for further expansion [28, 29]. The DEA is similar to a logistics mixer simulated by a stream mixer block that follows the principle of mass balance and energy balance. Fig. 2 shows the simplified simulation diagram of the turbine system. In this figure, the dashed line represents work and heat flow, whereas the solid line represents the working medium. The main stream of the cold side from HRH1 is divided into two parts, and flows into two external steam coolers, and is finally fed into the economizer for the next circulation. The input parameters of the simulation process are listed in Table 1. Compared with a conventional reheat unit, the double reheat unit has two regenerative heaters, two external steam coolers, and a VHP, which will certainly change exergy loss distribution. The previous studies determined that exergy loss distribution mainly occurred in the steam turbine, the condenser, and the regenerative system [21, 22]. A series of irreversible processes generated after the steam flows through the turbine system results in irreversible loss. This study adopts exergy loss and exergy efficiency as evaluation indices, which are defined as follows in a steady flow system.

$$I = E_{in} - E_{out} + E_{Q,in} - E_{Q,out} + W_{in} - W_{out} \quad (1)$$

$$\eta = \frac{\text{exergy earned}}{\text{exergy cost}} \quad (2)$$

The following assumptions are simplified to achieve the simulation of the thermal system:

- 1) Disregard the pressure and exergy loss of the pipe.
- 2) Disregard the shaft seal loss, mechanical loss, and thermal loss.
- 3) Consider the unit under stable TMCR condition.
- 4) Mechanical efficiency is equal to 0.98 during the different stages of the VHP, HP, IP, and LP turbines.
- 5) Temperature and pressure are equal to 298.15 K and 1atm, respectively, under ambient conditions.

3. Simulation results and discussion

The parameter contrast between the design values and the simulation results of the main stream are listed in Table 2. As indicated in the table, the tolerances of the analog values and the design values are within the allowable range, which proves the accuracy and reliability of the simulation results. The two-stage and four-stage extractions are the steam after the first and second reheated steam, which flow through the external steam cooler and then into the regenerative heater. The extraction temperature is reduced by 217.6 °C and 216.7 °C, and the superheat of the extraction steam is decreased by 215.9 °C and 216.3 °C, respectively, which reduces irreversible loss during the heat transfer process as well as the improvement of energy utilization. As mentioned earlier, the external steam turbine and the DEA share the fifth extraction, and the mass flows are 39.3 kg/s and 12.5 kg/s, respectively.

3.1 Exergy analysis of the regenerative system

The regenerative heater is divided into the surface heater and the hybrid heater according to different methods of internal steam and water contact, as shown in Figs. 3 and 4. Hot vapor comes

directly in contact with water and then mixed in hybrid heaters. The DEAs utilize the thermal deaeration method, whose construction is similar to that of a hybrid heater. The hot and cold fluids flow into different pipes in the surface regenerative heater and indirectly transfer heat through the metal tube wall. All the HRHs and LRHs adopt this type of heater in the studied unit. The external steam cooler is actually a part of the regenerative heater, as mentioned earlier. Thus, it can reduce the degrees of superheat and exergy loss in the heat exchange process. The input exergy is equal to the sum of the output exergy and the exergy loss caused by the exergy balance principle in the heater. Exergy loss analysis can be expressed in the following forms.

The exergy balance equation of the surface regenerative heater can be expressed as follows:

$$D_i e_i + D_{s(i-1)} e_{s(i-1)} + D_{w(i+1)} e_{w(i+1)} = D_{wi} e_{wi} + D_{si} e_{si} + I_i \quad (3)$$

Exergy efficiency:
$$\eta = \frac{D_{wi} (e_{wi} - e_{w(i+1)})}{D_i (e_i - e_{si}) + D_{s(i-1)} (e_{s(i-1)} - e_{si})} \quad (4)$$

The exergy balance equation of the hybrid regenerative heater can be expressed as follows:

$$D_i (e_i - e_{w(i+1)}) + D_{s(i-1)} (e_{s(i-1)} - e_{w(i+1)}) = I_i + D_{wi} (e_{wi} - e_{w(i+1)}) \quad (5)$$

Exergy efficiency:
$$\eta = \frac{D_{wi} (e_{wi} - e_{w(i+1)})}{D_i (e_i - e_{w(i+1)}) + D_{s(i-1)} (e_{s(i-1)} - e_{w(i+1)})} \quad (6)$$

The performance analysis of the regenerative system in this double reheat USC unit and its comparison with a traditional 1000MW single reheat unit [30] are shown in Fig. 5 and 6. Fig.5 shows the exergy loss and exergy efficiencies in the two types of units, and the Fig.6 expresses the exergy loss proportion in a certain heater to the sum of all regenerative heaters. The regenerative system of a single reheat unit has three HRHs, four LRHs and one DEA. The three-stage extraction comprises the steam next to the reheated steam. The parameters of the main steam and the reheat steam are 254bar/571°C and 41.1bar/569°C, respectively, under the same output power condition.

The pressure of the deaerator is 10.44bar similar to that in double reheat unit (11.3bar).

The corresponding exergy loss of the single reheat unit is more than that in the double reheat unit, as clearly shown in Fig. 5 and 6 in the regenerative system. The total exergy loss in double reheat unit, i.e., 24.2 MW, is less than the exergy loss of the single reheat unit, i.e., 28.9 MW. The exergy loss in HRHs, DEA and LRHs in double reheat USC is 9.99MW, 1.56MW and 12.64MW and all less than that 13.1MW, 3.1 MW and 12.7MW in single reheat unit. The average exergy loss of the regenerative system that adopts the double reheat unit is considerably lower than that in the single reheat unit. Meanwhile, the exergy efficiency of the double reheat unit is higher than that in the single reheat unit, which proves that the equipment layout of the double reheat unit is more reasonable. However, several regenerative heaters exhibit high exergy loss, and exergy loss distribution is uneven, which should be further optimized. Therefore, we perform the following analysis for the unit.

As shown in Fig.7, the mass flows of HRH2, HRH3, HRH4, LRH7, and LRH8 are the sum of the extraction at the corresponding level and the drain of the upper stage in the double reheat unit. The exergy loss is directly proportional to the temperature difference and the mass flow rate, but inversely proportional to the product of the heat transfer temperatures of the cold and hot fluids, as shown in the exergy loss equation, defined as $\delta E = mT_0\delta q(1/T_c - 1/T_h)$. That is the larger the temperature difference or mass flow would result in the larger the exergy loss. Fig. 5 and 6 show that exergy loss is mainly concentrated in HRH3, LRH6, LRH7 and LRH10. The reasons for exergy loss in these components are analyzed as follows: 1) Compared with HRH1 and HRH2, HRH3 has greater heat transfer temperature difference and larger mass flow, which lead to larger exergy loss; 2) For LRH6, the mass flow rate is significantly less than those in HRH1 and HRH2,

but the temperature difference is greater, which results in higher exergy loss. Temperature difference is the actual reason of the huge exergy loss in LRH6; 3) Compared with LRH6, LRH7 exhibits the same heat transfer temperature difference. However, its mass flow is nearly twice as that of LRH6, which causes exergy loss with the same multiple; 4) For LRH10, the temperature difference and mass flow rate are lower than those of HRH1 and HRH2, but the temperature of its heat transfer fluid is extremely low, thereby LRH10 exhibits higher exergy loss. In fact, LRH10 is extremely difficult to optimize in the actual production process. On the basis of the aforementioned results, exergy loss among different heaters is caused by various reasons. For the studied unit, the temperature of HRH3 and LRH6 should be reduced, along with mass flow in HRH3 and LRH7. Accordingly, if we intend to reduce exergy loss in the regenerative process or increase the efficiency of the regenerative system during the actual optimization process, various optimization measures for different heaters should be considered. To optimize the regenerative system we can split the steam extraction that has larger mass flow rate and heat transfer temperature difference into several streams, and boot part of the stream into the heaters with higher temperature in cold side. However, since the unreasonable structure is also a reason that leads to the larger temperature difference, changing the layout of the thermal system and the structure of the heater can also optimize the regenerative system performance.

3.2 Exergy analysis of the turbine

The turbine consists of one VHP, one HP, one IP, and two LPs. Its internal irreversible loss is mainly caused by irreversible expansion and throttling loss, including static friction, impeller blower, fast steam valve throttling, moisture loss, etc. Given these losses, valid enthalpy drop is less than the ideal enthalpy drop in the steam turbine, and thus, conducting analysis for each turbine

cylinder is necessary to understand the internal distribution of each turbine cylinders in the turbine.

The exergy balance equation can be expressed as:

$$D_j e_j = D_i e_i + D_{i+1} e_{i+1} + \dots + D_{i+m} e_{i+m} + D_j' e_j' + W + I_j \quad (7)$$

Table 3 provides the exergy parameters of the double reheat and single reheat USC units. Evidently, the exergy efficiency of each cylinder in the double reheat turbine demonstrates the same changing trend as the traditional single reheat unit, i.e., the IP turbine possesses maximum exergy efficiency, followed by the HP, and the LPs have the lowest. The first reheated steam from the primary reheater in the boiler flows into the HP cylinder, whereas the second reheated steam from the secondary reheater flows through the IP cylinder in the double reheat unit. By contrast, the main steam from the superheater in the single reheat unit is discharged into the HP cylinder, and the reheated steam flows into the IP cylinder. This process can illustrate that irreversible loss in the HP and IP turbines are lower, the huge exergy loss in the LP cylinders is mainly attributed to moisture loss in the last few levels. Simultaneously, the heat consumption rate in the double reheat USC unit is 7239.6 kJ/ kWh, which is 6.0% lower than that in the traditional USC unit (7726 kJ/ kWh [30]). However, exergy loss is considerably larger, which leads to an unfavorable exergy efficiency. As a VHP with low relative internal efficiency is added to the unit, the mass flow rate of the live steam becomes lower and the flow rate of the reheated steam becomes higher, thereby resulting in lower exergy efficiency. Evidently, exergy loss is larger in the double reheat unit, and steam consumption rate is lower. If the turbine is further optimized, such as by reducing the pressure ratio, and increasing the blade height and the tightness of the VHP, then exergy loss can be reduced and exergy efficiency can be increased. Consequently, the steam consumption rate can be further reduced, which implies that the designed unit has high potential for energy conservation.

3.3 Exergy loss of the condenser

The condenser is a heat exchanger essentially. It recovers condensate water, and its exergy loss comprises the heat exergy released during exhaust condensation. The heat in hot water is transferred to the condensed water in the condenser because the temperature of the latter is extremely low with low-grade thermal energy. Under the existing technical conditions, the low-grade thermal energy is difficult to utilize; hence, the heat exergy released by the condenser is equal to the difference between the inlet and outlet steam through the condenser. Without considering condenser exergy efficiency, heat exergy is calculated using the following formula:

$$I_c = D_{c1}(e_{c1} - e'_c) + D_{c2}(e_{c2} - e'_c) + D_{c3}(e_{c3} - e'_c) + D_{c4}(e_{c4} - e'_c) \quad (8)$$

The exergy loss of the condenser is 20.5 MW based on the calculation, which is considerably less than the exergy loss in the turbine. Ligang W et al. obtained a condenser exergy loss of 39.3 MW in a single reheat unit [30], approximately twice of that in the studied unit. Table 4 shows that the double reheat unit provides lower exhaust pressure and temperature in the LP. Accordingly, exhaust is discharged into the condenser with low temperature and mass flow. Therefore, the heat loss and irreversible loss are reduced during the heat transfer process. This result demonstrates that the use of energy in the condenser with the double reheat unit is more reasonable and ideal.

3.4 Energy analysis of turbine system

For the turbine, the energy balance equation can be defined as following:

$$0 = Q_{in} - Q_{out} - Q_{tur,loss} - W = m_{in}h_{in} - m_{out}h_{out} - Q_{tur,loss} - W \quad (9)$$

$$m_{in} = m_{out} \quad (10)$$

The energy efficiency of turbine is expressed as:

$$\theta = \frac{W}{Q_{in} - Q_{out}} = \frac{W}{m_{in}h_{in} - m_{out}h_{out}} \quad (11)$$

The results of the energetic analysis are shown in Table 5. The results show that the LPs have the highest energy efficiency as the irreversible loss and the quality of energy are ignored, which is different from the results of exergy analysis showing the IP has the highest exergy efficiency.

For the regenerative heaters the energy analysis and energy efficiency can be expressed as:

$$m_{hot} (h_{h,in} - h_{h,out}) = m_{cold} (h_{c,out} - h_{c,in}) + Q_{rh,loss} \quad (12)$$

$$\theta = \frac{m_{cold} (h_{c,out} - h_{c,in})}{m_{hot} (h_{h,in} - h_{h,out})} \quad (13)$$

For the regenerative heaters as adiabatic heat exchange processes, the energy loss can be ignored and the energy efficiency is nearly 100%. But according to the second law of thermodynamics, the exergy loss exists, as shown in Fig.5 and Fig.6, namely different heaters have different exergy efficiency. According to the second law of thermodynamics, in the view of energy-level the applications of this unit are not so reasonable and the heat transfer equipment is not perfect. However the exergy analysis can find out exergy loss and exergy-loss reasons in different parts accurately, providing remarkable insights into the operation and energy-saving optimization of such unit.

For the condenser, the average temperatures of hot and cold fluid are very low, so the exergy loss is just 20.5MW while the energy loss is about 1000MW due to the large mass flow of fluid. Therefore, the exergy analysis is more persuasive by considering not only the quantity but also the quality of energy.

4. Conclusions

The results show that the thermodynamic analysis is a good method to analyze the double reheat ultra-supercritical power plant, and can be generally applicable to different power reheat units. At the same time, it can also provide the fundamental information for the design and

optimization of the double reheat ultra-supercritical power plant. The main conclusions can be drawn as follows:

- 1) In the turbine system, exergy loss in the steam turbine is the largest, approximately more than twice the sum of those in the regenerative heater and the condenser. Therefore, optimizing the exergy loss of the steam turbine will provide the most significant benefits.
- 2) Compared with the single reheat 1000 MW USC turbine, exergy loss in the double reheat turbine is larger, leading to unfavorable exergy efficiency. Reducing exergy loss in the steam turbine should focus on optimizing the LP cylinders and the VHP cylinder, which comprises the majority of exergy loss in the steam turbine.
- 3) The regenerative system in the studied unit is enhanced with two additional extraction points. Thereby, exergy loss, particularly in the recovery heaters locating after the second reheating point, namely in RH2 and RH4 are apparently reduced.
- 4) The heat transfer temperature, mass flow rate and temperature of the hot and cold fluid in the heat exchanging process is the principal factor of exergy loss in a regenerative system. For the studied unit, exergy loss is mainly focused in HRH3, LRH6, LRH7 and LRH10. Exergy loss in different heaters is caused by different reasons.
- 5) Exergy loss in the condenser is reduced by half than that in the single reheat unit, because the double reheat unit allows the exhaust steam from the LP cylinders to have lower parameters, which indicates that adopting a second reheat condenser leads to less irreversible loss and more rational energy use.

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Figure lists:

Fig. 1. Diagram of ultra-supercritical double reheat system

Fig. 2. Process simulation diagram of turbine

Fig. 3. Schematic diagram of the surface heater

Fig. 4. Schematic diagram of the hybrid heater

Fig. 5. Exergy analysis of double reheat and single reheat unit

Fig. 6. Exergy loss analysis of double reheat and single reheat unit

Fig.7. Exergy loss analysis of double reheat unit

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Fig. 5. Exergy analysis of double reheat and single reheat unit

The regenerative heaters in single reheat unit are shown with “*”.

Fig. 6. Exergy loss analysis of double reheat and single reheat unit

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Fig.7. Exergy loss analysis of double reheat unit

Table lists:

Table 1 The main input parameters of turbine

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Table 1

The main input parameters of turbine

Item	Parameter	Value	Unit
VHP	Inlet/outlet steam pressure	310/112.67	bar
	Isentropic efficiency	0.88	—
HP	Inlet/outlet steam pressure	105.4/35.79	bar
	Isentropic efficiency	0.85	—
IP	Inlet/outlet steam pressure	32.14/4.26	bar
	Isentropic efficiency	0.80	—
LP1	Inlet/outlet steam pressure	4.26/0.045	bar
	Isentropic efficiency	0.65	—
LP2	Inlet/outlet steam pressure	4.26/0.045	bar
	Isentropic efficiency	0.65	—
External steam turbine	Outlet steam pressure	0.051	bar
	Isentropic efficiency	0.75	—
Superheated steam	Pressure	310	bar
	Temperature	600	°C
	Mass flow rate	730.4	kg/s
Reheater1	Pressure	105.4	bar
	Temperature	610	°C
Reheater2	Pressure	32.1	bar
	Temperature	610	°C

Table2

Simulation results and actual values of main streams

	Extraction	1	2	3	4	5	6	7	8	9	10
Design values	Mass flow(kg/s)	51.0	56.2	40.5	20.1	39.3/12.5	21.2	31.9	17.2	19.9	21.7
	Temperature(°C) (dryness)	430.3	527.1/309.5	434.0	526.3/309.6	447.0	393.8	312.5	191.3	116.1	0.982
	Pressure(bar)	112.70	62.60	35.80	18.85	11.30	7.82	4.26	1.34	0.62	0.23
	Saturation temperature(°C)	319.5	278.4	243.9	209.4	—	169.5	145.9	108.0	86.8	63.3
Simulation values	Temperature (°C)	430.9	525.9/310.0	434.0	526.3/310.0	445.5	391.8	310.6	191.6	121.1	0.992
	Temperature deviation(°C)	0.6	-1.2/0.5	0	0/0.4	-1.5	-2.0	-1.9	0.3	5.0	0.01
	Degree of superheat(°C)	111.4	247.5/31.6	190.1	316.9/100.6	—	222.3	164.7	83.6	29.3	—

Table 3

Comparison of exergy parameters between double-reheat and single reheat unit

Component s	This unit					Single reheat 1000MW USC unit [30]			
	Exergy cost	Work (MW)	Exergy loss (MW)	Exergy efficiency(%)	Relative internal efficiency(%)	Exergy cost (MW)	Work (MW)	Exergy loss (MW)	Exergy efficiency (%)
	(MW)								
VHP	210.2	188.1	22.1	89.52	94.7	—	—	—	—
HP	232.1	213.5	18.6	91.94	95.2	335.7	316.3	19.4	94
IP	343.9	324.1	19.8	94.24	97.4	443.5	423.9	19.6	95.5
LP1	167.9	143.8	24.1	85.67	98.0	205.8	184.1	21.7	89.5
LP2	170.3	145.2	25.1	85.23	97.7	209	187.3	21.7	89.6
Total	1124.4	1014.7	109.7	90.2	—	1194	1111.6	82.4	93

Table 4

Basic data of double-reheat and single reheat unit

Parameters	This unit	Single reheat 1000MW USC unit [30]
Live steam mass flow rate(kg/s)	730.3	861.6
The first reheat steam mass flow rate (kg/s)	679	715.6
The second reheat steam mass flow rate(kg/s)	582.5	—
Exhaust pressure(bar)	0.045	0.052

Table 5

Results of energy analysis

	VHP	HP	IP	LP1	LP2	total
Energy loss/MW	10.49	10.77	8.4	3.17	3.45	36.28
Efficiency/%	94.7	95.2	97.5	97.8	97.7	--