Performance Analysis of the Technology of High-Temperature Boiler Feed Water to Recover the Waste Heat of Mid–Low-Temperature Flue Gas

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1. INTRODUCTION

The temperature of flue gas discharged from coal-fired power plants in front of the desulphurization tower is 120−140 °C,1 which contains a lot of heat. A lot of heat energy would be lost if the exhaust gas discharged directly, and thus, it will reduce the economic effects. If the original flue gas is transferred by the flue gas heat exchanger2 to recover part of the waste heat and then discharged, the recovered heat is used for the process flow of boiler deaerator make-up water preheating, boiler supply air preheating, or boiler feed water heating.3,4 By this way, both the energy consumption of the boiler operation and the temperature of the inlet flue gas in the desulphurization tower decrease and this helps the full reaction with the limestone slurry to improve the desulphurization efficiency.4 In addition, wet plumes caused by excessive evaporation of limestone slurries could also be avoided to realize the effective utilization of waste heat from the exhaust gas, which exhibits obvious economic and environmental benefits.5

Some scholars started from the technical route of mid−low-temperature waste heat recovery; Kwak et al.8 proposed four technical routes for mid−low-temperature flue gas heat recovery and found that boiler feed water heating is the best choice in summer. Dai et al.9 studied the performance of different working fluids to recover low-grade waste heat by means of the genetic algorithm. However, these studies all ignored the effect of low-temperature corrosion in the system. At the same time, it is also believed that the operating parameters of the flue gas waste heat recovery system (FGWHRS) play a crucial role in heat exchange performance. Liu et al.10 established a flue gas waste heat cascade recovery system experiment bench, proving that flue gas temperature, bypass flue gas ratio, and air temperature are the important factors affecting the recovery of the flue gas residual heat, and suggested that the ratio of the bypass flue gas should be used as the regulating parameter of the system under variable working conditions. Xu et al.11 carried out technical and economic analyses on the heat recovery system using boiler exhaust gas. The research showed that exhaust gas energy is affected not only by the outlet flue gas temperature of the boiler side but also by the comprehensive solution of steam/water side exhaust gas. Liu et al.12 obtained the operating rules of High−Low Energy FGWHRS at an exhaust gas temperature of 85 °C. The results showed that the bypass flue gas flow rate plays a decisive role in improving the heat recovery efficiency, while the circulating water flow rate has little influence on the heat recovery efficiency. There are also many scholars who have carried out innovative design and optimization of the waste

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heat recovery system. Szulc et al.\textsuperscript{13} have established a pilot plant to recover the waste heat of flue gas through condensing the heat exchanger with water at about 40 °C. The experiment showed that the system in this paper has stronger heat recovery efficiency than the conventional waste heat recovery system without a condenser. Wang et al.\textsuperscript{14} studied how to recover waste heat from flue gas before entering flue gas desulfurization (FGD) while heating the condensate by installing a low-pressure economizer (LPE). The analysis results showed that it is feasible to install LPE in an exhaust gas system between the booster fan and the FGD unit with careful consideration of potential low-temperature acid corrosion. Xu et al.\textsuperscript{15} put forward a new conventional waste heat recovery system (WHRS) after in-depth analysis, in which the air preheater is divided into high-temperature and low-temperature air preheaters, and a low-temperature energy saver can be located between the electrodust collector and air preheater for excellent energy-saving performance, thus greatly improving the thermal efficiency of the system and optimizing the heat exchange performance of the system. Wang et al.\textsuperscript{16} analyzed the influence of installing LPE in front of the desulfurization tower on the related instruments and took the second law efficiency and heat balance efficiency as the indexes to evaluate the performance of the LPE system, but this index has little guiding significance for practical engineering, so it can only be a qualitative analysis, not quantitative analysis. Zhang et al.\textsuperscript{17} used two-stage flue gas heat exchangers and arranged them at the upstream of the electrodust collector or desulfurization tower of the 1000 MW unit respectively to study the thermodynamic properties of the unit after putting into operation, and the results showed that the heat consumption of the steam turbine and coal consumption of power supply are reduced significantly, and the temperature of flue gas was guaranteed to be higher than its acid dew point. Meanwhile, the permanence and security of the system should not be ignored. Li et al.\textsuperscript{18} found that when the gas temperature is lower than the acid dew point, it is the most effective method to eliminate the harm of low-temperature corrosion, but the flue gas temperature under this condition is not easy to achieve. Li et al.\textsuperscript{19} studied the variation rule of sulfuric acid corrosion temperature of heat exchange tube wall, and experiments showed that the wall temperature range of sulfuric acid corrosion in the heat exchanger tube was 65–71 °C; thus, the wall temperature of the heating surface should be higher than this range to ensure heat exchangers will work safely. Wei et al.\textsuperscript{20} advised that the flue gas temperature should be higher than 80 °C for the safe operation of heat exchangers. All of the above studies showed that the technology of high-temperature boiler feed water (above 80 °C) to recover the waste heat of mid–low-temperature flue gas is suitable. Nowadays, computational fluid dynamic (CFD) technology is widely used to complete some inconvenient experimental research. Ramos et al.\textsuperscript{21} used CFD modeling and numerical calculation to predict the heat exchange performance of medium–low-temperature flue gas heat exchangers. According to the results, the following conclusions can be drawn: higher temperature and mass flow of flue gas can achieve a higher heat transfer rate and improve heat exchange performance. Han et al.\textsuperscript{22} used CFD software to simulate the deposition process of heat transfer surface on the fin of a flue gas heat exchanger and calculated the transport and condensation characteristics of the multicomponent flue gas. The results showed that the increase of flue gas flow rate, acid vapor, and water vapor would aggravate the low-temperature corrosion of fins, while the increase of flue gas temperature would reduce the risk of low-temperature corrosion.

In previous research studies, most of the working fluids used in mid–low-temperature FGWHRS are low-temperature boiler supply air or condensate water in the flue gas condenser\textsuperscript{23} with the temperature below 30 °C. This will cause low-temperature corrosion of the system\textsuperscript{24} and produce a fouling ash layer.
seriously affecting the performance of the heat exchanger in the system. On the contrary, using high-temperature boiler feed water can greatly reduce its harm. However, there is little thermodynamic analysis and research on the technology of high-temperature boiler feed water recovery of mid–low-temperature flue gas waste heat. At the same time, in the research of flue gas waste heat recovery technology, the research directions lay particular emphasis on the experimental design of the system and basic operation parameters study; most of them do not give specific optimal values for input parameters such as the working fluid temperature and flow rate, which have certain limitations for the engineering application.

Therefore, in this study, an experimental apparatus was set up to study the influence of different performance parameters on the heat transfer performance of the waste heat recovery system using high-temperature boiler feed water to recover the waste heat of mid–low-temperature flue gas at the inlet of the desulfurization tower of a 330 MW generator unit in Xinjiang. Moreover, based on the operation parameters of the power plant, the gas–liquid temperature ratio and gas–liquid flow ratio are proposed to further analyze the heat transfer performance of the system, and through the curve fitting method, the optimal water temperature and water flow rate of mid–low-temperature FGWHRs at the upstream of the desulfurization tower for heating boiler feed water under certain working conditions are found, which provides a theoretical basis and engineering guidance for determining the optimal waste heat recovery scheme of the mid–low-temperature flue gas at the upstream of the desulfurization tower for heating boiler feed water before entering the desulfurization tower.

2. DESIGN OF MID–LOW-TEMPERATURE FGWHRS

2.1. System Processing. A prototype experimental apparatus for flue gas waste heat recovery at mid–low-temperature was independently designed and built based on the actual system on a 330 MW generator unit in Xinjiang. The experimental bench flow chart of flue gas waste heat recovery is shown in Figure 1. The original flue at downstream of the electrostatic precipitator (ESP) and at upstream of the desulfurization tower is connected to the flue bypass, and a certain amount of flue gas enters the test device. The system is composed of three small systems: a flue gas system, a circulating water system, and a steam soot blowing system. The flue gas from the original flue is let out of the bypass flue by the induced draft fan, after a two-stage staggered finned tube bundle heat exchanger, the flowing water in the finned tube is heated, and the temperature of the flue gas also decreases and finally returns to the absorption tower to complete the circulation of flue gas in the waste heat recovery system. The circulating water of the water system flows into the buffer tank from the desalted water pipe then flows into the flue gas heat exchanger through the inlet pipe, and finally, flows into the outlet pipe after being heated by the flue gas in the heat exchanger. A part of the water flows into the buffer tank for circulation and a part of the water flows into the air cooling heat exchanger for circulating water cooling and then flows into the buffer tank, while the proportion of the water flow rate in the two directions is regulated by the stop valve on each water path, which can not only regulate the temperature of circulating water but also prevent the water loss caused by steam escaping due to the vaporization of circulating water to complete the circulation of circulating water in the waste heat recovery system. The experimental bench also installed a steam soot blowing device in the flue gas inlet section to prevent the deposition of ash and scale on the flue gas heat exchanger tube wall and improve the uniformity of the flue gas by rectifying the flue gas introduced by the bypass flue, which is more conducive to the efficient recovery of flue gas heat and helps to improve the heat exchange efficiency of the flue gas heat exchanger as well as the heat exchange performance of the system.

According to the above process, an experimental bench was built on a 300 MW power plant generator unit in Xinjiang, and relevant experiments were completed. Figure 2 shows the scene photo of the experiment. In the experimental bench, the diameter of the circular flue is 500 mm. The flue gas-induced draft fan is a centrifugal fan, and the air volume regulation range is 6000–11 000 m³/h. The two-stage heat exchanger is a staggered finned tube bundle heat exchanger, and the fins are ring fins. Both in the flue gas side and water side, the inlet and outlet are, respectively, equipped with temperature sensors, and the outlet of the water side of the heat exchanger is equipped with electromagnetic flow sensors. The circulating water in the first and second stage heat exchangers indirectly

Figure 2. Experimental apparatus of mid–low-temperature FGWHRs.
exchanges heat with the original flue gas successively. The specific geometric dimension parameters of the heat exchanger are listed in Table 1; the diameter of the circulating water pipe is 10 mm; the air-cooled heat exchanger is also a bundle by a temperature sensor and an electromagnetic pressure gauge EJA430E-JAS5J-912EA (0−600 kPa) 0.5.

Table 1. Dimension Parameters of the Staggered Tube Bundle Heat Exchanger

| parameter                        | unit | value |
|----------------------------------|------|-------|
| height of the heat exchanger     | mm   | 500   |
| width of the heat exchanger      | mm   | 1000  |
| length of the heat exchanger     | mm   | 320   |
| wall thickness of the tube bundle| mm   | 1     |
| tube bundle diameter             | mm   | 25    |
| heat exchange tube bundle        |      | 24    |
| number of single-tube fins       |      | 180   |
| fin pitches                      | mm   | 12.5  |
| fin spacing”c”                   | mm   | 5     |
| fin thickness                    | mm   | 1     |

is 10 mm; the circulating water pump is a horizontal centrifugal pump with a flow regulation range of 3−10 m³/h and a lift of 10 m; the air-cooled heat exchanger is also a finned tube heat exchanger, and the adjustable frequency of the fan on the heat exchanger is 15−25 Hz. The inlet and outlet temperatures of the flue gas and circulating water are directly read out in the central control room by a temperature sensor and an electromagnetic flow sensor. The above-measured data were transmitted to an IPC-610h industrial computer in the control room through an X108 industrial switch for subsequent processing. A data acquisition system based on WinCC software was established, and Origin software was used subsequently to process the data. By changing the frequency of the frequency converter of the induced draft fan, the flue gas flow rate can be controlled, and the data could be manually read by a Laoying 3012H-c automatic smoke (gas) tester; the pressure drop of the heat exchanger would be read by the pressure gauge, and the fine adjustment rate of the air cooling circulating water would be controlled by the fan frequency of the frequency converter of the induced draft fan in the central control room. The inlet temperature of circulating water is roughly adjusted by controlling the opening of the pipe valve flowing through the air-cooled heat exchanger and finely adjusted by changing the fan frequency of the axial flow fan on the air-cooled heat exchanger in the central control room. The parameters measured in the experiment include temperature, velocity, flow rate, and pressure.

For this waste heat recovery system, the heat exchange of the flue gas side and circulating water side is directly related to the effect of waste heat recovery. Therefore, flue gas side heat transfer quantities and water side heat transfer quantities are also used as key performance parameters of the system. In this experiment, the gas side temperature of the system is above the acid dew point of the flue gas, and the water side temperature is below the evaporation temperature; as a result, the phase change heat transfer quantity of the system is small, so the latent heat could be ignored, and the calculation is carried out using formulas 1 and 2, respectively.

\[
Q_1 = c_{p,g} \rho_g A_g v_g (T_{in,g} - T_{out,g})
\]

\[
Q_2 = c_{p,w} A_{in,w} (T_{in,w} - T_{out,w})
\]

where \(c_{p,g}\) and \(c_{p,w}\) are the specific heat capacities of flue gas and circulating water at constant pressure, which are 1.082 and 4.208 kJ/(kg°C), respectively, \(\rho_g\) is the density of flue gas, taken as 0.849 kg/m³, \(A_g\) is the flow cross-sectional area of flue gas, taken as 0.5 m², \(v_g\) is the velocity of flue gas, \(q_{in,w}\) is the flow rate of circulating water, \(T_{in,g}\) and \(T_{out,g}\) are the temperatures of flue gas inlet and outlet, respectively, and \(T_{in,w}\) and \(T_{out,w}\) are the temperatures of the water outlet and inlet, respectively.

According to the principle of the waste heat recovery system, the recovered flue gas waste heat is finally used to save steam to heat the deaerator heat source and boiler feed water to improve the economy and work efficiency of the unit. The amount of steam saved increases with an increase of waste heat recovery. Therefore, this paper defines a gas−water heat recovery ratio \(\eta\) as follows to evaluate the waste heat recovery.

\[
\eta = \frac{\xi Q_1}{Q_2} \times 100\%
\]

where \(Q_1\) and \(Q_2\) are determined by eqs 1 and 2, respectively, and \(\xi\) is the correction coefficient of the heat exchanger, whose value range in this experiment is 0.91−1.11.

2.2. Performance Parameters. Under the variable working conditions of the waste heat recovery unit, the operating parameters change with the varying parameters of different working conditions. Too low exhaust temperatures may lead to low-temperature corrosion and fouling of the flue gas heat exchanger, and too high exhaust temperatures would reduce the efficiency of waste heat recovery. The flue gas temperature is controlled by the operating load of the power plant, generally 120−140 °C, and the circulating water flow rate is controlled by the opening of each valve in the pipeline. The flue gas velocity is regulated by the frequency of the induced draft fan controlled by the frequency converter of the induced draft fan in the central control room.

Table 2. Experimental Measuring Instruments and Precision

| measuring instruments           | model                     | accuracy  |
|---------------------------------|---------------------------|-----------|
| flue gas velocimeter            | Laoying 3012H-c automatic smoke test | ±3%       |
| temperature sensor              | WZP-431 (0−300 °C)        | 0.1 °C    |
| electromagnetic flow sensor     | WP-LUA(50)11-CN1201SW (40 m³/h) | 0.2       |
| pressure gauge                  | EJA430E-JAS5J-912EA (0−600 kPa) | 0.5       |

At the same time, the thermodynamic performance and thermal efficiency of the heat exchanger also have a non-negligible impact on the recovery efficiency of the waste heat recovery system. As a result, the total heat transfer coefficient \(K\) has been regarded as one of the standards to judge the heat recovery effect of the waste heat recovery system. In this experiment, the calculation of the total heat transfer coefficient only involved the convective heat transfer coefficient inside and outside the tube without considering the influence cast by fouling thermal resistance and pipe wall thermal resistance. Since the heat exchange area of the heat exchanger is known, the total heat transfer coefficient of the heat exchanger is expressed by formula 4.

\[
K = \frac{Q_2}{A \Delta T_{in}}
\]
where $Q_{2}$ is the heat absorption on the water side, that is, the actual heat recovered by the waste heat recovery system. According to formula 2, $A$ is the heat exchange area of the heat exchanger, which is known to be $6.28$ m$^2$ in this experiment, and $\Delta t_{m}$ is the logarithmic mean temperature difference of the heat exchanger. The heat exchanger adopts the crossflow mode for heat exchange. Crossflow heat exchange refers to the heat transfer process in which the flow directions of two kinds of fluids are vertically crossed. In engineering calculations, if the number of twists and turns exceeds four times, it can be regarded as pure countercurrent. The calculation formula of the logarithm mean temperature difference $\Delta t_{m}$ could be written as follows

$$\Delta t_{m} = \frac{\Delta t_{\text{max}} - \Delta t_{\text{min}}}{\ln \frac{\Delta t_{\text{max}}}{\Delta t_{\text{min}}}}$$  \hspace{1cm} (5)

where $\Delta t_{\text{max}}$ and $\Delta t_{\text{min}}$ represent the temperature difference between the flue gas inlet temperature $T_{\text{in,g}}$ and the circulating water inlet temperature $T_{\text{in,w}}$ and the temperature difference between the flue gas outlet temperature $T_{\text{out,g}}$ and the circulating water inlet temperature $T_{\text{in,w}}$, respectively.

The gas–liquid flow ratio and gas–liquid temperature ratio are also proposed as two influencing factors to measure the thermal efficiency of the waste heat recovery system, and the gas–liquid flow ratio and gas–liquid temperature ratio are defined as $\kappa_1$ and $\kappa_2$, respectively. The gas–liquid flow ratio $\kappa_1$ and gas–liquid temperature ratio $\kappa_2$ of the waste heat recovery system are respectively expressed by formulas 6 and 7

$$\kappa_1 = \frac{\rho_{g} A_{V_{g}}}{q_{\text{in,w}}}$$  \hspace{1cm} (6)

$$\kappa_2 = \frac{\Delta T_{\text{in,g}}}{\Delta T_{\text{in,w}}}$$  \hspace{1cm} (7)

The formula for selecting acid dew point temperature of flue gas in the system is based on the standard method for thermal calculation of boiler units\cite{27} published by the former Soviet Union in 1973 combined with the pilot waste heat recovery system, which is expressed by formula 8

$$t_{A,d} = \frac{\beta_{v} S_{\text{ar},z_s}}{1.05 \alpha_{\text{fl}} A_{\text{ar},z_s}} + t_{w,d}$$  \hspace{1cm} (8)

where $t_{A,d}$ is the acid dew point of flue gas, $\beta$ is a constant related to excess air coefficient on the flue gas side in front of the desulfurization tower, $S_{\text{ar},z_s}$ is the fuel conversion sulfur, $\alpha_{\text{fl}}$ is the fly ash share, $A_{\text{ar},z_s}$ is the fuel conversion ash, and $t_{w,d}$ is the water dew point for flue gas.

3. RESULTS AND DISCUSSION

3.1. Effect of Flow on the Heat Transfer Performance.  
3.1.1. Influence of Circulating Water Flow Rate on the Heat Transfer Performance. The flue gas temperature was set at 135 °C, and the flue gas flow rate was set at 5 m/s. When the temperature of circulating water was 90 °C, the relationship between the circulating water flow rate and the heat transfer performance parameters of the waste heat recovery system in this experiment was studied with the circulating water flow rate varying in the range of 5–9 m$^3$/h.

As shown in Figure 3, with an increase of the circulating water flow rate in the waste heat recovery system, the temperature difference between the inlet and outlet of flue gas gradually increases, and the increasing trend gradually slows down. However, the temperature difference of circulating water decreases with an increase of the circulating water flow rate, while the decreasing trend slows down gradually. At the same time, $\eta$ and $K$ also decrease with the increase of circulating water flow rate, and the downward trend of the two curves appears to basically slow down due to the fact that when the parameters of the flue gas side remain unchanged, the heat release of the flue gas side is basically stable. When the heat release is constant, the lower the water flow rate, the higher the flue gas temperature difference. Meanwhile, as the circulating water volume increases, the flue gas temperature decreases, so the flue gas density decreases slightly and thus the heat exchange on the flue gas side decreases slightly, resulting in a
slow down in the decrease of the circulating water temperature difference. Therefore, the increasing rate of water temperature difference is greater than the increasing rate of the flue gas temperature difference, which are 48 and 11%, respectively, so η also decreases with an increase of the circulating water flow rate. K decreases with an increase of the circulating water flow rate and the logarithm mean temperature difference.

Therefore, increasing the circulating water flow rate may cause the reduction of η, and as a subsequent result, the efficiency of the heat exchanger would be reduced as well as the heat exchange performance of the waste heat recovery, which is not conducive to the efficient and economical operation of the waste heat recovery system. So, the water flow rate of about 5 m³/h is used as the circulating water flow rate of the waste heat recovery system.

3.1.2. Influence of Flue Gas Velocity on the Heat Transfer Performance and System Resistance. The circulating water temperature was set at 90 °C, the circulating water flow rate was set at 9 m³/h, and the flue gas temperature was controlled at 135 °C. The relationship between the circulating water temperature and the heat transfer performance parameters of the waste heat recovery system in this experiment was studied when the flue gas velocity was controlled to change within the range of 3.35–5.21 m/s.

As shown in Figure 4, with the gradual increase of the flue gas velocity, the temperature difference between the inlet and outlet of flue gas gradually decreases by 14.7%, while the temperature of circulating water gradually increases by 81.25% with an increase of flue gas velocity, and the variation range is larger than that of the temperature difference between the inlet and outlet of flue gas. At the same time, with an increase of flue gas velocity, η and K gradually increase. According to the basic heat transfer equation, the temperature difference of circulating water also increases; it can be explained that the increase of the flue gas flow rate would shorten the heat exchange time on the gas side of the heat exchanger and reduce the heat transfer slightly. Therefore, the temperature difference between the inlet and outlet of flue gas would decrease slightly. With an increase of the flue gas flow rate, the uniformity of the flow field of the heat exchanger is strengthened relatively, the utilization rate of the heat exchange area is greater, η and K become higher, and the heat transfer performance of FGWHRs becomes greater. It can be seen that a high flue gas flow rate is more conducive to the water side heat recovery of the waste heat recovery system.

At the same time, the temperature of circulating water was set at 85 °C and the flow rate of the circulating water was set at 4 m³/h when the flue gas temperature was controlled at about 130 °C. The relationship between the flue gas flow rate and the inlet and outlet pressure drop of the heat exchanger in the system was studied when the flue gas velocity was controlled in the range of 3.5–6.16 m/s.

As illustrated in Figure 5, with the gradual increase of flue gas velocity, the pressure drop at the inlet and outlet of the heat exchanger increases gradually, and the increase is much larger. It can be seen that the resistance of the heat exchanger increases with the flue gas velocity, and the increasing speed becomes sharper as the flue gas velocity increases. This
experimental result can be explained by Bernoulli’s principle. Therefore, the increase of flue gas velocity would improve the heat transfer performance of the system but also lead to the increase of system resistance. As a result, it is necessary to take a reasonable value of increase of system resistance. As a result, it is necessary to take a reasonable value of flue gas velocity. Combined with the above research in this paper, a flue gas velocity of 4–5 m/s is more suitable to the high heat transfer performance and low resistance operation of the system.

3.1.3. Influence of Gas–Liquid Flow Ratio on the Heat Transfer Performance. The influence of gas–liquid flow ratio on the heat exchange performance of the waste heat recovery system is studied when the flow rate of flue gas is 9000–11 000 m$^3$/h.

As displayed in Figure 6, when increasing the gas–liquid flow ratio of the three types of flue gas flow rates, the gas–water heat recovery ratio first increases slowly and then increases sharply, and then the growth trend gradually slows down and finally stabilizes slowly, approaching the limit of $\eta$ under the working condition of the FGWHRS. This trend occurs due to the fact that at low gas–liquid flow, the water flow rate is relatively large, the flow resistance in the pipe is large, and the heat exchange time is short, so $\eta$ is small and the growth rate is slow; at the medium gas–liquid flow rate, with the gradual decrease of the water flow rate, the main influence on $\eta$ is cast by the circulating water flow rate. According to the previous analysis, $\eta$ increases with both the decrease of the circulating water flow rate and the increase of flue gas velocity. Therefore, $\eta$ increases greatly at the middle gas–liquid flow. However, with the increase of the gas–liquid flow ratio, the heat exchanger efficiency gradually approaches the best state, the heat transfer performance of the waste heat recovery system would reach the limit, and $\eta$ increases slowly and tends to balance.

When the flue gas flow rate is 9000 m$^3$/h, the gas–water heat recovery ratio increases slowly by 2% when the gas–liquid flow ratio is below 1100. When the gas–liquid flow ratio was between 1100 and 1800, $\eta$ increases sharply by 58% with an increase of $\eta$. When the gas–liquid flow ratio was above 1800, $\eta$ slows down again with an increase of the gas–liquid flow ratio, slowly increases by 3.2% and tends to balance. When the flue gas flow rate was 10 000 m$^3$/h, $\eta$ increases slowly by 2% with an increase of the gas–liquid flow ratio below 1333. When the gas–liquid flow ratio is between 1333 and 1900, $\eta$ increases sharply with an increase of the gas–liquid flow ratio, from 0.52 to 0.81. When the gas–liquid flow ratio is above 1900, $\eta$ slows down again with an increase of the gas–liquid flow ratio, slowly increases by 3.7% and tends to balance. When the flue gas flow rate is 11 000 m$^3$/h, $\eta$ increases slowly by 4.8% when the gas–liquid flow ratio is below 1466. When the gas–liquid flow ratio was between 1466 and 2076, $\eta$ increases sharply by 59.6% with an increase of the gas–liquid flow ratio. When the gas–liquid flow ratio is above 2076, $\eta$ slows down again with an increase of gas–liquid flow ratio, slowly increases by 2.3% and tends to balance.

It can be seen that when the gas–liquid flow ratio is low and the gas–liquid flow ratio was high, $\eta$ changes slightly with the gas–liquid flow ratio, but when the gas–liquid flow ratio is medium, $\eta$ changes greatly with the gas–liquid flow ratio, so there exist two critical points where the increase rate of $\eta$ changes sharply. At the same time, it can be seen that, with an increase in flue gas flow rate, the increase rate of $\eta$ gradually increases and the final stable $\eta$ value decreases, and 2076, 1900, and 1800 are the three critical points of gas–liquid flow ratio when $\eta$ reaches a stable maximum under the three flue gas flow rates, and these provide a reference for selecting optimal circulating water flow rate afterward.

Therefore, with an increase of the gas–liquid flow ratio, $\eta$ of the waste heat recovery system increases and the increasing rate first increases and then decreases. The larger the flue gas flow rate, the larger the $\eta$ and the better the heat transfer performance when the system tends to balance. Meanwhile, for the economic operation of the system, the gas–liquid flow ratio should be selected as the reference value of the optimal gas–liquid flow ratio at the critical point where the rate of increase of gas–water heat recovery drops sharply. By optimizing the flow ratio and combining it with the actual situation of the flue gas flow rate, the optimal circulating water flow rate under different conditions of the flue gas flow rate in the system is determined, and then, the relationship between the flue gas flow rate $q_{fl}$ and the optimal circulating water flow rate $q_{sw}$ is obtained through the curve-fitting application program of MATLAB. In the flue gas flow rate range of 6000–11 000 m$^3$/h, the formula can be expressed as follows

$$q_{sw} = 5.393 - \frac{0.769}{1 + \left( \frac{q_{fl}}{7837.28} \right)^{5.727}}$$

(9)

By drawing the curve fitting in Figure 7, we can see that in this system, the calculation value of the optimal circulating water flow rate (through formula 9) and the experimental value can be found. It can be seen that the calculated values fit the experimental data well (the correlation coefficient is 0.997).

3.2. Effect of Temperature on the Heat Transfer Performance. 3.2.1. Influence of the Circulating Water Temperature on the Heat Transfer Performance. When the flue gas temperature was set at 135 °C, the flue gas flow rate was set at 5.5 m/s, and the circulating water flow rate is controlled at 5 m$^3$/h; the relationship between the circulating water temperature and the heat transfer performance parameters of the waste heat recovery system in this experiment was studied when the circulating water temperature varied in the range of 84–94 °C.

It can be seen from Figure 8 that with an increase of the circulating water temperature, the temperature difference between the inlet and outlet of flue gas basically presents a
linearly decreasing trend, while the decreasing trend of the temperature difference between the inlet and outlet of circulating water gradually increases; $\eta$ and $K$ are basically the same with the change of the circulating water temperature. When the temperature of circulating water is about $84-88^\circ C$, they basically remain at a high level unchanged, while the critical point of the temperature of circulating water is $88^\circ C$. Then, with an increase of water temperature, $\eta$ and $K$ values gradually decrease, and the decline rate becomes much faster. It can be seen that when the circulating water temperature is about $84-88^\circ C$, the heat exchange performance of the heat exchanger is the best, and $\eta$ is as high as 0.9; almost all of the heat released from the flue gas side is absorbed by the circulating water. It can be explained that with the increase of the temperature of the circulating water, the circulating water is much close to the saturated state after being heated by flue gas, which makes it more difficult to heat up, leading to much lower heating efficiency. At the same time, a part of the high-temperature water has been vaporized into water vapor, absorbing a part of the latent heat of vaporization and reducing $\eta$. It can be seen from Figure 8 that when the temperature of the circulating water increases by about 11%, $\eta$ decreases by about 0.3. As it is predictable, the temperature of the circulating water has a great influence on the heat transfer performance of the waste heat recovery system, and the relatively low temperature of about $88^\circ C$ is more conducive to the operation of the waste heat recovery system.

3.2.2. Influence of the Gas−Liquid Temperature Ratio on the Heat Transfer Performance. The circulating water flow rate was controlled at 5 m$^3$/h, and the flue gas flow rate was controlled at 5 m/s. The influence of the gas−liquid temperature ratio on the heat transfer performance of the waste heat recovery system was studied when the flue gas temperature was 125, 130, and 135°C, respectively.

As shown in Figure 9, with an increase of the gas−liquid temperature ratio, $\eta$ values of the three flue gas temperatures

![Figure 7. Optimal circulating water flow rate at different flue gas flow rates.](image)

![Figure 8. Effect of circulating water temperature on the heat transfer performance: (a) relationship between the circulating water temperature and the temperature difference between the inlet and outlet of flue gas and water and (b) relationship between the circulating water temperature and $\eta$ and $K$ of the heat exchanger.](image)

![Figure 9. Relationship between $\kappa_2$ and $\eta$.](image)
all increase sharply at first, then slow down, and finally tend to be stable. When the gas–liquid temperature ratio gradually increases, the inlet temperature of the circulating water gradually decreases from 94 °C, and it is much difficult for the system to reach a saturated state. From the previous analysis, it can be seen that η would increase gradually, and with approaching the optimal state of heat exchanger efficiency, the growth tendency would gradually slow down and eventually tend to the optimal state of heat exchange performance. The increase rate of the gas–water heat recovery ratio increases with an increase of the flue gas temperature, and the higher the flue gas temperature, the higher the eventual η obtained and the larger the gas–liquid temperature ratio at the sudden drop critical point of the increase rate of η, whose values are 1.49, 1.5, and 1.53, respectively. This can be explained by the fact that an increase of the flue gas temperature leads to an increase of the water saturation pressure and the threshold of reaching the saturation state. Therefore, the change range of water temperature difference is larger, the heat exchange performance is stronger, and the critical η is larger.

Results have revealed that the gas–liquid temperature ratio has a great influence on the heat transfer performance of FGWHRs. A higher gas–liquid temperature ratio and a higher gas–water heat recovery ratio are more conducive to the high heat transfer performance of the system, and the higher the flue gas temperature, the greater the gas–water heat transfer ratio when the system tends to balance and the better the heat transfer performance of the system. At the same time, for the economic operation of the system, the gas–liquid temperature ratio at the critical point where the increase rate of η suddenly drops should be selected as the reference value of the optimal κ of the system to find the optimal value of the circulating water temperature of the system through the optimal gas–liquid temperature ratio. Combined with the flue gas temperature under the actual working conditions, the optimal circulating water temperature under different flue gas temperature conditions can be determined, and the relationship between the flue gas temperature and the optimal circulating water temperature is obtained through the curve-fitting application program of MATLAB

$$T_{in,w} = -0.2647T_{in,g}^2 + 7.2972T_{in,g} - 415.74$$

(10)

As depicted in Figure 10, the curve-fitting result indicates that in this waste heat recovery system, the flue gas temperature and the optimal circulating water temperature are polynomial, and the comparison between the calculated value of the optimal circulating water temperature (through eq 10) and the experimental value can be found, where the calculation results compared fairly well with the experimental data (the correlation coefficient is 0.99971).

3.3. Analysis of Low-Temperature Corrosion of the System. Combined with the calculation and analysis of the coal used in the power plant, the local meteorological conditions, and formula 10, the acid dew point of the flue gas of the system can be calculated, and the calculation results are shown in Table 3.

At the same time, when the flue gas system was at a low flue gas flow rate (9000 m³/h) and low flue gas temperature (125 °C) (not conducive to the heat exchange tube wall temperature higher than the acid dew point of the flue gas) the operation, the temperature variation of the heat exchange tube wall temperature under different circulating water flow rates and circulating water temperatures was studied and then the influence of low-temperature corrosion on the system was further analyzed.

Based on Figure 11, it can be stated that the wall temperature of the heat exchange tube not only decreases with an increase of the circulating water flow rate but also increases with an increase of the circulating water temperature and shows a linear trend. Obviously, under the experimental conditions, the temperature of the heat exchange tube wall is higher than the acid dew point temperature of 95.7 °C; thus, the dilute sulfuric acid steam in the flue gas will not condense and adhere to the wall surface to produce acid corrosion, which

**Figure 10.** Optimal circulating water temperature at different flue gas temperatures.

**Table 3. Calculation of Acid Dew Point Temperatures of the Flue Gas in the System**

| Project | received sulfur/% | ash received/% | low caloric value/(kJ/kg) | water dew point of flue gas/°C | acid dew point of flue gas/°C |
|---------|-----------------|---------------|-------------------------|-------------------------------|-----------------------------|
| coal    | 0.8             | 17.6          | 15 130                  | 48.3                          | 95.7                        |

**Figure 11.** Relationship between $T_{in,w}$ and wall temperature of heat exchange tube at different circulating water flow rates.
means that low-temperature corrosion of the flue gas heat exchanger would not occur, and it is difficult to produce ash deposition and scaling on the tube wall of the heat exchanger. It also highlights the advantages of mid−low-temperature flue gas waste heat used in boiler feed water heating technology. At the same time, when the circulating water flow rate is 9 m²/h (the most unfavorable condition for the heat exchange tube wall temperature to be higher than the acid dew point of the flue gas), the relationship curve between the circulating water temperature and the heat exchange tube wall temperature can be expressed as eq 11

\[
T_c = 0.5472 + 58.01422T_{in, w}
\]  

(11)

where \(T_c\) is the pipe wall temperature, and it can be seen from the above formula that when the circulating water temperature decreases below 68.87 °C, the heat exchange tube wall temperature will be lower than the flue gas acid dew point temperature of the system device; that is, when the inlet water temperature is controlled above 69 °C, the waste heat recovery system device will not be affected by low-temperature corrosion, and this critical temperature is within the starting low-temperature corrosion temperature range obtained by Li. Thus, the long-term use and safety of the device will be ensured.

4. CONCLUSIONS

In this paper, a thermodynamic analysis of the technology of recovering the waste heat of mid−low-temperature flue gas by high-temperature boiler feed water (above 80 °C) is carried out. The main conclusions are as follows:

(1) Increasing the flow rate and temperature of the circulating water will reduce the gas−water heat recovery ratio and the heat transfer performance of the system, while increasing the flow rate of the flue gas will increase the gas−water heat recovery ratio. A high flue gas flow rate is more conducive to the water side heat transfer of the waste heat recovery system and improves the heat transfer performance of the system. However, the increase of the flue gas flow rate will lead to an increase of pressure drop at the inlet and outlet of the heat exchanger and system resistance.

(2) An increase in the gas−liquid flow ratio brings an improved gas−water heat recovery ratio of the system, as well as the heat transfer efficiency. The rate of increase is observed to first increase and then decrease and finally approaches the best heat transfer performance of the system. For a flue gas flow rate range of 6000−11 000 m³/h, the relationship between the flue gas flow rate and the optimal circulating water flow rate is obtained (Formula 9).

(3) As the gas−liquid temperature ratio increases, the gas−water heat recovery ratio of the FGWHRs increases, and the increase rate remains constant at first, then starts to decrease, and finally, will reach the best heat transfer performance of the system. Meanwhile, the higher the flue gas temperature, the greater the maximum value of the gas−water heat recovery ratio and the better the peak heat exchange performance. For a flue gas temperature range of 125−135 °C, the relationship between the flue gas temperature and the optimal circulating water temperature is also given (Formula 10).

(4) Based on our measurements, the relationship curve between the circulating water temperature and the wall temperature of the heat exchange tube is obtained. The curves suggest that when the inlet water temperature is controlled above 69 °C, the FGWHRs will be protected from the harm of low-temperature corrosion.

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Notes

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