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Design Optimization of Boiler Tail Flue in Supercritical Carbon Dioxide Power Generation System

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Abstract:
The S-CO_2 top-bottom combined cycle based on overlap energy utilization can lead to excessive heating area, due to the small temperature difference and the large thermal load for the heating surface at the tail of the boiler. Therefore, reasonable optimization indexes are needed for design optimization. Common optimization indexes include heating area and working medium pressure drop, but lower working medium pressure drop usually leads to large heating area, for example, with the increase of tube inner diameter or boiler width, the pressure drop decreases but the heating area increases. Thus, if both are used as optimization indexes, it will be difficult to choose the optimum tube inner diameter and boiler width. In this paper, exergy loss analysis is used, in combination with economic analysis, the optimization index is unified to the cost per unit heat transfer of the heating surface. The thermal calculation and pressure drop calculation models are established for the heating surface at the tail of the boiler. The optimized heating surface can greatly improve the economic benefit.
Keywords: *S-CO_2 cycle; calculation model; exergy loss analysis; optimization of the boiler*

1 Introduction

In supercritical carbon dioxide power generation system, the working medium in the cycle is CO_2 in supercritical state, and the cycle is called S-CO_2 cycle. The advantage of S-CO_2 cycle is high thermal efficiency and compact system. With the need to reduce carbon emission, the electric power grid needs to adapt the renewable energy generated power such as wind and solar energy. Due to the unstable nature of wind and solar energy, the fossil fuel power generation systems need to respond quickly to the load variation to avoid wasting of wind or solar energy. The S-CO_2 coal-fired power generation system has fewer components and more compact system, which provides possibility for rapid load variation. Therefore, S-CO_2 cycles are more advantageous than steam Rankine cycles [1].

Efforts have been made to study the boilers in S-CO_2 power cycles. Bai et al. [2] carried out the conceptual design of the boiler in a 300 MW coal-fired power plant with single reheat recompression S-CO_2 Brayton cycle. To utilize the waste heat of flue gas at the tail of boiler, split heater is used. Zhou et al. [3] put forward the calculation method for coal-fired boiler for S-CO_2 cycle, and used this calculation method to optimize the design and calculation of 1000 MW S-CO_2 coal-fired system with main parameters of 30.70 MPa and 606 ℃. The calculation shows that in terms of boiler heat transfer, part of the radiative heat transfer of S-CO_2 boiler becomes convective heat transfer. To solve the problems of large pressure drop of working medium and low temperature of reheating working medium under partial load conditions, Tong et al. [4] proposed a new layout scheme of heating surface of a 300 MW S-CO_2 coal-fired boiler, and studied its performance under load variation. The new S-CO_2 boiler is a single boiler with double tangential circle arrangement without central wall, which increases the proportion of reheat radiant heating surface.

Due to the small specific enthalpy increase of CO_2 in boiler, the mass flow rate of working medium in S-CO_2 boiler is 8 times that of steam Rankine cycle system, which causes larger pressure drop. Xu et al. [5] proposed the partial flow strategy for pressure drop reduction of boiler, and carried out modular design of the boiler heating surface. The working fluid is divided into two steams before entering the boiler and flows through different heating surfaces. Compared to the traditional water steam system, in the partial flow mode, the heating surface of the boiler tail is significantly different, and the inlet temperature and heat transfer characteristics of S-CO_2 in partial flow mode are also different. Because the inlet temperature of CO_2 is high, the temperature

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difference between flue gas and working medium is small, which will result in a large heating surface area. Therefore, it is necessary to optimize the design of the heating surface in tail flue of the boiler. The objectives of the optimization is to reduce the area of the heating surface as much as possible and to ensure that the pressure drop of boiler working medium and flue gas is within a reasonable range. There is a trade-off between these two objectives, since increasing the flue gas flow rate can increase the heat transfer coefficient and reduce the heating area, but will increase the resistance on the flue gas side.

In this work, for the boiler of a 1000 MW coal-fired S-CO_2 cycle based on overlap energy utilization, this paper established the model for the thermal calculation of the heating surface in tail flue of the boiler and the model for calculating the working medium pressure drop on both sides of the heating surface. Combined with the minimum entropy production theory and the method of exergy loss analysis, the heating surface is optimized to minimize the cost per unit heat transfer. This work provides a reference for the optimal design of the heating surface in tail flue of S-CO_2 boiler.

2 Calculation Model and Methodology

S-CO_2 unit is quite different from water unit in terms of working medium thermophysical properties and working medium flow. At the same time, the diversion mode also brings changes to the heating surface, the most obvious of which is the change in flue gas pressure drop and heating area. Therefore, it is necessary to design and study the boiler tail heating surface. In order to explore the characteristics of the heating surface of a large capacity system, this work establishes the calculation model for the heating surface in tail flue of a 1000 MW coal-fired S-CO_2 boiler with overlap energy utilization and top-bottom combined cycle [6]. In Figure 1, the heating surface is represented by Heater 4a.

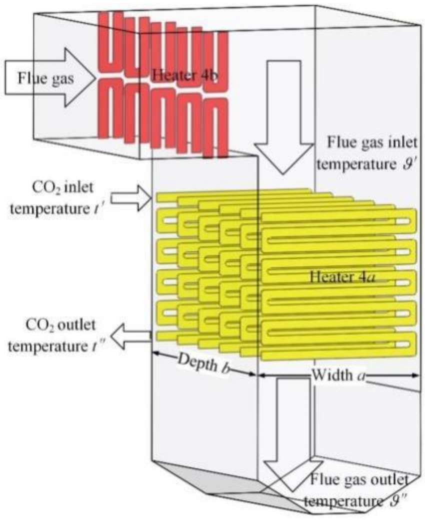

Figure 1 Heating surface in tail flue of an S-CO_2 cycle based on overlap energy utilization

Table 1 Prime table

Nomenclature			
t'	CO_2 inlet temperature	λ	Thermal conductivity
t''	CO_2 outlet temperature	ν	Viscosity
θ'	Flue gas inlet temperature	C_s	Correction factor for tube bundle arrangement
θ''	Flue gas outlet temperature	C_z	Correction factor of tube rows in flue pass
b	Depth	ε	The ash fouling coefficient
a	Width	Ψ	Thermal effective coefficient
Q	Heat absorbed	Pr	Prandtl number
D	Mass flow rate of the working fluid	α_2	Surface heat transfer coefficient
i'	Fluid enthalpy at the inlet	K	Heat transfer coefficient
i''	Fluid enthalpy at the outlet	H	Heating area
B_j	Calculate fuel consumption	Q^{tr}	Heat transfer calculated by heat transfer equation
φ	Thermal efficiency	Re	Reynolds
I'	Flue gas enthalpy at the inlet	ΔP	Pressure drop
I''	Flue gas enthalpy at the outlet	ζ	Friction factor
$\Delta\alpha$	Air leakage coefficient	ρ	Density
I^0_{ca}	Enthalpy corresponding to cold air	ΔS	Entropy production
Δt	Temperature difference	ΔE	Exergy loss
d_o	Outer diameter	τ	Conversion coefficient
d_i	Inlet diameter	C_F	Operating cost of the heating surface
s_1	Transverse pitch	c_F	Operating cost per unit heat transfer
s_2	Longitudinal pitch	N	Average annual operating hours
l	Tube length	C_1	Heating surface investment cost
δ	Thickness	c_1	Cost per unit area
α_1	Heat transfer coefficient on the flue gas side	γ	Cost coefficient for operation and maintenance
α_d	Convective heat transfer coefficient of flue gas	CRF	Investment recovery coefficient
α_f	Radiative heat transfer coefficient	i	Loan interest rate
w	Flow velocity	n	Payback period of investment

2.1 Thermal calculation model of heating surface in tail flue

In a boiler, the heat transfer process of the heating surface in tail flue contains not only convection, but also radiation. To facilitate the calculation, the radiative heat transfer is converted to convective heat transfer and calculated by using Newton's law of cooling [7]. The calculation process is shown below.

(1) With given inlet and outlet parameters of the working medium, the heat absorbed by the working fluid is:

$$Q=D(i''-i')/B_j \qquad (1)$$

Where D is the mass flow rate of the working fluid with a unit of kg/s, i' and i'' are the fluid enthalpy at the inlet temperature t' and outlet temperature t'', respectively, and B_j is the calculated fuel mass with a unit of kg/s.

(2) Based on energy balance, the heat released by the flue gas is the same as the heat absorbed by the working medium, which can be calculated as:

$$Q=\varphi(I'-I''+\Delta\alpha I_{ca}^{0}) \qquad (2)$$

Where φ is the thermal efficiency taken as 0.99, I' and I'' represent the inlet and outlet enthalpy corresponding to the inlet and outlet temperature ϑ' and ϑ'' of the flue gas, Δα represents the air leakage coefficient, and I_{ca}^{0} represents the enthalpy corresponding to cold air (20 - 30 ℃).

(3) The temperature difference Δt is calculated as:

$$\Delta t=\frac{\Delta t_{max}-\Delta t_{min}}{\ln\frac{\Delta t_{max}}{\Delta t_{min}}} \qquad (3)$$

Where Δt_{max} and Δt_{min} are the maximum and minimum temperature differences between flue gas and working medium on the heating surface, respectively. The position of them depends on the flow directions.

(4) Design the structural parameters of heating surface, and give the values of tube length, inner and outer diameters, transverse pitch, longitudinal pitch, and other parameters. Since there is no comprehensive reference for the design parameters of S-CO_2 system, these values are designed according to the design experience of heating surface of traditional water steam systems [8]. The ranges of structural parameters of heating surface in tail flue are as follows: the outer diameter of tube d_o ranges from 20-60 mm, the transverse pitch s_1 ranges from 20-100 mm, the longitudinal pitch s_2 ranges from 20-400 mm, the number of tubes in a single tube panel n_s ranges from 2 to 16, the tube length l ranges from 30-500 m. The relevant structural parameters are shown in Figure 2.

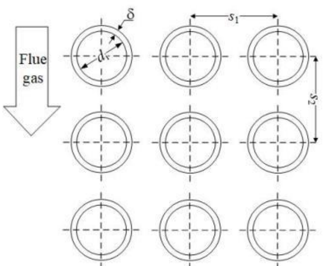

Figure 2 Structural parameters of the heating surface in tail flue

(5) For the flue gas side, after converting radiative heat transfer to convective heat transfer, a converted heat transfer coefficient [9] is obtained, which is called the radiative heat transfer coefficient α_f. The heat transfer coefficient on the flue gas side α_1 can be calculated by the following formula:

$$\alpha_1=\xi(\alpha_d+\alpha_f) \qquad (4)$$

Where α_d is the convective heat transfer coefficient of flue gas on the surface, and ξ is the correction coefficient. When the flue gas transversely flows on the in-line tube bundle, take ξ = 1.0.

$$\alpha_d=0.2C_sC_z\frac{\lambda_y}{d_0}\left(\frac{w_yd_0}{v_y}\right)^{0.65}Pr_y^{0.33} \qquad (5)$$

$$\alpha_f=5.7\times10^{-8}\,\frac{a_{hb}+1}{2}\,\alpha_hT^3\times\frac{1-\left(\frac{T_{hb}}{T}\right)^4}{1-\left(\frac{T_{hb}}{T}\right)} \qquad (6)$$

C_s refers to the structural correction coefficient, which is related to σ_1 and σ_2 and can be calculated as:

$$C_s=\left[1+\left(2\sigma_1-3\right)\left(1-\frac{\sigma_2}{2}\right)^3\right]^{-2} \qquad (7)$$

Where $\sigma_1=s_1/d_o$ and $\sigma_2=s_2/d_o$ with s_1 and s_2 being the transverse and longitudinal pitch sizes, respectively. When $\sigma_1\leqslant1.5$ or $\sigma_2\geqslant2$, C_s=1; when $\sigma_1<2$ and $\sigma_2>3$, take σ_1=3.

C_z represents the correction coefficient of the number of tube rows, and its value is related to the average number of rows z_2 of each tube group of the calculated tube bundle, which is calculated according to the following formula:

$$C_z=0.91+0.0125(z_2-2)\ \text{ when } z_2<10 \qquad (8)$$

$$C_z=1\ \text{ when } z_2\geqslant10 \qquad (9)$$

d_o is the tube outer diameter, w_y is the flow velocity of the flue gas, λ_y, v_y and Pr_y represent thermal conductivity, viscosity, and Prandtl number of the flue gas under average temperature, respectively.

a_{hb} is the blackness of the tube wall, taken as 0.82. T represents the average temperature of flue gas, T_{hb} represents the temperature of the heating surface with ash deposition, a_h represents the blackness of ash containing flue gas at temperature T.

(6) For the working medium side, since CO_2 is in supercritical state and there is no phase change process in the tube, the surface heat transfer coefficient α_2 can be calculated by the following formula:

$$\alpha_2=0.023C_tC_l\frac{\lambda_g}{d_{dl}}\left(\frac{w_gd_{dl}}{v_g}\right)^{0.8}Pr_g^{0.4} \qquad (10)$$

Where subscript g represents parameters for working medium side, d_{dl} is the equivalent diameter (taken as the inner diameter for circular tube). C_t represents the correction coefficient between fluid temperature and tube wall temperature, which can be taken as 1. C_l represents the correction coefficient of the relative length of the flowing heating surface and can also be taken as 1.

(7) According to the basic principle of heat transfer [10], the general expression of heat transfer coefficient K is:

$$K=\frac{1}{\frac{1}{\alpha_{1h}}+\frac{\delta_h}{\lambda_h}+\frac{\delta_b}{\lambda_b}+\frac{\delta_{sg}}{\lambda_{sg}}+\frac{1}{\alpha_2}} \qquad (11)$$

Where α_{1h} is the surface heat transfer coefficient of flue gas to the tube wall surface with ash deposition. δ_h is the thickness of ash layer, and λ_h is the thermal conductivity of ash layer. δ_b is the thickness of tube wall, and λ_b is the thermal conductivity of tube wall. The thermal resistance of the metal tube wall $\frac{\delta_b}{\lambda_b}$ is much smaller than other terms and is negligible. δ_{sg} is the thickness of scale layer, and λ_{sg} is the thermal conductivity of scale layer. Scale deposition is not allowed on the inner wall of the pipe, so that the thermal resistance $\frac{\delta_{sg}}{\lambda_{sg}}$ can be omitted.

In the actual calculation, the following two methods are usually adopted for simplification of the calculation:

(1) Introducing the ash fouling coefficient ε to correct the surface heat transfer coefficient on the flue gas side:

$$\frac{1}{\alpha_{1h}}+\frac{\delta_h}{\lambda_h}=\frac{1}{\alpha_1}+\varepsilon \tag{12}$$

The heat transfer coefficient can then be written as:

$$K=\frac{1}{\frac{1}{\alpha_1}+\varepsilon+\frac{1}{\alpha_2}} \tag{13}$$

(2) Introducing the thermal effective coefficient Ψ to correct the change of heat transfer coefficient caused by pollution:

$$\Psi=\frac{K}{K_0} \tag{14}$$

The heat transfer coefficient can then be written as:

$$K=\Psi\frac{1}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}} \tag{15}$$

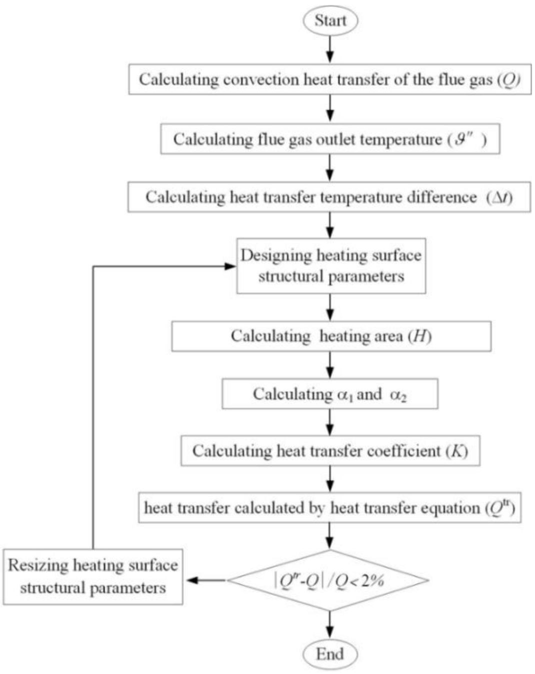

Figure 3 Flow chart of the thermal calculation model

(3) The convective heat transfer of the heating surface is obtained using $Q^{tr}=KH\Delta t/B_j$, where H is the heating area. If the difference between the calculated convective heat transfer Q^{tr} and Q is within an acceptable range $\left|\frac{Q^{tr}-Q}{Q}\right|\cdot 100\% \leq 2\%$, the design process of the heating surface is completed. Otherwise, return to step 4, adjust the structural parameters of the heating surface according to the existing calculation results, and repeat the process. The process is shown in Figure 3. The thermophysical properties of supercritical CO_2 were from NIST REFPROP V9.0[11],and thermophysical properties of flue gas were from Ref [6].

2.2 Pressure drop calculation model of heating surface in tail flue

To measure the performance of a heating surface, the pressure drops on the flue gas side and the CO_2 side are important indexes, and can also be used to judge whether the design of the heating surface is reasonable. For the tail flue, the temperature and velocity in the pressure drop calculation formula are taken as the average temperature and average speed [12].

Pressure drop of flue gas ΔP_y can be calculated by:

$$\Delta P_y=\xi_0 z_2\frac{w_y^2}{2}\rho_y \tag{16}$$

Where ξ_0 is the friction factor of a single tube row in the tube bundle, and ρ_y is the equivalent flue gas density. ξ_0 can be calculated based on the structural parameters of the heating surface.

When $\sigma_1 \leq \sigma_2$:

$$\xi_0=2\left(\frac{s_1}{d_0}-1\right)^{-0.5}Re_y^{-0.2} \tag{17}$$

When $\sigma_1>\sigma_2$:

$$\xi_0=0.38\left(\frac{s_1}{d_0}-1\right)^{-0.5}\left(\frac{s_1-d_0}{s_2-d_0}-0.94\right)^{-0.59}Re_y^{\overline{-0.2}\left(\frac{s_1-d_0}{s_2-d_0}\right)^2} \tag{18}$$

Where Re_y is the Reynolds number on the flue gas side.

According to the principle of momentum conservation, when the fluid flows in the tube, the total pressure drop ΔP_g consists of gravitational pressure drop, frictional pressure drop and local resistance loss pressure drop. In the current model, the gravitational pressure drop and local resistance loss pressure drop are ignored, and the frictional pressure drop is calculated using the empirical formula for turbulent flow:

$$\Delta P_g=\zeta\frac{l}{d_{dl}}\frac{w_g^2}{2}\rho_g \tag{19}$$

Where ζ is the friction factor in the turbulent flow regime calculated as ζ=0.0055[1+(0.16/d_i+$10^6/Re_g)^{1/3}$], and l is the length of the tube.

The data for the type of coal selected in this work (Table 2), the structural parameters of heating surface

(Table 3), and the design parameters of the boiler (Table 4) are obtained from Ref [9].

Table 2 Properties of the designed coal-type

C_{ar}	H_{ar}	O_{ar}	N_{ar}	S_{ar}	A_{ar}	M_{ar}	V_{daf}	$Q_{net,ar}$
61.7	3.67	8.56	1.12	0.6	8.8	15.55	0.3473	23442

C(carbon), *H*(hydrogen), *O*(oxygen), *N*(nitrogen), *S*(sulfur), A(ash), *M*(moisture), *V*(Volatile)

Table 3 Structural parameters of the heating surface

Parameters	Unit	Formula	Values
Width (a)	m	set value	10
Depth (b)	m	set value	30
Inner diameter (d_i)	mm	set value	27
Thickness (δ)	mm	$10^6P'\times d_i/(156\times10^6-10^6P')+1$	7.583
Outer diameter (d_o)	mm	$d_i+2\delta$	42.166
Transverse pitch (s_1)	mm	$2d_o$	105.416
Longitudinal pitch (s_2)	mm	$1.5d_o$	63.249
Tube length (l)	m	set value	233.8
Number of tube panels (n_1)	/	$(30-0.12)/(s_1)+1$	284
Number of tubes in tube panels (n_2)	/	set value	7
Total number of tubes (n)	/	n_1n_2	1988
Area of heating surface (H)	m^2	$\pi d_o l\times n$	61571.038
Cross-sectional area of CO_2 flow (f)	m^2	$\pi/4\times d_i^2\times n$	1.138
Cross-sectional area of flue gas flow (F_y)	m^2	$30\times14-(14-0.12)d_on_1$	181.685
Effective thickness of radiation layer (S)	m	$0.9d_o(4s_1s_2/\pi d_o^2-1)$	0.143

Table 4 Parameters for the boiler

Parameters	Unit	Values
Boiler efficiency (η_b)	%	94.43
Exit flue gas temperature (ϑ_{exg})	℃	123
Primary air temperature	℃	320
Primary air flow rate ratio	%	19
Secondary air temperature	℃	336
Cold secondary air temperature	℃	21
Environment temperature (t_{ca})	℃	20
Calculate fuel consumption (B_j)	kg/s	87.4

2.3 Model verification

First of all, the thermal calculation and pressure drop calculation is verified for a heating surface of a 1000 MW S-CO_2 coal-fired boiler [13], such as Table 5, it can be seen that the thermal calculation and pressure drop calculation models compiled in this paper can meet the deviation of the heating surface is basically within the allowable range.

Table 5 Model Verification of 1000 MW S-CO_2 coal-fired boiler.

Parameters	Unit	Literature value	Calculated value	Deviation (%)
heat absorption Q	MW	223.46	225.57	0.94
heat transfer coefficient K	W/(m^2°C)	53.10	54.02	1.73
temperature difference Δt	℃	55.38	55.38	0
heating area H	m^2	67004	66107	1.34
Pressure drop of flue gas ΔP_y	Pa	602.4	598	0.73
Pressure drop of fluid ΔP_g	MPa	0.0957	0.0969	1.25

3 Optimal Design Based on Exergy Loss Analysis

3.1 Entropy production of the heating surface in tail flue

Due to the existence of temperature difference and flow resistance in the heating surface, the total entropy production of the heating surface in tail flue ΔS_g is composed of heat transfer entropy production $\Delta S_{g\Delta T}$ and flow entropy production $\Delta S_{g\Delta P}$. The heat transfer entropy production can be calculated as [14]:

$$\Delta S_{g,\Delta T}=\dot{m}_f c_f \ln\frac{T_{f,o}}{T_{f,i}}+\dot{m}_y c_y \ln\frac{T_{y,o}}{T_{y,i}} \tag{20}$$

Where the subscripts f, y, i, and o represent working medium side, flue gas side, inlet, and outlet, respectively. $\dot{m}$ is the mass flow rate.

The flow entropy production can be calculated as [15]:

$$\Delta S_{g,\Delta P}=\dot{m}_f\frac{\Delta P_f}{\rho_f}\frac{\ln(T_{f,o}/T_{f,i})}{T_{f,o}-T_{f,i}}-\dot{m}_f R_f \ln\frac{P_{y,o}}{P_{y,i}} \tag{21}$$

Finally, the total entropy production of the heating surface is:

$$\Delta S_g=\Delta S_{g,\Delta T}+\Delta S_{g,\Delta P} \tag{22}$$

3.2 Exergy loss of the heating surface in tail flue

The exergy loss is used to judge whether the design of the heating surface is reasonable. In the process of heat transfer, there are operation costs of the heating surface, including cost of heat transfer exergy loss, cost of flow exergy loss, and investment cost of the heating surface. The heat transfer exergy loss $\Delta E_{\Delta T}$ and flow exergy loss $\Delta E_{\Delta P}$ can be obtained by [16]:

$$\Delta E_{\Delta T}=T_0\cdot\Delta S_{g,\Delta T} \tag{23}$$

$$\Delta E_{\Delta P}=T_0\cdot\Delta S_{g,\Delta P} \tag{24}$$

It is worth noting that the costs of heat transfer exergy loss and flow exergy loss cannot be simply added, because the flow of working medium is driven by the pump and

Table 6 Calculated parameters with different inner diameters (Width=10 m)

Number	d_i/(mm)	d_o/(mm)	s_1/(mm)	s_2/(mm)	l/(m)	H/(m^2)	K/W/(m^2·°C)	ΔP_g/(MPa)
1	24	37.7	94.3	56.6	226	59589	62.23	0.172
2	25	39.2	98.0	58.8	229.5	60328	61.47	0.153
3	26	40.7	101.7	61.0	231.8	60964	60.82	0.136
4	27	42.2	105.4	63.3	233.8	61571	60.22	0.121
5	28	43.7	109.1	65.5	236.5	62209	59.60	0.109
6	29	45.1	112.9	67.7	238.8	62821	59.03	0.098
7	30	46.6	116.6	69.9	240.4	63354	58.52	0.088
8	31	48.7	120.3	72.2	242.6	63920	58.00	0.080
9	32	49.6	124.0	74.4	245.4	64515	57.47	0.074
10	33	51.1	127.7	76.6	247.4	65046	57.00	0.067

consumes mechanical work, this paper need to convert the cost of flow exergy loss before adding. The conversion coefficient between them is τ=3-5, and the operating cost of the heating surface C_F can be calculated by [17]:

$$C_F=c_F\cdot(\Delta E_{\Delta T}+\tau\cdot\Delta E_{\Delta P})\cdot N \quad (25)$$

Where c_F represents the operating cost per unit heat transfer and is taken as 40 ¥/GJ. N represents the average annual operating hours and is taken as 5400 h.

The heating surface investment cost C_1 can be calculated by the following formula :

$$C_1=c_1\cdot H\cdot(\mathrm{CRF}+\gamma) \quad (26)$$

Where c_1 represents the cost per unit area, taken as 650 ¥/m^2, and H represents the area of the heat exchanger. γ represents the cost coefficient for operation and maintenance which is taken as 0.06, and CRF represents the investment recovery coefficient, which can be obtained by [18]:

$$\mathrm{CRF}=[i\cdot(1+i)^n]/[(1+i)^n-1] \quad (27)$$

Where i represents the loan interest rate which is 0.08, and n represents the payback period of investment which is 20 years.

Therefore, this paper can obtain the total cost C of the heating surface as the sum of the operating cost C_F and the investment cost C_1, i.e., $C=C_F+C_1$. The cost per unit heat transfer of the heating surface is obtained as:

$$\overline{C}=\frac{C}{Q} \quad (28)$$

Where Q is the total thermal load of the heating surface.

4 Results and Discussion

4.1 Optimization results and analysis with same boiler width and different inner diameters

Based on the thermal calculation and pressure drop calculation models, the corresponding heat transfer coefficient, heating area and working medium pressure drop on both sides of the heating surface can be obtained for the same boiler width (10 m), as shown in Table 6. In Figure 4, it can be seen that with the change of tube inner diameter d_i, the heating area H and working medium pressure drop ΔP_g show opposite trends: with the increase of tube inner diameter, the heating area increases and the working medium pressure drop decreases.

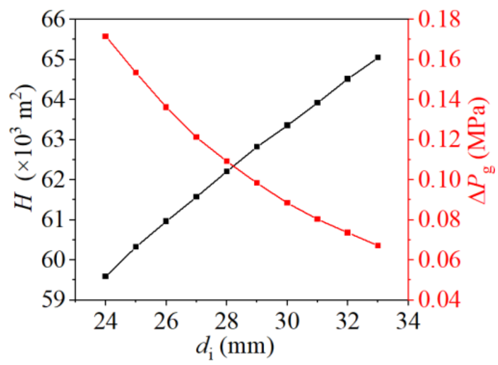

Figure 4 Variations of heating surface area and CO_2 pressure drop with inner diameter

By combining the exergy loss analysis and converting the irreversible loss into the economic index, it is found that when the boiler width is 10 m, with the increase of the inner tube diameter, the cost per unit heat transfer of the heating surface does not show a monotonic variation; it decreases and then increases as shown in Figure 5. Therefore, there is an optimal tube diameter. From Figure 5, it can be seen that the optimal tube diameter is 27 mm. The cost per unit heat transfer of the corresponding heating surface is the lowest, only 4.705 ¥/GJ.

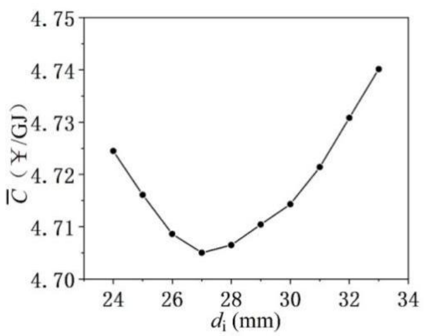

Figure 5 Variation of the cost per unit heat transfer of the heating surface with different inner diameters

4.2 Optimization results and analysis of optimal inner diameter with different boiler widths

The optimal structural parameters corresponding to

Table 7 Optimal structural parameters with different boiler widths

a/(m)	d_i/(mm)	d_o/(mm)	s_1/(mm)	s_2/(mm)	l/(m)	H/(m^2)	K/W/($m^2\cdot{}^oC$)	ΔP_y/(Pa)
8	27	42.17	105.4	63.2	207.7	54698	67.80	931.7
9	27	42.17	105.4	63.2	221.1	58227	63.68	739.7
10	27	42.17	105.4	63.2	233.8	61571	60.22	601.6
11	27	42.17	105.4	63.2	245.9	64758	57.26	498.9
12	27	42.17	105.4	63.2	257.4	67786	54.70	420.6
13	27	42.17	105.4	63.2	268.4	70683	52.45	359.3
14	28	43.65	109.1	65.5	281.9	74150	50.00	309.9

the boiler width of 8-14 m are calculated, and the results are shown in Table 7. Figure 6 shows the changes of heating area H and flue gas pressure drop ΔP_y with the boiler width, it is found that the changes of heating area and flue gas pressure drop with different boiler widths are greater than those when the inner diameter is varied with the same boiler width. With varying boiler width, the heating area and flue gas pressure drop also change with opposite trends. Therefore, as the boiler width is varied, the designed heating surface still needs to be optimized by the method of exergy loss analysis.

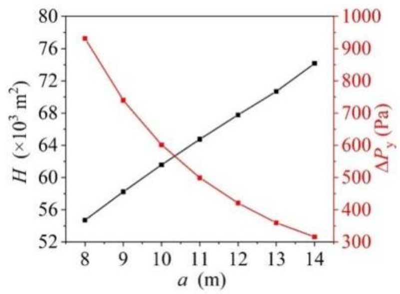

Figure 6 The variations of heating surface area and gas pressure drop with the boiler width

Figure 7 shows the exergy loss analysis results. It shows that with the increase of boiler width a, the cost per unit heat transfer of the heating surface also decreases first and then increases, creating an optimal boiler width of 10 m. The optimal boiler width corresponds to the lowest the cost per unit heat transfer of heating surface of 4.705 ¥/GJ.

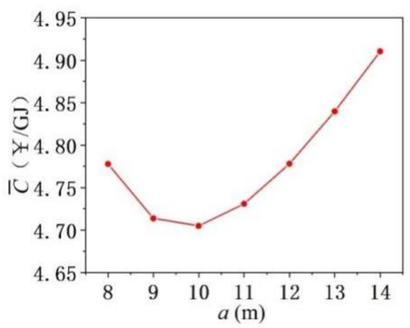

Figure 7 Variation of the cost per unit heat transfer of heating surface with different boiler widths

4.3 Sensitivity analysis

After the calculation, the sensitivity analysis of economic parameters is carried out: cost per unit area c_1, operating cost per unit heat transfer c_F and the conversion coefficient τ fluctuate by 20%. The results are shown in Figure 8: with the change of c_1, the change of $\overline{C}$ is the most obvious. Therefore, c_1 has the greatest influence, so in the process of optimization, the setting of c_1 must be in a reasonable range. After calculation, when c_1 is 650 ¥/m^2, the parameters of the heating surface, such as flue gas velocity, flue gas pressure drop, fluid pressure drop and so on, are the most reasonable.

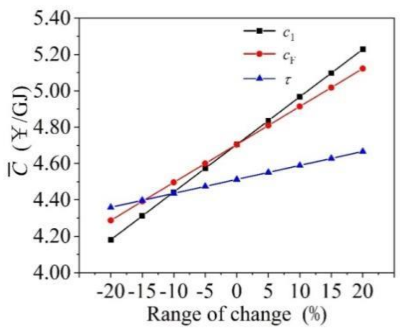

Figure 8 Sensitivity analysis of economic parameters

5 Conclusions

In order to achieve tasks of "carbon peak emission" and "carbon neutrality", it is imperative to develop more efficient and cleaner power generation technologies. The S-CO_2 cycle can greatly improve power generation efficiency compared with water steam Rankine cycle, meanwhile reduce carbon emissions. The heating surface in tail flue of an S-CO_2 top-bottom combined cycle based on overlap energy utilization has small temperature difference and large thermal load, which will lead to excessive heating area. This work develops the models to optimize the structural parameters of the heating surface in tail flue. Combined with the exergy analysis method, the heating surface in tail flue of a 1000 MW power generation system is optimized. It is concluded that the optimal boiler width is 10 m and the optimal inner tube diameter is 27 mm.

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