PERFORMANCE IMPROVEMENT OF A 330 MW_e POWER PLANT BY FLUE GAS HEAT RECOVERY SYSTEM

by

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Original scientific paper
DOI: 10.2298/TSCI140104099X

In a utility boiler, the most heat loss is from the exhaust flue gas. In order to reduce the exhaust flue gas temperature and further boost the plant efficiency, an improved indirect flue gas heat recovery system and an additional economizer system are proposed. The waste heat of flue gas is used for high-pressure condensate regeneration heating. This reduces high pressure steam extraction from steam turbine and more power is generated. The waste heat recovery of flue gas decreases coal consumption. Other approaches for heat recovery of flue gas, direct utilization of flue gas energy and indirect flue gas heat recovery system, are also considered in this work. The proposed systems coupled with a reference 330 MW_e power plant are simulated using equivalent enthalpy drop method. The results show that the additional economizer scheme has the best performance. When the exhaust flue gas temperature decreases from 153 °C to 123 °C, power output increases by 6.37 MW_e and increment in plant efficiency is about 1.89%. For the improved indirect flue gas heat recovery system, power output increases by 5.68 MW_e and the increment in plant efficiency is 1.69%.

Key words: *waste heat recovery, flue gas, coal power plant, efficiency*

Introduction

Coal is a very important fossil fuel. It is abundant and widely distributed in geography, but the coal utilization is related to environmental issues. In China, coal-fired power dominates power production sources. It is reported that by the end of 2011, the total installed capacity of conventional thermal power was 1055.76 GW, the majority of which was coal-fired power plants (over 765.4 GW) [1]. Therefore, better energy efficiency in coal-fired power plant is demanded.

Due to the more and more stringent requirements of energy conservation and emissions reduction, there is a growing concern over the efficiency increase of coal-fired power plants. The largest heat loss in a boiler is in the exhaust flue gas, which greatly affects the thermal efficiency. It is widely accepted that 1% of the coal can be saved if the flue gas temperature is reduced by 12~15 °C [2].

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Currently flue gas waste heat has been recovered in a certain extent, which is used to heat condensed water, cold air, and hot water of heating network [3-8]. Qun *et al*. [3] investigated technologies which exploit the low grade heat available from a flue gas condensing system through industrial condensing boilers. Biščan and Filipan [4] specifically analyzed the flue gas waste heat potential in Croatian industrial sector. Blarke [6] integrated a heat pump using low-temperature heat recovered from flue gas in distributed co-generation. However there is still great potential to recover the heat of the exhaust flue gas from a boiler [9, 10]. Exhaust temperature of utility boiler is 120~140 °C. Potential of flue gas energy utilization greatly depends on the temperature of flue gas dew point [11, 12]. With the progress of flue gas desulphurization and denitration technologies, the flue gas dew point could be reduced to 90 °C. But in practical operation, the flue gas temperature of supercritical or ultra-supercritical unit is generally higher than the designed value, increasing the heat loss due to exhaust gases. Lots of factors can lead to higher exhaust flue gas temperature, such as discrepancy between the fired coal and the designed coal, ash sticking of the heating surface, and unsuitable heating surface arrangement [13].

The conventional methods to reduce the exhaust flue gas temperature and to enhance the waste heat recovery are to increase the heating surface. One is to increase the heat transfer surface of air preheater. More heat exchange can be obtained in the air preheater. However, it is not always better for larger heat transfer surface. First, heat transfer surface increase could lead to a capital cost increase. Second, there is an air temperature limitation in order to avoid fires in the primary air line. Third, the too low temperature of flue gas can cause corrosion in the air preheater. The other alternative is to enlarge the heat transfer surface of economizer. This measure can decrease the flue gas temperature but the recovery of heat is limited.

Different approaches to recover the flue gas waste heat have been proposed and developed. However, most of these designs are to cool the flue gas directly with water and the heated water temperature can not be high enough. For example, Kolev and Kolev [14] designed a new lamellar-type heat exchanger and it is especially appropriate for heating of the low temperature feed water for boilers . For material conditions, Wang *et al*. [15] developed an advanced flue gas waste heat recovery technology using the patented transport membrane condenser. This technology is particularly beneficial to high-moisture coal fired power plant because the latent heat of water vapor from flue gas is utilized. Westerlund *et al.* [16] installed an open absorption system in a heat production unit. The design can not only recover flue gas heat, but also fulfill a reduction of particles in the flue gas. But those two technologies pay little attention to regenerative heating cycle. Other methods to use the flue gas waste heat is to install a low pressure (LP) economizer to heat condensate (*i. e*. LP feed water – LP- FW) [17]. Its performance and benefits were analyzed using equivalent enthalpy drop method (EEDM) [18]. Xu *et al.* [19] conducted a techno-economic analysis and optimization design of four typical flue gas heat recovery schemes.The system design is simple, economical and reliable with reasonable increase of efficiency. Several flue gas heat recovery schemes were simulated using Aspen Plus to analyze power output and net efficiency increase of a supercritical plant [20]. It is pointed out that indirect flue gas heat recovery system is superior to direct use of flue gas energy. The *plastic* heaters used in flue gas heat recovery system are desirable to further develop. The utilization of exhaust energy in the heating of condensed water is a relatively mature technology. It can save a large amount of steam to increase unit efficiency and to reduce energy consumption. Generally high-stage steam substitute scheme shows better energy-saving effect [19]. However, very few studies have focused on high-stage flue gas temperature and almost no heat cascade utilization schemes are presented.

In this paper, an improved indirect flue gas heat recovery system and an additional economizer scheme are proposed. Performances of the proposed schemes coupled with a 330 MW_e power plant are investigated and compared with the direct and indirect flue gas heat recovery system using EEDM.

Flue gas waste heat utilization schemes

Direct utilization of flue gas energy

The flue gas waste heat is used directly to heat LP-FW feed water, as seen in fig. 1 [20]. In this case, the rotary air preheater is not modified and an additional flue gas-water heat exchanger is used to enforce the heat exchange between flue gas and LP condensate. Flue gas from economizer enters the rotary air preheater to preheat the primary and secondary air. Then the flue gas enters the LP-FW heater, releasing heat to lower temperature condensate. Finally, the flue gas is cooled down and vented to flue gas desulturization (FGD). The additional flue gas energy substitutes regenerative heating of the LP condensate to save LP steam extraction from steam turbine, increasing both steam power output and net efficiency of the plant. The amount of condensate extracted from regenerative heating cycle depends on the amount of flue gas heat recovery. The extracted condensate is heated up in the LP-FW heater and then returns to the regenerative heating cycle.

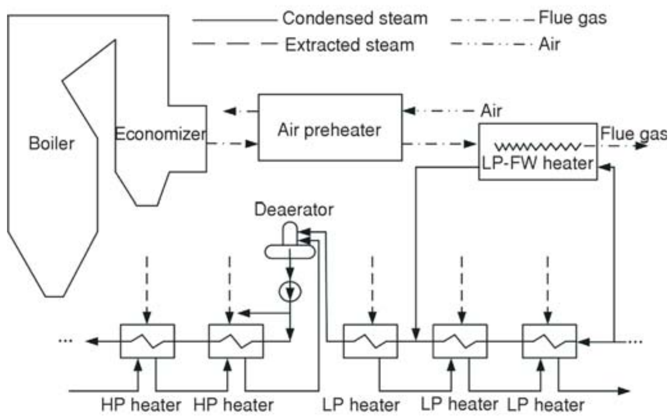

Figure 1. A schematic system of direct utilization of flue gas energy

Indirect flue gas heat recovery system

Indirect flue gas heat recovery system, as seen in fig. 2, includes an indirect air preheating unit (a flue gas-conduction media heat exchanger + a conduction media-air heat exchanger), a high pressure feed water (HP-FW) heater, and a LP-FW heater [20]. The amount of recovery energy is equal to that used directly in

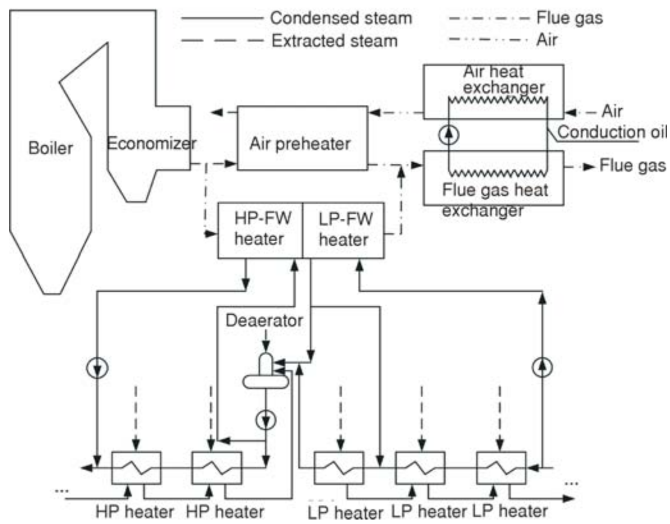

Figure 2. A schematic system of indirect flue gas heat recovery system

scheme 1. In this case, a certain amount of flue gas from economizer is bypassed before rotary air-flue gas preheater, and enters HP-FW heater and LP-FW heater successively, releasing heat to HP condensate and LP condensate. Then the bypassed flue gas, together with the flue gas from the rotary air-flue gas preheater, goes down to the indirect air preheating unit, releasing heat to air. Air temperature is raised before going to the rotary air-flue gas preheater. The recovered flue gas energy is transferred to the energy released from the bypassed flue gas by an indirect air preheating unit, but the bypassed flue gas has a higher temperature and is used to heat HP condensate, saving HP steam extracted from steam turbine for regenerative heating of condensate, thereby increasing the plant power. Because of the indirect air preheating unit, the temperature of air that exits from rotary air preheater can be kept unchanged even though the flue gas from economizer to rotary air preheater is reduced. In addition, the cold end corrosion of rotary air preheater is avoided, and the cold-warm end deformation of air preheater is also alleviated owing to the decrease of temperature difference between two sides of the air preheater.

Improved indirect flue gas heat recovery system

Improved indirect flue gas heat recovery system, as seen in fig. 3, has only one HP-FW heater as compared to the system in fig. 2. Indirect air preheating unit is also needed. Comparing with indirect flue gas heat recovery system, bypassed flue gas shares more from the total. The amount of the bypassed flue gas energy only heats HP condensate. Therefore, this system contains only HP-FW heater. The design is relatively simple. The recovered flue gas energy saves more HP steam extraction from turbine, and more extra power is generated.

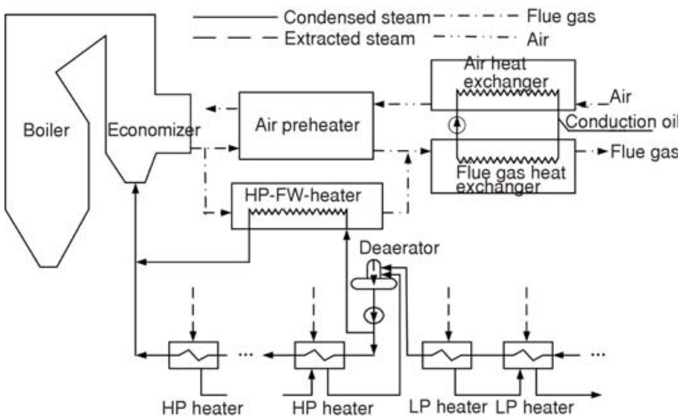

Figure 3. A schematic system of improved indirect flue gas heat recovery system

It should be noted that the mixed flue gas before the indirect air preheating unit has higher temperature, therefore more heating surfaces of indirect air preheating unit are needed.

Additional economizer scheme

The temperature of flue gas from economizer to rotary air preheater can be lowered and the extra heat can be used to heat the feed water by adding an additional economizer (HP-FW heater), as shown fig. 4(a). The HP-FW heater is used as the last stage HP regenerative FW heater. As compared with scheme 2 and scheme 3, the configuration is much simpler and the highest extraction steam is saved to increase the power output. The scheme of fig. 4(a) can be simplified further as fig. 4(b). In fig. 4(a) a HP-FW heater is needed between the economizer and the rotary air preheater, while in fig. 4(b), additional heating surface of economizer is directly added to the economizer. In order to keep the approaching temperature difference at the outlet of economizer, bypassed feed water from the inlet of last stage HP regenerator to its outlet is selectable.

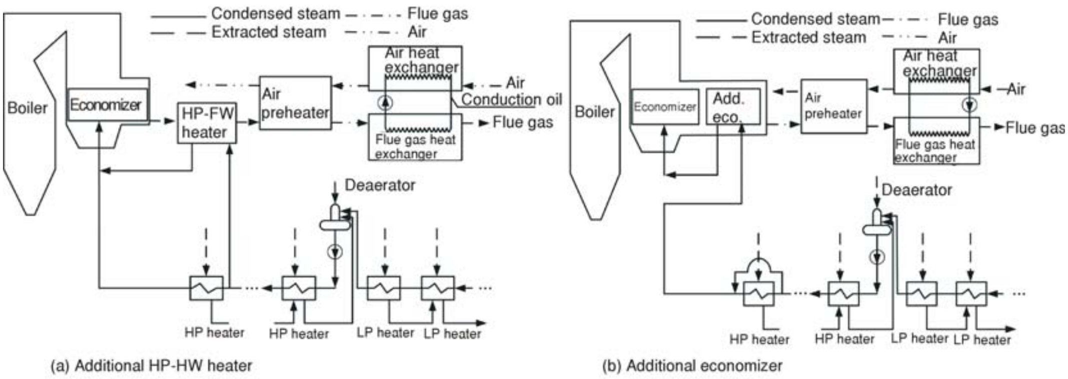

Figure 4. Schematic systems of additional economizer scheme including (a) HP-FW heater and (b) additional economizer

Equivalent enthalpy drop method analysis

The EEDM is used to evaluate the performances of all the four schemes described.

The flue gas waste heat is used to heat the condensate in steam turbine regenerative heat system, consequently extraction steam is reduced. Based on EEDM, it is equivalent to the increased work of steam.

The work done by flue gas energy utilization ΔH [kJh^{-1}] can be expressed [21]:

$$\Delta H = \sum^{j} \eta_j \Delta Q_j \tag{1}$$

where j is the number of regenerator, $\eta_j = H_j/q_j$ – the extraction steam efficiency of j regenerator, H_j – the extraction steam equivalent enthalpy drop of j regenerator, q_i – the releasing heat of extraction steam of j regenerator, and ΔQ_j [kJh^{-1}] – the heat derived from flue gas energy of j regenerator. Using EEDM, η_j can be obtained from the enthalpy values in steam turbine thermal equilibrium diagram. Based on ultimate analysis data of selected coal and each stage of flue gas temperature, the heat of flue gas energy utilization can be determined, and then ΔQ_j is calculated by j regenerator's share of feed water enthalpy rise.

The relative variation of the system efficiency is defined:

$$\delta\eta = \frac{\Delta H}{H + \Delta H} \tag{2}$$

where H [kJkg^{-1}] is live steam equivalent enthalpy drop.

Thus the variation of standard coal consumption Δb [gkW^{-1}h^{-1}] can be calculated:

$$\Delta b = b\delta\eta \tag{3}$$

where b [gkW^{-1}h^{-1}] is standard coal consumption for power plant.

Case study and performance calculation

In this work, we attempt to figure out the influence of power plant efficiency with the four kinds of flue gas waste heat utilization schemes. A reference subcritical plant with a rated power of 330 MW is selected. Boiler efficiency is 88.07% (HHV basis) and 92.16% (LHV ba-

Table 1. Reference power plant data

Power plant data (100% load)	
Gross electrical output	330 MW
Main steam	1016.55 t/h
Main steam temperature	537 °C
Main steam pressure	16.7 MPa
Reheat steam temperature	537 °C
Reheat steam pressure	3.654 MPa
Extraction pressure	1.0 (1.0-1.2) kPa
Rated extraction steam	80 t/h
Maximum extraction steam	180 t/h
Condenser pressure	4.9 kPa
Feed water temperature	282.1 °C
Net heat rate	7871 kJ/kWh
Steam rate	3.08 kg/kWh

Table 2. Coal ultimate analysis data

Coal elemental analysis data (as-received)		
Carbon	C_{ar}	51.32%
Hydrogen	H_{ar}	3.1%
Oxygen	O_{ar}	4.73%
Nitrogen	N_{ar}	1.06%
Sulphur	S_{ar}	0.79%
Moisture	M_{ar}	9%
Ash	A_{ar}	30%
LHV	$Q_{ar,l}$	20098 kJ/kg
HHV	$Q_{ar,h}$	21031 kJ/kg

sis). The overall efficiency of the power plant is 41.7%. The steam cycle data of the reference power plant are listed in tab. 1. Table 2 shows the coal ultimate analysis data.

The flow sheet of the power cycle is shown in fig. 5. The steam turbine regenerative system consists of three HP heaters, a deaerator and four LP heaters. According to the calculated data given in fig. 5, the related parameters based on EEDM can be obtained, as seen in tab. 3 (the enthalpy unit is kJ/kg).

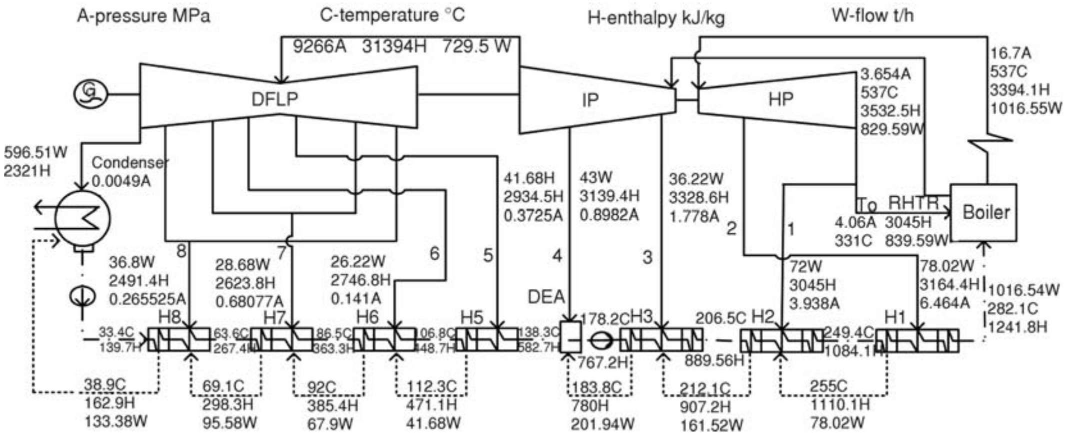

Figure 5. Flow sheet of the reference power cycle

The flue gas enters the rotary air preheater at 392 °C and leaves at 153 °C, which is higher than the designed exhaust temperature. The thermal parameters of the air preheater can be calculated by the coal composition and unit data, see tab. 4.

It is assumed that the flue gas temperatures are all the same down to 123 ° when the regenerative system is integrated with different flue gas waste heat recovery schemes. The minimum temperature difference between flue gas and condensate is considered to be 20 °C~30 °C.

(1) *Direct utilization of flue gas energy*

Flue gas waste energy is used to heat condensate of H5 and H6, as shown in fig. 6(a). A fraction of LP condensate is extracted from the inlet of H6 (T_{fwh6} = 86.5 °C), then it is heated up to

Table 3. Power plant steam cycle parameters based on EEDM

Heater, *j*	*j*	H1	H2	H3	DEA	H5	H6	H7	H8
Inlet feed water enthalpy	$h_{w(j+1)}$	1084.1	889.56	767.2	582.7	448.7	363.3	267.4	139.7
Outlet feed water enthalpy	h_{wj}	1241.8	1084.1	889.56	742.6	582.7	448.7	363.3	267.4
Extraction enthalpy	h_j	3164.4	3045	3328.6	3139.4	2934.5	2746.8	2623.8	2491.4
Drain enthalpy	h_{dj}	1110.1	907.2	780	0	471.1	385.4	289.3	162.9
Primary drain enthalpy	$h_{d(j-1)}$	0	1110.1	907.2	780	0	471.1	385.4	289.3
Heat of extraction steam	q_j	2054.3	2137.8	2548.6	2556.7	2463.4	2361.4	2334.5	2328.5
Drain heat	r_j	0	202.9	127.2	197.3	0	85.7	96.1	126.4
Feed water heat	t_j	157.7	194.54	122.36	159.9	134	85.4	95.9	127.7
Feed water ratio	A_j	1	1	1	1	0.752	0.752	0.752	0.752
Drain ratio	B_j	0	0.077	0.16	0.2	0	0.041	0.067	0.095
Extraction ratio	α_j	0.077	0.084	0.04	0.047	0.041	0.026	0.028	0.036
Extraction enthalpy drop	H_j	1062.6	1042.1	882.2	750.96	577.49	404.47	293.55	170.4
Extraction efficiency (η)	ef_j	0.517	0.487	0.346	0.294	0.234	0.171	0.126	0.073

T_{fwh5} = 123 °C and returns to the cycle at the inlet of H5. Consequently, the extraction steams of numbers 5 and 6 from LP turbine are reduced. The flue gas temperature finally falls to 123 °C. The heat Q_d recovered from flue gas is 12323.6 kW. The increase power of steam turbine is: $\Delta H_d = \eta_6 \Delta Q_6 + \eta_5 \Delta Q_5 =$ = 2455.3 kW.

(2) *Indirect flue gas heat recovery system*

The LP-FW heater is used to heat the LP condensate extracted from the inlet of H5 and then the condensate returns to the deaerator, as shown in fig. 6(b). The HP-FW heater is used to heat HP condensate extracted from the inlet of H3 (T_{fwh3} = 178.2 °C) and the condensate returns to the cycle at the outlet of H1 (T_{fwh1} = 282.1 °C). The temperature difference between flue gas and condensate is assumed to be 30 °C. The sum of flue gas energy obtained by HP and LP-FW heaters is equal to the direct use of heat as scheme in fig. 1, $Q_{ih} + Q_{il} = Q_d$. A 11.22% flue gas is bypassed from the inlet of rotary air preheater to HP and LP heaters. Flue gas is cooled from 392 °C to 158 °C in rotary air preheater

Table 4. Calculated thermal parameters of the air preheater

Parameters	Value	Unit
Inlet flue gas mass flow	1306960	kg/h
Inlet RO_2 volume	136277.6	Nm^3/h
Inlet N_2 volume	587982	Nm^3/h
Inlet H_2O volume	79425	Nm^3/h
Inlet excess air volume	185690.6	Nm^3/h
Inlet flue gas temperature	392	°C
Inlet RO_2 enthalpy	28578.9	kW
Inlet N_2 enthalpy	84310.1	kW
Inlet H_2O enthalpy	13523.4	kW
Inlet excess air enthalpy	27383.2	kW
Inlet flue gas enthalpy	153795.6	kW
Air flow leakage	84823	kg/h
Ambient temperature	22.8	°C
Air leakage enthalpy	592.2	kW
Outlet flue gas temperature	153	°C
Outlet RO_2 enthalpy	10169.3	kW
Outlet N_2 enthalpy	32486.0	kW
Outlet H_2O enthalpy	5120.5	kW
Outlet excess air enthalpy	10471.9	kW
Outlet air leakage enthalpy	3699.6	kW
Outlet flue gas enthalpy	61947.3	kW
Flue gas heat release in air heater	92440.5	kW

and the by-passed flue gas is cooled from 392 °C to 145 °C in the FW heaters, then the two parts are mixed to indirect air preheating unit. The flue gas temperature is further reduced to 123 °C. The steam turbine power increase is: $\Delta H_i = \eta_1 \Delta Q_1 + \eta_2 \Delta Q_2 + \eta_3 \Delta Q_3 + \eta_5 \Delta Q_5 = 5054.4$ kW.

(3) *Improved indirect flue gas heat recovery system*

The by-passed flue gas is used to heat HP condensate derived from the inlet of H3 regenerator (T_{fwh3} = 178.2 °C) and the condensate goes back to the cycle at the outlet of H1 (T_{fwh1} = 282.1 °C), as shown in fig. 6(c). The recovered flue gas energy by HP-FW heater is kept the same, $Q_\mathrm{r} = Q_\mathrm{d}$. Accordingly 15.08% flue gas is by-passed to HP-FW heater. The flue gas to rotary air preheater is cooled from 392 °C to 160 °C while by-passed flue gas is from 392 °C to 210 °C. The two flue gases are mixed to indirect air preheating unit and cooled to 123 °C. The power increase is: $\Delta H_\mathrm{r} = \eta_1 \Delta Q_1 + \eta_2 \Delta Q_2 + \eta_3 \Delta Q_3 = 5680.20$ kW.

(4) *Additional heating surface of economizer*

Flue gas energy is recovered by an additional economizer, which is used for H1 regenerator, and its temperature drops to 361 °C before entering rotary air preheater, as shown in fig. 6(d). Flue gas is cooled to 218 °C in the air preheater and then enters the indirect air preheating unit. The power increase is: $\Delta H_\mathrm{a} = \eta_1 \Delta Q_1 = 6374.3$ kW.

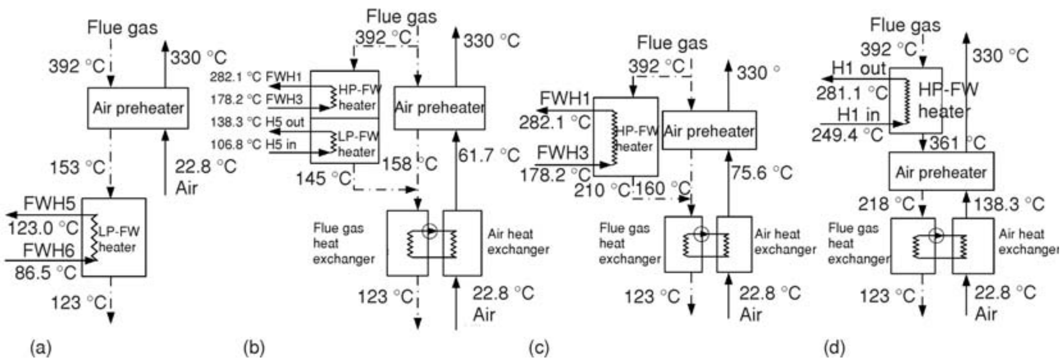

Figure 6. (a) scheme 1: direct utilization of flue gas energy; (b) scheme 2: indirect flue gas heat recovery system; (c) scheme 3: improved indirect flue gas heat recovery system; (d) scheme 4: additional heat surface of economizer

Cost analysis

The proposed waste heat cascading utilization schemes, from fig. 6, can bring about an increase of power and efficiency, but the heat transfer surface increment certainly lead to more investment cost. In order to analyze which scheme is more profitable or which scheme is the best option, an economic analysis of the different schemes is mandatory.

Based on the quantity of heat exchange in each heat exchanger, the required heat transfer area of the added heat exchangers can be calculated. Then the additional cost is estimated proportionally. The cost of heat exchangers in the different configurations is ΣC_{HEX}. In the scheme (a), there is only a LP-FW heater ($\Sigma C_{\mathrm{HEX}} = C_{\mathrm{LP}}$). In the scheme (b), there is the indirect system includes a HP-FW heater, a LP-FW heater and an indirect air preheating unit ($\Sigma C_{\mathrm{HEX}} = C_{\mathrm{HP\text{-}FW}} + C_{\mathrm{LP\text{-}FW}} + C_{\mathrm{F\text{-}HEX}} + C_{\mathrm{A\text{-}HEX}}$). In the scheme (c),there is the indirect system made of a HP-FW heater and an indirect air preheating unit ($\Sigma C_{\mathrm{HEX}} = C_{\mathrm{HP\text{-}FW}} + C_{\mathrm{F\text{-}HEX}} + C_{\mathrm{A\text{-}HEX}}$). In the scheme (d), there is additional economizer ($\Sigma C_{\mathrm{HEX}} = C_{\mathrm{HP\text{-}FW}} + C_{\mathrm{F\text{-}HEX}} + C_{\mathrm{A\text{-}HEX}}$).

The total cost can be estimated:

$$C_{\mathrm{total}} = \Sigma C_{HEX} + \Sigma C' \tag{4}$$

where $\Sigma C'$ refers to the other expenses include material costs, design fees, construction costs, and gross profit. The values and units for all the parameters used in cost analysis can be seen in tab. 5.

Additional incomes of the power plant are related to the power output increase:

$$\Delta I = \Delta H h_{\mathrm{eq}} P_{\mathrm{e}} \tag{5}$$

where ΔH is the power increase, h_{eq} – the equivalent operating hours per year, and P_{e} – the electricity price. The values are also showed in tab. 5.

The static investment payback period P can be expressed:

$$P = \frac{C_{\mathrm{total}}}{\Delta I} \tag{6}$$

Table 5. Data for the cost analysis

Parameters	Indirect utilization scheme	Cascading utilization schemes	Unit
Heat transfer coefficient	50	50	W/m^{2}K
Steel unit price of feed water heater exchangers	15000	15000	¥/t
Steel unit price of indirect air preheating unit	–	10000	¥/t
Material cost	1.7	5	Million ¥
Design fee	0.3	1	Million ¥
Construction cost	0.5	2	Million ¥
Gross profit	3	10	Million ¥
h_{eq}	6000	6000	h/year
P_{e}	0.43	0.43	¥/kwh
CNY to dollar conversion	0.1627	0.1627	¥/$

Results and discussion

Table 6 shows the calculation results of the four schemes from fig. 6. Table 7 shows the cost analysis results of different schemes. The flue gas waste energy is recovered by nearly 30 °C in all the four schemes, but the increments in power and power plant efficiency are not the same. The performance of indirect utilization scheme is much better than that of direct utilization scheme. Compared with scheme1, scheme 2, 3, and 4 make use of energy in stages according to the level of flue gas temperature, realizing energy cascading utilization. The additional economizer scheme, *i.e.* scheme 4, has the best results. It generates extra power of 6.37 MW$_e$ and higher plant efficiency of 1.89%. Heat recovered by additional economizer saves the extraction of the highest pressure steam, which has the highest entropy drop in steam turbine compared with the other steam extractions.

The proposed waste heat cascading utilization schemes have no affect on heat transfer distribution, hydrodynamic flow, and heat transfer in the boiler. Temperatures of the primary and the secondary air from rotary air preheater can be kept the same as before. The exhaust flue

Table 6. Results for four schemes

Scheme	Power increase [kW]	Equivalent enthalpy increase per kg main steam, ΔH [kW/kg]	Efficiency increase Δh [%]	Decrease of standard coal consumption Δb [g/(kWh]	Allocated feed water [t/h]
Scheme 1	2455.30	8.70	0.74	2.36	289.55
Scheme 2	5054.40	17.90	1.50	4.81	72.90
Scheme 3	5680.20	20.12	1.69	5.4	93.48
Scheme 4	6374.30	22.57	1.89	6.05	281.32

Table 7. Cost analysis results

Parameter	Scheme 1	Scheme 2	Scheme 3	Scheme 4	Unit
Area of feed water heat exchanger	7436.4	3632.6	3913.6	2225.5	m^2
Area of the flue gas heat exchanger	–	6088.6	6442.4	18213.5	m^2
Area of air heat exchanger	–	5531.6	5877.9	19162.6	m^2
ΣC_{HEX}	4.20	7.85	8.35	13.90	million¥
ΣC'(other expenses)	5.5	18	18	18	million¥
C_{total}	9.70	25.85	26.35	31.90	million¥
C_{total} ($)	1.58	4.20	4.29	5.19	million$
ΔI	6.34	13.04	14.66	16.45	million¥
ΔI ($)	1.03	2.12	2.38	2.68	million$
P	1.53	1.98	1.80	1.94	year

gas temperature can be reduced as needed, a drop of 20~30 °C or higher. However, due to the enlargement of heat transfer surface of indirect air preheating unit, air pressure drop becomes a little bigger. As to the flue gas, the pressure drop must not get bigger because a fraction of flue gas is bypassed from the inlet of rotary air preheating unit and the flue gas velocity decreases in it. Because air is pre-heated before entering rotary air preheater, air heater can be omitted and the low temperature corrosion in the air preheater is avoided by using indirect air preheating unit. In addition, thermal deformation of rotary air preheater gets alleviated and the air leakage is reduced due to the temperature difference between cold end and hot end of rotary air preheater is diminished.

The cost analysis shows that the direct utilization of flue gas energy is at least investment. The cascading utilization will add investment cost because of heat transfer surface increment. Compared to scheme 2, scheme 3 has the advantage in the benefits. Scheme 4 has the largest heat transfer surface, so it has the largest investment cost. However, the investment payback period of scheme 4 only a litter longer than scheme 3, therefore scheme 4 is still one of the best choices in the long term.

Conclusions

The proposed improved indirect flue gas heat recovery system and additional economizer scheme obtain higher plant efficiency. Compared to direct flue gas heat recovery, the proposed schemes make use of energy in stages according to the level of flue gas temperature. Increments in power plant efficiency and power are obviously raised. The main conclusions drawn from this work are:

- the recovery of flue gas waste heat in the power plant can reduce flue gas temperature and lead to an increase of power plant efficiency,
- four schemes are integrated with a 330 MW coal-fired unit to reduce flue gas temperature from 153 °C to 123°C; a maximum efficiency increment of 1.89% is obtained for the proposed schemes. The direct utilization of flue gas energy has only an increment of 0.74% in efficiency and that of the indirect flue gas heat recovery system is 1.5%, and
- the initial investment of cascading utilization schemes is larger than that of direct utilization of flue gas energy, but they have higher revenue and are worthwhile.

Acknowledgment

This work was carried out with a financial grant from National High Technology Research and Development Program of China (2012AA051801) and the National Natural Science Foundation of China (51176033).

Nomenclature

C_{HEX} – heat exchanger cost
$C_{A\text{-}HEX}$ – air heater capital cost
$C_{F\text{-}HEX}$ – flue gas heater capital cost
$C_{HP\text{-}FW}$ – HP feed water heater capital cost
C_{LP} – LP heater capital cost
$C_{LP\text{-}FW}$ – LP feed water heater capital cost
C_{total} – total cost
ΔH – enthalpy drop, [kJkg^{-1}]
HHV – higher heating value, [kJkg^{-1}]
h_{eq} – equivalent hours
LHV – lower heating value, [kJkg^{-1}]
P_e – electricity price
P – investment payback period
Q – heat, [kJkg^{-1}]
ΔI – additional incomes per year of new configuration
η – efficiency

Subscript

ar – as received

Acronyms

EEDM – equivalent enthalpy drop method
FW – feed water
FGD – flue gas desulfurization
LP – low pressure
HP – high pressure

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Paper submitted: January 4, 2014
Paper revised: August 22, 2014
Paper accepted: September 1, 2014