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Parameter Optimization and Economic Performance Comparison of Single-Pressure Steam Cycle, Steam Flash Cycle and Dual-Pressure Steam Cycle for Waste Heat Recovery

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ABSTRACT

This paper focuses on the parameter optimization and economic performance comparison of single-pressure steam cycle (SRC), dual-pressure steam cycle (SDC) and steam flash cycle (SFC) based on the waste heat source with initial temperature of 400-700℃ and mass flow rate of 40kg/s. We take the heat exchanger area per unit power output (APR) as the economic evaluation index and use genetic algorithm (GA) for optimization. The derivation of the formulas shows that as the pinch point temperature difference between the flue gas and working fluid increases, the APRs of SRC, SFC and SDC decrease accordingly. The APRs of SFC and SDC are higher than that of SRC, which is due to the fact that the effect of the increase of total heat exchanger area caused by the SDC and SFC is greater than that of the increase of net power output. With the increase of the initial temperature of flue gas, the optimum APRs of SRC, SFC and SDC decrease accordingly and the degree of decline gradually decreases. When the initial temperature of flue gas reaches 700℃, the optimum APRs of SRC and SFC increase slightly.

Keywords: waste heat recovery; economic performance optimization; performance comparison; APR

1. INTRODUCTION

A great deal of waste heat energy with exhaust gas as the carrying medium is available in the industrial processes [1]. The waste heat recovery in an economic and feasible way is beneficial to alleviating the energy shortage and improving the comprehensive utilization rate of energy.

It showed that the power cycle configuration adopted by the flue gas waste heat power recovery system is mainly based on the Rankine cycle and water is used as the working fluid (Steam-Rankine cycle, SRC) [2]. SRC has good economic performance when it is used in large capacity flue gas waste heat source with high initial temperature [3].

As a configuration of power cycle, SRC has many advantages. For example, the Rankine cycle is the most widely used classic steam power cycle; Water has excellent thermal stability and excellent transport properties; The SRC system is technologically mature and has efficient and reliable components.

However, there are still some disadvantages. From the perspective of thermodynamics, the constant pressure heat absorption process of RC includes the constant temperature evaporation phase change section of the working fluid, which leads to an inherent and non-zero minimum heat transfer temperature difference between the flue gas and working fluid. In the process of heat transfer between flue gas and working fluid, the distribution of heat transfer temperature difference is obviously uneven, and the thermodynamic perfection and waste heat utilization of the cycle are low.

At present, the improved cycles based on the SRC mainly include the steam flash cycle

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(SFC) and the dual-pressure steam cycle (SDC). The applications of SFC and SDC help to improve the matching between the exhaust gas and the working fluid in the heat addition process, which can effectively reduce the exhaust temperature of the flue gas and increase the utilization rate of waste heat. The net power outputs of SFC and SDC are greater than that of SRC, but the logarithmic mean temperature difference between the flue gas and the working fluid is reduced, which will result in a significant increase in the heat exchanger area of the system for a given heat load and increase the cost of the system consequently.

The purpose of this paper is to investigate and compare the economic performance of SRC, SFC and SDC with the heat exchanger area per unit power output (APR) as the economic evaluation index based on the medium and high temperature exhaust gas waste heat source. The effect of parameters on the performance of the cycle is analyzed and the parameter optimization is performed by means of genetic algorithm to minimize the APR.

2.SYSTEM DESCRIPTION

The schematic and T-s diagram of single-pressure steam cycle are shown in figure 1 and figure 2.The high-pressure and high-temperature vapor generated in heat recovery steam generator (HRSG) expands through the turbine to output the power (1–2). The turbine exhausts are condensed to saturated liquid in condenser (2–3). The saturated liquid at the condenser outlet is pressurized adiabatically by the pump (3–4) and then flows into the HRSG to absorb the heat from exhaust gas (4–1) and a new cycle begins.

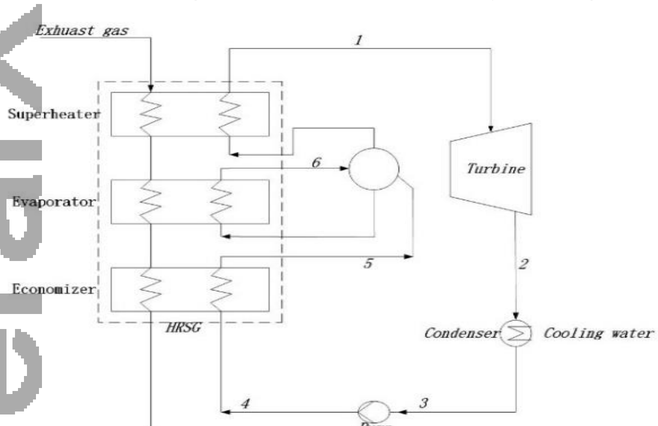

Fig. 1. Schematic diagram of single-pressure steam cycle

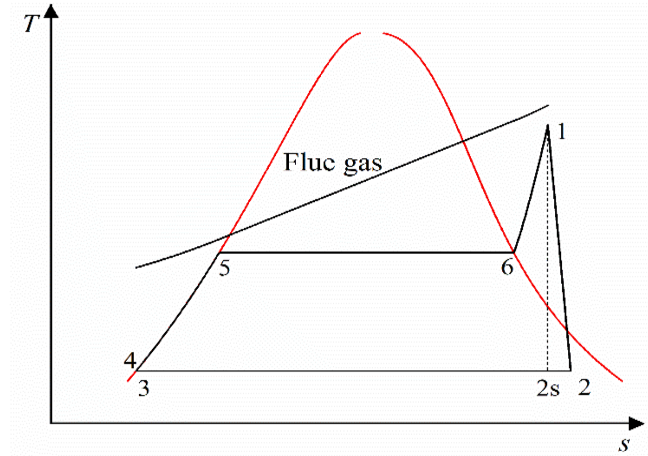

Fig. 2. T-s diagram of single-pressure steam cycle

Figure 3 and figure 4 show the schematic diagram of SFC and SDC. For SFC, a portion of saturated water from the high-pressure economizer expands in the flasher (7–8) into saturated vapor (10) and saturated liquid (9) of lower pressure. The vapor is sent to the turbine 2 to generate power (10-11) and the liquid is mixed with water from low-pressure economizer and reenter the high-pressure economizer (6-7) after being pressurized by the high-pressure pump (5,9-6). The SDC adds a set of low-pressure water circuit in the HRSG, which has a lower evaporating temperature and can effectively reduce the flue gas exhaust temperature and increase the heat absorption of the system.

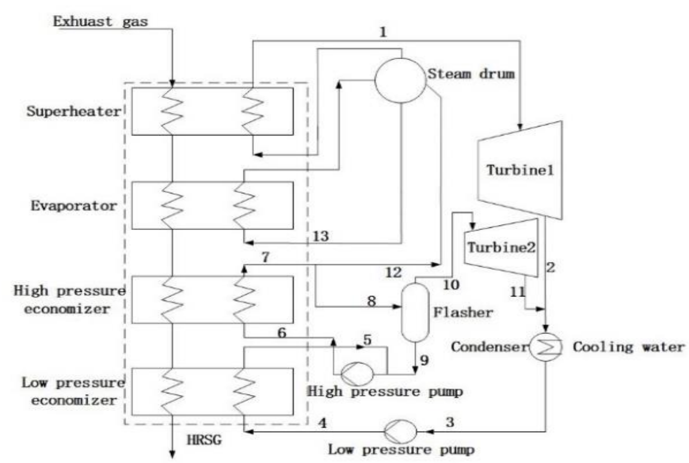

Fig. 3. Schematic diagram of steam flash cycle system

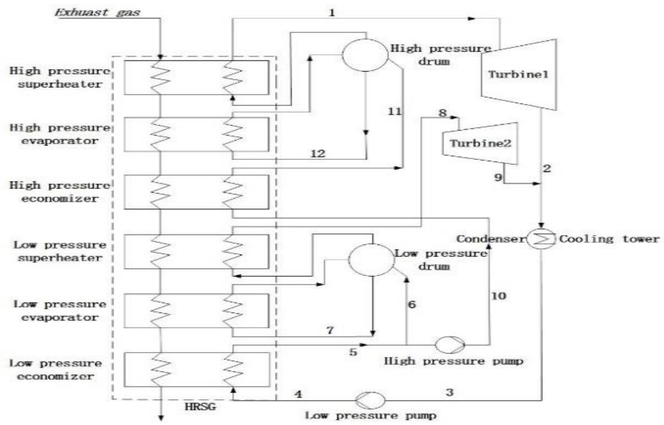

Fig. 4. Schematic diagram of dual-pressure steam cycle system

3. OPTIMIZATION AND ASSUMPTION

In order to simplify the theoretical model, the following assumptions are made:

(1)The system operates in a steady state; (2)The heat loss and pressure drop on the components and pipelines of the system are ignored; (3)The isentropic efficiency of the turbine and pump are set to a fixed value; (4)The kinetic energy and potential energy changes of the working fluid in each component are ignored.

The design of heat exchanger and corresponding heat transfer correlations used have been described in our previous work [4]. The research shows that the H_2O-RC usually operates at temperatures above 673K [5]. Therefore, in this paper, the medium and high temperature flue gas of 400-700℃ is used as the waste heat source. The mass flow rate of flue gas is set as 40kg/s, and the specific heat capacity of flue gas is set as 1.1kJ/(kg·K). Moreover, concerns have been expressed when steam temperatures exceed 627℃ due to the material limitations [6], hence the maximum temperature of the cycle is not more than 600℃. Some other assumptions are summarized in table 1.

Table 1. Main assumptions for the cycles

Parameters	Unit	Value
Turbine isentropic efficiency	%	75
Pump isentropic efficiency	%	80
Pinch point temperature difference in HRSG	℃	⩾15
Approach temperature difference	℃	5
Temperature of exhaust gas	℃	⩾120
Minimum degree of dryness of turbine exhaust	%	88

Based on the MATLAB platform, the genetic algorithm (GA) is selected as the optimization method to obtain the optimum APR and the system parameters accordingly. The key control parameters of genetic algorithm are as follows: The number of individuals is 100; The number of subpopulations is 5; The crossover probability is 0.95; The mutation probability is 0.07; The minimum number of generation limit is 300.

APR is the result of the combined effect of net output power and total heat exchanger area. The corresponding formulas are as follows:

$$APR = \left(A_{CON} + A_{HRSG}\right) / W_{\mathrm{net}} \tag{1}$$

$$APR_1 = \frac{A_{HRSG}}{W_{\mathrm{net}}} = \frac{\dfrac{Q_1}{K_1 \cdot LMTD_1}}{\eta \cdot Q_1} = \frac{1}{K_1 \cdot LMTD_1 \cdot \eta} \tag{2}$$

$$APR_2 = \frac{A_{CON}}{W_{\mathrm{net}}} = \frac{\dfrac{Q_2}{K_2 \cdot LMTD_2}}{\eta \cdot Q_1} = \frac{\dfrac{Q_1 \cdot (1-\eta)}{K_2 \cdot LMTD_2}}{\eta \cdot Q_1} = \left(\frac{1}{\eta} - 1\right) \cdot \frac{1}{K_2 \cdot LMTD_2} \tag{3}$$

For APR_1, with the increase of the pinch point temperature difference in HRSG, the physical properties of the flue gas will change a little, which makes the heat transfer coefficient K_1 between flue gas and working fluid increase slightly. The logarithmic mean temperature difference between flue gas and working fluid increases with the increase of the pinch point temperature difference. The thermal efficiency remains unchanged. So APR_1 will decrease; For APR_2, the logarithmic mean temperature difference between working fluid and cooling water remains unchanged, and the heat transfer coefficient K_2 basically stays the same, so it can be considered that APR_2 remains unchanged.

Therefore, APR will decrease with the increase of the pinch point temperature difference under the combined effect of APR_1 and APR_2. It reminds us that no matter which cycle configuration is considered, the APR of the system will decrease with the increase of the pinch point temperature difference in HRSG accordingly. In order to compare the SRC, SFC and SDC, the pinch point temperature difference in HRSG is fixed at 15℃.

The optimization variables of SRC are evaporating temperature, turbine inlet temperature and condensing temperature. In addition to above parameters, SFC also includes flash temperature, mass flow ratio of the flash stream to the main stream (MFR). Compared with SRC, the optimization variables of SDC also include low-pressure evaporating temperature and low-pressure turbine inlet temperature.

4. RESULT AND DISCUSSION

Figure 3 shows the change of APR of SRC with evaporating temperature and condensing temperature when the initial temperature and mass flow rate of flue gas are 400℃ and 40kg/s respectively. The inlet temperature of turbine is set to 385℃.

Given the condensing temperature, with the increase of evaporating temperature, although the logarithmic mean temperature difference between flue gas and working fluid in HRSG decreases, the influence of the decrease of heat load in HRVG caused by the rise of flue gas exhaust temperature will be greater. The combined effect of the two will lead to the reduction of heat exchanger area in HRSG. The increase of evaporating temperature causes the decrease of the heat load of the condenser, so the heat exchanger area of the condenser decreases accordingly. Therefore, the total heat exchanger area decreases. The net power output increases at first and then decreases with the increase of evaporating temperature. Since the effect of the reduction of the heat exchanger area is greater than the effect of the change of the net power output, the APR decreases with the increase of the evaporating temperature.

Given the evaporating temperature, the net power output of the system decreases with the increase of the condensing temperature. The increase of the average exothermic temperature reduces the heat exchanger area in the condenser, which reduces the total heat exchanger area. The APR first decreases and then increases with the combined effect of the net power output and the total heat exchanger area.

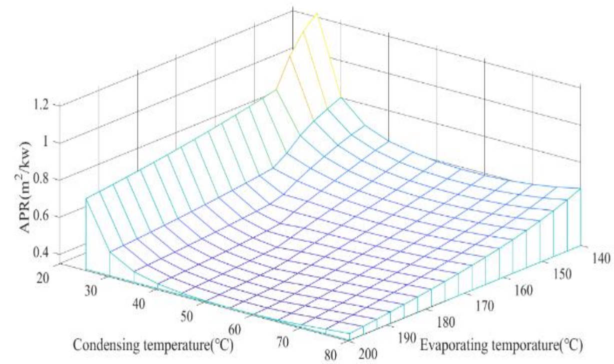

Fig. 3. Effect of the evaporating temperature and condensing temperature on the APR of SRC

Figure 4 shows the effect of the MFR on SFC. The steam flash cycle can reduce the flue gas exhaust temperature, sacrifice part of the thermal efficiency to improve the waste heat utilization and the thermodynamic performance of the system. Meanwhile, it also increases the total heat exchanger area of the system. Obviously, it can be seen that APR reaches the optimum value when the MFR is 0, indicating that the effect of the increase of the total heat exchanger area caused by the flash process is greater than the effect of the increase of the net power output, which leads to the increase of APR.

Therefore, under the condition of the flue gas heat source given in this paper, when APR is used as the economic evaluation index, SFC is not conducive to the economic performance of the system compared with SRC.

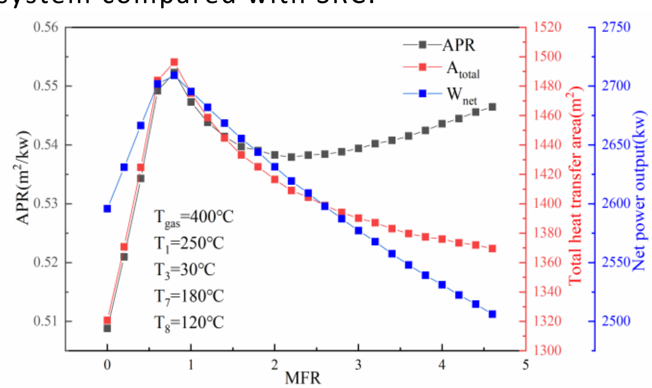

Fig. 4. Effect of mass flow ratio on the system performance of SFC

Figure 5 shows the effect of the low-pressure evaporating temperature $T_{eva,LP}$ and the turbine 2 inlet temperature TIT_{LP} on the APR of the SDC. Given the $T_{eva,LP}$, APR increases gradually with the increase of TIT_{LP}. It can be seen that when TIT_{LP} is equal to $T_{eva,LP}$ (nonoverheated), APR reaches the minimum value. Given the TIT_{LP}, APR first decreases and then increases with the increase of the $T_{eva,LP}$ and there is an optimum APR.

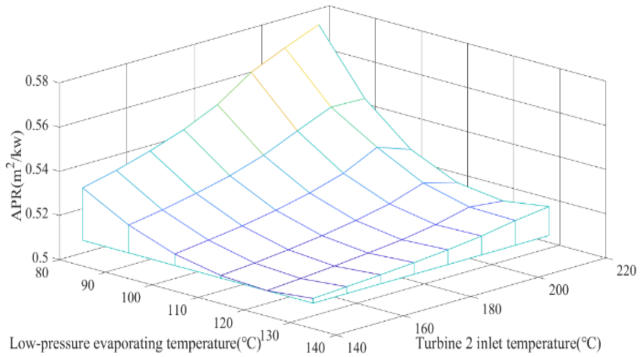

Fig. 5. Effect of the low-pressure evaporating temperature and turbine 2 inlet temperature on the APR of SDC

Table 2 and table 3 show the optimum APRs and corresponding parameters of SRC and SDC when the initial temperature and mass flow rate of flue gas are 400℃ and 40kg/s respectively. The MFR is 0 when SFC reaches the optimum APR, so the SFC is equivalent to the SRC. It can be seen that SDC can reduce the flue gas exhaust temperature $T_{g,out}$ and increase the waste heat utilization of the system Φ and the net power output W_{net}, but the total heat exchanger area A_{total} also increases. The effect of the increase of heat exchanger area is greater than the effect of the increase of the net power output. Therefore, the APR of SDC is higher than SRC, which indicates that the economic performance of SDC is poor.

Table 2. Optimized variables and the system performance of the SRC (and SFC)

$T_{g,out}$(℃)	η(%)	W_{net}(kW)	A_{total}(m^2)	Φ(%)	APR(m^2/kW)
234.79	25.18	1830.53	533.75	59.00	0.29

Table 3. Optimized variables and the system performance of the SDC

$T_{g,out}$(℃)	η(%)	W_{net}(kW)	A_{total}(m^2)	Φ(%)	APR(m^2/kW)
183.11	22.70	2166.35	786.81	77.46	0.36

Figure 7 shows the change of the optimum APRs of the SRC, SDC and SFC with the increase of the initial temperature of flue gas. It can be seen that the APR of SFC is always equal to that of SRC, and the APR of SDC is always higher than that of SRC. This indicates that SFC and SDC can't improve the economic performance of the system than SRC when APR is used as the economic evaluation index. The reason is that although SFC and SDC can improve the thermodynamic performance of the system, the heat exchanger area of the system also increases and has a greater impact on APR. Therefore, it is not conducive to the economic performance of the system.

At the same time, it can be seen that with the increase of the initial temperature of flue gas, the optimum APRs of the three cycles decrease accordingly, and the reduction gradually becomes smaller. When the initial temperature of flue gas reaches 700℃, the optimum APRs of SRC and SFC increase slightly.

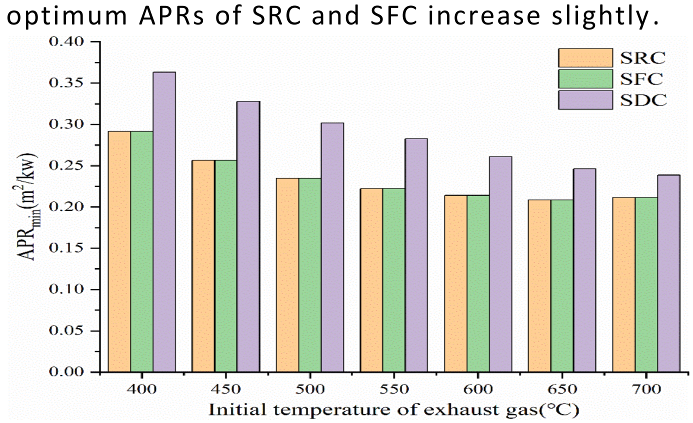

Fig.6. The optimum APRs of SRC, SFC and SDC under the various initial temperature of exhaust gas

5. CONCLUSION

This study focuses on the parameter optimization and economic performance comparison of SRC, SFC and SDC based on the waste heat source with initial temperature of 400-700℃ and mass flow rate of 40kg/s and uses genetic algorithm to optimize the APRs of the three cycles. Main conclusions are summarized as follows.

(1) The APR of SRC decreases with the increase of evaporating temperature, decreases at first and then increases with the increase of condensing temperature.

(2) No matter which cycle configuration is considered, the APR of the system will decrease accordingly with the increase of the pinch point temperature difference in HRSG.

(3) Although the net power outputs of SFC and SDC are higher than SRC, their APRs are also higher than SRC.

(4) With the increase of the initial temperature of flue gas, the optimum APRs of SRC, SFC and SDC decrease accordingly and the degree of decline gradually decreases. When the initial temperature of flue gas reaches 700℃, the optimum APRs of SRC and SFC increase slightly.

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REFERENCE

[1] Bundela, P.S.; Chawla, V. Sustainable development through waste heat recovery. Am. J. Environ. Sci. 2010, 6, 83–89.

[2] Tianyou Wang, Yajun Zhang, Zhijun Peng, Gequn Shu. A review of researches on thermal exhaust heat recovery with Rankine cycle. Renewable and Sustainable Energy Reviews 2011; 15: 2862-2871.

[3] Hung TC, Shai TY, Wang SK, A review of organic Rankine cycles (ORCs) for the recovery of low-grade waste heat, Energy, 22(7):661-667,1997

[4] Ren L, Liu J, Wang H. Thermodynamic optimization of a waste heat power system under economic constraint. energies, 2020, 13.

[5] Chan C W, Ling-Chin J, Roskilly A P. A review of chemical heat pumps, thermodynamic cycles and thermal energy storage technologies for low grade heat utilisation. Applied Thermal Engineering, 2013.

[6] Dostal, Vaclav. A supercritical carbon dioxide cycle for next generation nuclear reactors[J]. Massachusetts Institute of Technology, 2005, 154(3): 265-282.