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Comparative study of waste heat steam SRC, ORC and S-ORC power generation systems in medium-low temperature

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ABSTRACT

For Steam Rankine Cycle (SRC), Organic Rankine Cycle (ORC) and Steam-Organic Rankine Cycle (S-ORC) power systems, in this paper, mathematical models are developed to explore the feasibility that combines the fluid-low temperature (150-350 °C) waste heat steam and low-boiling point organic working fluids for power generation. Using the numerical models, we calculate and compare thermal efficiency, exergy efficiency, operation pressure, generating capacity, etc. of three power systems, namely SRC, ORC and S-ORC under the same heat source conditions. The results show that under the condition of 150-210 °C heat source, ORC has the highest thermal efficiency, exergy efficiency and power generation; while at 210-350 °C, the performance of the S-ORC has a distinct advantage. Its thermal efficiency and exergy efficiency are higher than those of the SRC and ORC power systems.

HIGHLIGHTS

SRC, ORC and S-ORC three kinds of power systems are computed and compared under the condition of the same heat source.

- For low enthalpy heat, ORC is more efficient than SRC in recovering the latent heat of condensation of steam.
- S-ORC power generation system has a better matching of heat source temperature.
- The ORC is optimal under the condition of 150 °C to 210 °C, while S-ORC has obvious advantages under the condition of 210 °C to 350 °C.

Keywords: medium-low temperature waste heat steam; Rankine cycle; comparison of power generation systems

1. Introduction

Waste heat steam is one of the important forms of waste heat emission which is widely used in industrial enterprises such as power plants and iron and steel
enterprises. Varieties of waste heat resources account for 68% of all production energy consumption [1], but the utilization rate of waste heat is only 32%, of which the low temperature waste heat utilization rate is almost zero. If we could utilize and recycle this part of the energy, it will not only solve the energy problem of our country, but also reduce the environmental pollution in the process of energy production. Based on the status of low waste heat utilization [2-3], energy recycling technology in fluid-low temperature has become a research focus increasingly in the related fields.

At present, there are mainly two forms of waste heat recovery, heat to heat recovery and heat to electricity recovery. The heat energy, recycled by the first form is not easy to store, and was limited by the transmission distance, so it is not usually adopted. The second form primarily has three forms, steam Rankine cycle (SRC), organic Rankine cycle (ORC), Steam-Organic Rankine cycle (S-ORC) [4]. Heat to electricity will convert low-grade thermal energy into electrical energy, which is mainly used in solar thermal power, industrial waste heat power generation, geothermal power generation, biomass power generation, ocean thermal power generation and so on.

Traditional SRC employs water and high pressure steam as the circulating working fluid and the power generation technology which is mainly appropriate for high temperature heat source like T > 500°C is relatively mature. Due to the water source is rich, with the big latent heat of vaporization, high boiling point, large specific heat capacity, non-toxic and tasteless, no polluting to the environment, so it's widely used in thermal engineering. Although SRC is the most common technology in the heat to electricity recovery process, it is not suitable for low temperature and pressure condition due to the necessity of high operational temperature and pressure and when the exhaust temperature and pressure are same, the exhaust steam enthalpy of SRC is larger, which increases the heat of the cold source.

ORC adopting low boiling point organic pure or mixed working fluid is generally preferred for the processes having low temperature like T < 150 °C, so it can recover low temperature heat energy of different temperature range. The main advantages of ORC are high thermal efficiency, simple and compact system structure, widely used heat source, and availability in recovering low temperature heat energy [5]. At present, the research on the ORC power system mainly focuses
on the following aspects: thermodynamic properties and environmental performance of working fluid, the application of mixed working fluid, and the optimization of thermodynamic cycle, etc. [6-8]. The ORC researches carried out by scholars at home and abroad are still in the theoretical research and small-scale experimental stage and the engineering application is rare [9-11]. V. Maizza and A. Maizza [12] considered 24 working fluids, including eight zeotropic blends and one azeotropic blend, for waste heat recovery ORC applications. Chen H.J. et al. [13] proposed to use zeotrope as working fluid to recover waste heat in the supercritical cycle. The results showed that when the heat source temperature is 120-200°C, the system thermal efficiency of the super critical cycle adopting zeotrope improved 10.8%-13.4% than subcritical cycle adopting pure working fluid. Meanwhile, the exergy loss has decreased by 14.6%.

S-ORC employs steam as high temperature working fluid and organic working fluid as low temperature working fluid and couples together cycles of different operating temperature range in an appropriate way to form a combined cycle, which can expand the working temperature range of the circulation, and effectively improve the circulating heat efficiency. The thermal efficiency of combined cycle system is higher than that of the single cycle system, and the S-ORC could utilize energy comprehensively in accordance with quality and quantity which embodies the idea of making use of energy according to the grade. For S-ORC system, L. H. Zhang et al. made a research and analysis in the literature [14], indicating that it is feasible to take advantage of waste heat for the S-ORC power generation system adopting steam with large latent heat of vaporization and organic working fluid with small latent heat of vaporization. S-ORC not only realizes the ladder utilization of energy, but also could recover the waste heat below 200°C which is difficult to recover. J. Huang et al. [15] focused on saturated steam at 160°C of rolling mill, by using combined double cycles waste heat power generation system, the total power generation efficiency is up to 5.4%. In order to reduce the exhaust steam of steam turbine in thermal power plant, H.J.Li et al. [16] took NZK660-24.2/566/566 unit as an example, established the model of steam and organic working fluid combined cycles and carried out thermal economic analysis. The method proposed by SRC shares some similarities with gas steam turbine combined cycle, however, there are still different points between the two cases. At present, system and detailed
introduction to such combined cycle in domestic and foreign information can’t be found.

In this paper, we focus on the application of waste heat at the temperature range of 150-350°C. Primarily, we confirmed the design conditions of the power generation system at each temperature stage, according to the principle of cyclic operation, equipment requirements and environmental factors. Secondly mathematical models were established by employing the VBA module of Excel, meanwhile we called working fluid property query libraries REFFPROP 8.0 and simulated the thermal efficiency, energy efficiency and power generation of each power generation system under the same heat source condition. Then, based on the system thermal efficiency, exergy efficiency and irreversible loss, the optimum working fluid of each temperature stage is selected. Ultimately, we compare the different systems under different temperature stage to probe the suitable power generation system. In addition, we compare the simulation data with the experimental data to verify the relevant conclusions of the simulation. By means of the above research, not only certain bases for large-scale application of fluid-lows temperature power generation technology is provided, but also new ideas about waste heat utilization are expounded.

| Nomenclature       | P     | generating capacity of unit working fluid [kJ/kg] |
|---------------------|-------|--------------------------------------------------|
| Symbols             | w_i   | the specific power of working fluids in turbine [kJ/kg] |
| Q                   | \(\eta_g\) | the efficiency of generator [%] |
| W                   | \(\eta_t\) | turbine isentropic efficiency [%] |
| E                   | \(\eta_r\) | pump isentropic efficiency [%] |
| \(E_w\)             | \(\eta_c\) | cycle thermal efficiency [%] |
| \(E_{ex}\)          | \(\eta_{ex}\) | cycle exergy efficiency [%] |
| \(m_s\)            | in    | at inlet |
| \(m_f\)            | out   | at outlet |
| \(h\)              | L     | cold source |
| \(s\)              | H     | heat source |
| T                   | L1    | cold source of first stage cycle |
|                     | L2    | cold source of second stage cycle |

\(\eta_g\) \(\eta_t\) \(\eta_r\) \(\eta_c\) \(\eta_{ex}\) \(m_s\) \(m_f\) \(h\) \(s\) \(T\)
2. System model

2.1. Working principle and mathematical models of SRC and ORC

Rankine cycle waste heat power generation system consists of three subsystems: the heat source (middle-low temperature waste heat steam) system, Rankine cycle system and cooling source (cooling water) system. The basic principle and process of the ORC system and the traditional SRC system are the same, but the difference lies in that the ORC system employs low-boiling point organic working fluid instead of water and high pressure steam. Fig.1 and Fig.2 represent the schematic diagram of two cycles, while Fig.3 and Fig.4 are Temperature-Entropy diagrams, the working processes are as follows.

State 1 to 2 represents an adiabatic expansion process, high temperature and high pressure fluid (state 1) enters in the turbine, and it leaves from turbine as low pressure fluid and generates electricity. The heat loss of working fluid can be neglected, so the expansion process is regarded as an adiabatic process.

State 2 to 3 represents an isobaric exothermic process. Exhausted steam comes out of the turbine after doing work (state 2) and releases heat to cooling water isobarically in the condenser, then it becomes saturated water after condensation (state 3), which is a process of constant pressure and constant temperature.

State 3 to 4 represents an adiabatic compression process. Liquid working fluid after condensation is pressurized in the working fluid pump. Ignore the heat loss, the compression process can be regarded as an adiabatic process.

State 4 to 1 represents an isobaric endothermic process. Cold working fluid...
(state 4) enters the evaporator in the function of the working fluid pump, exchanges heat with waste heat steam at constant pressure. The working fluid experiences three stages: preheating, evaporating, over heat, and then complete the heat absorption process at constant pressure.

![Fig.1. Schematic of SRC cycle.](image1)

![Fig.2. Schematic of ORC cycle](image2)

![Fig.3. T-S diagram of SRC cycle.](image3)

![Fig.4. T-S diagram of ORC cycle](image4)

The following formulas can be obtained according to the parameters of the working state in the T-S diagram.

State 1 to 2 is an adiabatic expansion process of working fluid in the turbine, the ideal reversible adiabatic expansion process is regarded as an isentropic process, but the turbine efficiency $\eta_t$ needs to be considered in the actual calculation process, then the output power of working fluid in the turbine is $W_t$:
\[ W_t = m_f (h_1 - h_2) \eta_t \]  

(1)

The irreversible loss of the turbine is \( I_t \):

\[ I_t = m_f T_0 (s_2 - s_1) \]  

(2)

State 2 to 3 is an isobaric exothermic process of working fluid in the condenser, the heat released is \( Q_c \):

\[ Q_c = m_f (h_2 - h_3) \]  

(3)

The irreversible loss of the condenser is \( I_c \):

\[ I_c = m_f T_0 (s_3 - s_2 - \frac{h_3 - h_2}{T_L}) \]  

(4)

Here, for the cold source whose temperature is changeable, the temperature of the cold source is \( T_L \):

\[ T_L = \frac{T_{L_{out}} - T_{L_{in}}}{\ln\left(\frac{T_{L_{out}}}{T_{L_{in}}}\right)} \]  

(5)

Where \( T_{L_{in}} \) and \( T_{L_{out}} \) are the temperature of the cooling water at the condenser inlet and outlet, K.

State 3 to 4 is an adiabatic compression process in the working fluid pump, the ideal reversible adiabatic compression process is considered as an isentropic process, but the turbine efficiency \( \eta_t \) needs to be considered in the actual calculation process, and the output power of the working fluid in the turbine is \( W_p \):

\[ W_p = \frac{m_f (h_4 - h_3)}{\eta_p} \]  

(6)

The irreversible loss of the working fluid pump is \( I_p \):

\[ I_p = m_f T_0 (s_4 - s_3) \]  

(7)

State 4 to 1 is an isobaric endothermic process in the evaporator. The heat absorbed in the evaporator is \( Q_e \):

\[ Q_e = m_f (h_1 - h_4) \]  

(8)

The irreversible loss of the evaporator is \( I_e \):

\[ I_e = m_f T_0 (s_1 - s_4 - \frac{h_1 - h_4}{T_H}) \]  

(9)
Here, for the heat source whose temperature is changeable, the temperature of it is $T_H$:

$$T_H = \frac{T_{H\text{ in}} - T_{H\text{ out}}}{\ln\left(\frac{T_{H\text{ in}}}{T_{H\text{ out}}}\right)} \quad (10)$$

Where $T_{L\text{ in}}$ and $T_{L\text{ out}}$ are the temperature of the hot fluid at the evaporator inlet and outlet, K.

The power generation capacity of unit quality working fluid is $P$:

$$P = \frac{E}{m_f} = W_t \eta_s \quad (11)$$

Where $P$ is system power generation capacity, kW; $W_t$ is the specific work of working fluid, kJ/kg; $\eta_s$ is the generator efficiency, %.

The thermal efficiency of the system is $\eta_e$:

$$\eta_e = \frac{W_t - W_p}{Q_e} \quad (12)$$

The exergy efficiency of the system is $\eta_{ex}$:

$$\eta_{ex} = \frac{W_{net}}{E_{in}} = \frac{W_t - W_p}{E_{in}} = \frac{T_H - T_L}{Q_e} \quad (13)$$

$$E_{in} = Q_e \frac{T_H - T_L}{T_H} \quad (14)$$

Where $E_{in}$ represents the exergy of the system under ideal condition.

For a stable system, the irreversible loss of the system $I$ is calculated as the following formula:

$$I = T_0 \left( \sum_{out} s - \sum_{in} s - \sum \frac{q_i}{T_i} \right) \quad (15)$$

Here, in the formula (15), $q_i$ has a positive and negative sign. When the working fluid absorbs heat, $q_i > 0$; while when the working fluid releases heat, $q_i < 0$.

Where $m_f$ is the mass flow rate of working fluid, kg/s; $h$ is the specific enthalpy of working fluid, kJ/kg; $T_0$ is the ambient temperature, K; $s$ is the specific entropy of working fluid, kJ/(kg·K). The footnotes 1-4 represent each state point.
2.2. Working principle and mathematical model of S-ORC

The combined cycle S-ORC is made up of dual cycles with binary working fluids, including the first stage SRC and the second stage ORC. The basic generating equipment involves a turbine, a generator, a condenser of the first stage (namely the second stage evaporator) and a turbine, a generator, a condenser of the second stage, a working fluid pump etc. as sketched in Fig. 5, while the corresponding Temperature-Entropy diagrams are illustrated in Fig. 6 and Fig. 7.

State a to b represents an adiabatic expansion process of steam in the turbine. And b-c represents an isobaric exothermic process of steam in the first stage condenser namely the second stage evaporator.

State 1 to 2 represents an adiabatic expansion process of the organic working fluid in the second stage turbine. State 2 to 3 represents an isobaric exothermic process in the second stage condenser. State 3 to 4 represents an adiabatic compression process in the working fluid pump. State 4 to 1 represents an isobaric endothermic process in the second stage evaporator.

Fig. 5. Schematic diagram of S-ORC cycle.
For the first stage cycle, State a to b represents an adiabatic expansion process of steam in the turbine. Considering the internal efficiency of the turbine $\eta_t$ in actual calculation, the output power of steam in the first stage turbine is $W_{t1}$:

$$W_{t1} = m_s (h_a - h_b) \eta_t$$  \hspace{1cm} (16)

The irreversible loss of steam in the first stage turbine is $I_{t1}$:

$$I_{t1} = m_s T_0 (s_b - s_a)$$  \hspace{1cm} (17)

State b to c represents an isobaric exothermic process of steam in the first stage condenser (the second stage evaporator), the released heat is $Q_{c1}$,

$$Q_{c1} = m_s (h_b - h_c)$$  \hspace{1cm} (18)

The irreversible loss of the first stage condenser is $I_{c1}$:

$$I_{c1} = m_s T_0 (s_c - s_b - \frac{h_c - h_b}{T_{L1}})$$  \hspace{1cm} (19)

Here, for the steam in the first stage condenser, the low temperature organic working fluid is a cold source. The temperature of the cold source is changeable, and its temperature $T_{l1}$ is:

$$T_{l1} = \frac{T_4 - T_1}{\ln(\frac{T_4}{T_1})}$$  \hspace{1cm} (20)

Where $T_4$ and $T_1$ are the temperature of the cold source at the first stage condenser inlet and outlet, K.

For the second stage cycle, the state 4 to 1 process represents organic
working fluid in the second stage evaporator (the first stage condenser) absorbing primary circulation turbine outlet exhaust steam heat. The heat absorbed by the organic working fluid is $Q_{e2}$,

$$Q_{e2} = Q_{e1} = m_f (h_1 - h_4)$$  \hspace{1cm} (21)

The irreversible loss of the second stage evaporator is $I_e$,

$$I_e = m_f T_0 (s_1 - s_4 - \frac{h_1 - h_4}{T_H})$$  \hspace{1cm} (22)

Here, for the low temperature organic working fluid in the second stage evaporator, the high temperature steam is a heat source. The temperature of the heat source is changeable, and its temperature $T_H$ is:

$$T_H = \frac{T_b - T_c}{\ln\left(\frac{T_c}{T_b}\right)}$$  \hspace{1cm} (23)

Where $T_b$ and $T_c$ are the temperature of the heat source at the second stage evaporator inlet and outlet, K.

State 1 to 2 represents an adiabatic expansion process in the second stage turbine of the organic working fluid. Considering the internal efficiency of the turbine $\eta_t$ in actual calculation, the output power of second stage turbine is $W_{t2}$:

$$W_{t2} = m_f (h_1 - h_2)\eta_t$$  \hspace{1cm} (24)

The irreversible loss of the second stage turbine is $I_{t2}$:

$$I_{t2} = m_f T_0 (s_2 - s_1)$$  \hspace{1cm} (25)

State 2 to 3 represents an isobaric exothermic process of organic working fluid in the second stage condenser, the heat released is $Q_{c2}$:

$$Q_{c2} = m_f (h_2 - h_3)$$  \hspace{1cm} (26)

The irreversible loss of the second stage condenser is $I_{c2}$:

$$I_{c2} = m_f T_0 (s_3 - s_2 - \frac{h_3 - h_2}{T_{l2}})$$  \hspace{1cm} (27)

Here, the temperature of the cold source is changeable, and its temperature can be calculated as:

$$T_{l2} = \frac{T_{l2, out} - T_{l2, in}}{\ln\left(\frac{T_{l2, out}}{T_{l2, in}}\right)}$$  \hspace{1cm} (28)
Where $T_{L2,\text{in}}$ and $T_{L2,\text{out}}$ are the temperature of cooling water at the second stage condenser inlet and outlet, K.

State 3 to 4 represents an adiabatic compression process of the working fluid in the working fluid pump, considering the inner efficiency of the working fluid pump $\eta_p$ in actual calculation, the power consumption of the working fluid pump is $W_p$:

$$W_p = \frac{m_f(h_4 - h_3)}{\eta_p} \quad (29)$$

The irreversible loss of the working fluid pump is $I_p$:

$$I_p = m_f T_0 (s_4 - s_3) \quad (30)$$

The power generation capacity of the unit working fluid is $P$:

$$P = \frac{E}{m_f} = w_i \eta_s \quad (31)$$

The thermal efficiency of the system is $\eta_e$:

$$\eta_e = \frac{W_{\text{net}}}{Q_{el}} = \frac{Q_{el1} - Q_{el2}}{Q_{el1}} \quad (32)$$

Take the whole device as a thermodynamic system. From the first law of thermodynamics we know that, $Q = W + \Delta U$. For a closed system, the variation of thermodynamic energy is zero, in other words, $\Delta U = 0$, so $Q = W$.

$$\eta_e = \frac{m_s(h_y - h_t) - m_f(h_2 - h_3)}{m_s(h_y - h_t)} \quad (33)$$

The exergy efficiency of the system is $\eta_{ex}$:

$$\eta_{ex} = \frac{W_{\text{net}}}{E_{in}} \quad (34)$$

$$E_{in} = W_{\text{net}} + \sum I \quad (35)$$

For a stable system, the irreversible loss of the system is $I$, which can be calculated according to the following formula:

$$I = T_0 (\sum s - \sum s - \sum \frac{q_i}{T_i}) \quad (36)$$

Here, in the formula (36), $q_i$ has a positive and negative sign. When the working fluid absorb heat, $q_i > 0$, while when the working fluid release heat, $q_i < 0$. 


Where, \( m_s \) is the mass flow rate of steam in the first stage cycle, kg/s; \( m_f \) is the mass flow rate of organic working fluid in the second stage cycle kg/s; \( h_a, h_b, h_c \) are the specific enthalpy of steam at each state point in the first stage cycle, kJ/kg; \( s_a, s_b, s_c \) are the specific entropy of steam at each state point in the first stage cycle, kJ/(kg K); \( W_{net} \) is cyclic network, kW; \( h_{a-4} \) is the specific enthalpy of organic working fluid at each state point in the second stage cycle, kJ/kg; \( s_{a-4} \) is the specific entropy of organic working fluid at each state point in the second stage cycle, kJ/(kg K).

3. The set of calculation conditions and selection of organic working fluid

3.1. The set of calculation conditions

In this paper, the 150-350 °C waste heat steam is used as a heat source, setting waste heat steam flow rate is 2.5 kg/s. For the S-ORC cycle, a tube fin heat exchanger is used as the evaporator of the two stage cycles, the pinch point of evaporator is about 15°C. Set the inlet temperature of working fluid in the second stage cycle is 85-175°C, then the outlet temperature of the exhaust steam in the first stage cycle is 105-190°C. A plate heat exchanger is adopted as the condenser. According to the climate conditions in Shanghai, condensation temperature is 37 °C. Taking into account the compression capacity of the equipment as well as the system cost, we consider the maximum pressure of the system is the evaporation pressure of the working fluid when the evaporation pressure of organic working fluid is less than 3MPa, otherwise the maximum pressure of the system is 3MPa. To avoid the condensation pressure getting too low and leading to air infiltration, the lowest condensation pressure of the SRC system is set to 9kPa. Compared with the SRC power generation system, the maximum pressures of the ORC and S-ORC systems are larger, the largest is 2.711 MPa and this pressure fulfills the maximum pressure of 3MPa under design conditions. The maximum pressure of the three kinds of cyclic power generation systems increases as the heat source temperature rise. The condensing pressure of SRC and S-ORC power systems in the first stage cycle is a negative pressure, but it is larger than the minimum system pressure of 9kPa in the design working conditions; the condensing pressures of ORC and S-ORC secondary circulation are all positive pressures. The design parameters of the three power generation systems are shown in Table 1.

It's worth noting that, we choose a hydraulic diaphragm metering pump as
the working fluid pump and the turbine isentropic efficiency $\eta_t$ is 0.7 is given by the product supplier, Shen Bei pumps manufacturing Co, Ltd. In addition, after resorting to the statements of J. Bao et al. in Ref. [17] on various types of ORC system expansion characteristics and the selection, the centripetal turbine expander was chosen in this paper. It has compact structure, easy manufacturing, and the characteristics of large enthalpy drop in a single stage. Because of its expansion efficiency is relatively high, the centripetal turbine expander applies to the occasion of output power ranges from 50 kW to 5 MW, and the equal entropy efficiency is between 0.78-0.85 [18]. According to the expected power generation and the selected seven kinds of organic working fluids, the approximate pump isentropic efficiency $\eta_p$ is 0.8.

### Table 1
Calculation condition table.

| Circular name | Turbine inlet Temperature (°C) | Turbine inlet Pressure (MPa) | Condensation Temperature (°C) | $\eta_p$ | $\eta_t$ |
|---------------|-------------------------------|------------------------------|-----------------|--------|--------|
| SRC           | 150-350                       | 0.2-1.4                      | 45               | 0.7    | 0.8    |
| ORC           | 100-180                       | 1.5-3                        | 37               |        |        |
| S-ORC I cycle | 150-350                       | 0.2-1.4                      | 45               |        |        |
| S-ORC II cycle| 90-180                        | 1.5-3                        | 37               |        |        |

### 3.2. Introduction and selection principles of organic material

One main concern in ORC and S-ORC technology is the proper selection of the working fluid. The thermodynamic properties of working fluid not only have important effects on the efficiency of ORC and S-ORC, but also the size of the system components. Therefore, the selection of working fluid plays a key role in the performance and economic efficiency of ORC system. Many aspects need to be taken into account, namely, thermodynamic properties, global warming potential (GWP), thermal stability [19], safety and environmental aspects [20], toxicity, flammability, auto-ignition temperature, costs [21], and availability.

At present, the commonly used organic working fluids in the ORC, S-ORC power generation system, can be divided into alkanes, silicones and working fluids, etc. In accordance with the shape of the saturated steam line characteristic curve in the Temperature-Entropy diagram ($T$-$S$ diagram), working fluids can be divided into dry working fluid ($dS/dT > 0$), adiabatic working fluid ($dS/dT = 0$) and wet working fluid ($dS/dT < 0$) [22]. Wherein, the dry working fluid and adiabatic working fluid can flow out in the state of superheated steam after
working in the turbine, so they can flow into the turbine in the saturated steam state without overheating, thus the heat exchanger area can be reduced and also can avoid erosion of the turbine blade by droplets. Therefore, dry working fluids and adiabatic working fluid are often used as working fluids of organic Rankine cycles [23].

The properties of the organic working fluid have a significant effect on the system performance. Domestic and foreign scholars have carried out extensive researches [24-27] on the selection of ORC working fluid and summarized the principles of ORC working fluids selection [28-32].

- The critical temperature should be slightly higher than the maximum temperature in the cycle to avoid many potential problems caused by a trans-critical cycle.
- The condensation pressure should not be too low and it is better to keep a positive pressure to prevent the outside air from penetrating and affecting the performance of the cycle.
- The best selection is dry working fluid, in the T-S diagram of saturated vapor line, \( \frac{dS}{dT} > 0 \).
- A small latent heat of vaporization, low viscosity, and high heat transfer coefficient and good thermal stability.

In addition, the working fluids not only have to meet the above-mentioned thermodynamic properties, but also have to meet the requirements of environmental protection and safety. Combustibility and toxicity are two important indexes of working fluids security. According to the China National Working fluid safety classification GB/T7778-2008, working fluid safety is divided into six types: A1, A2, A3, B1, B2, and B3, as shown in Table 2. Ozone depleting potential (ODP) and global warming potential (GWP) of working fluids are two important indicators of environmental protection. With the approach of the HCFC and CFC working fluid phase out deadline regulated in the Montreal protocol, national industry associations and the government actively promulgate or amend the relevant regulations to promote the smooth transition of domestic working fluid use. To this end, good thermodynamic performance, safety, low ODP and GWP values, low price and easy access are several of the main principles in the selection of working fluids.
Table 2
Safety classification of working fluids.

| Combustibility                      | Low toxicity | High toxicity |
|-------------------------------------|--------------|--------------|
| Nonflammable, nonflame spread       | A1           | B1           |
| Flammability                        | A2           | B2           |
| Explosibility                       | A3           | B3           |

3.3. The screening of working fluids in calculating conditions

150-350 °C waste heat steam can be divided into 150, 200, 250, 300 and 350 °C five stages. According to the temperature of each kind of waste heat steam we divide the first grade circulating outlet temperature and take the heat exchanger pinch point temperature as 15°C. Considering the critical temperature, safety and environmental protection in accordance with [33, 34], we select R141b, R123, R245ca, R245fa, R236ea, R600a and R134a these seven kinds of commonly used organic compounds as working fluids of S-ORC and ORC in this paper. Properties of each organic working fluid are shown in Table 3.

Table 3
Physical parameters of the working fluid

| Working fluid | Critical temperature (°C) | Critical pressure (MPa) | Evaporation temperature (°C) | Security stage | ODP  | GWP | Wet and dry |
|---------------|---------------------------|-------------------------|-----------------------------|----------------|------|-----|-------------|
| R141b         | 204.35                    | 4.212                   | 32.05                       | A2             | 0.11 | 630 | Dry         |
| R123          | 183.68                    | 3.661                   | 27.82                       | B1             | 0.02 | 93  | Dry         |
| R245ca        | 174.42                    | 3.925                   | 25.13                       | B2             | 0    | 560 | Dry         |
| R245fa        | 154.01                    | 3.650                   | 15.14                       | B3             | 0.1  | 0.1 | Dry         |
| R236ea        | 139.29                    | 3.502                   | 5.0                         | B1             | 0.1  | 0.1 | Dry         |
| R600a         | 134.66                    | 3.629                   | -11.8                       | B2             | 0    | 770 | Dry         |
| R134a         | 101.96                    | 4.0593                  | -26.07                      | A1             | 0    | 0.1 | Dry         |

4. Comparison and analysis of SRC, ORC and S-ORC power generation systems

4.1. Simulation results of the SRC power generation system

The working fluid of SRC is water. To ensure that the SRC and S-ORC system have the same heat source, the outlet temperature of the turbine is set as 45 °C, outlet pressure is the corresponding saturation pressure, working fluid condensing temperature is 37°C and the turbine isentropic efficiency is 0.8, the
pump isentropic efficiency is 0.7.

Table 4 shows the simulation results of the SRC power generation system, it can be seen that the thermal efficiency of the SRC power generation system increases with the increase in heat source temperature, and the maximum thermal efficiency reaches 13.19%; when the heat source temperature is 150℃, the thermal efficiency of the system is only 4.97%. The exergy efficiency of the system increases with the increase of heat source temperature, the maximum exergy efficiency is 39.76% while the minimum is only 28.15%.

| Working fluid | $Q_{in}$ (kW) | $W_t$ (kW) | $W_p$ (kW) | $E$ (kW) | $E_{in}$ (kW) | $\eta_{t}$ (%) | $\eta_{ex}$ (%) |
|---------------|---------------|------------|------------|---------|--------------|----------------|----------------|
| water         | 7487.74       | 992.23     | 4.38       | 987.84  | 2484.53      | 13.19          | 39.76          |
| water         | 7234.98       | 815.29     | 3.44       | 811.85  | 2172.15      | 11.22          | 37.38          |
| water         | 6988.60       | 642.83     | 2.50       | 640.33  | 1848.73      | 9.16           | 34.64          |
| water         | 6752.10       | 477.28     | 1.55       | 475.72  | 1510.85      | 7.05           | 31.49          |
| water         | 6535.26       | 325.49     | 0.61       | 324.88  | 1153.97      | 4.97           | 28.15          |

### 4.2. Simulation results of the ORC power generation system

Table 5 is the simulation result for the ORC power generation system. In the ORC system, the working fluid suitable for 150 ℃ heat source is R245fa, and R141b for 200-350℃ heat source. The inlet and outlet temperature of the heat source is the same as that of the S-ORC power generation system. At the same time, the condensing temperature of working fluid is 37 ℃ and the turbine isentropic efficiency is 0.8, the pump isentropic efficiency is 0.7.

| Working fluid | $Q_{in}$ (kW) | $m_f$ (kg/s) | $W_t$ (kW) | $W_p$ (kW) | $E$ (kW) | $E_{in}$ (kW) | $\eta_{t}$ (%) | $\eta_{ex}$ (%) |
|---------------|---------------|--------------|------------|------------|---------|--------------|----------------|----------------|
| R141b         | 7404.14       | 24.24        | 1235       | 65         | 1170.71 | 2457.24      | 15.81          | 47.64          |
| R141b         | 7151.37       | 23.41        | 1193       | 63         | 1130.75 | 2147.50      | 15.81          | 52.65          |
| R141b         | 6905.00       | 22.61        | 1152       | 60         | 1091.79 | 1827.07      | 15.81          | 59.76          |
| R141b         | 6668.50       | 21.83        | 1113       | 58         | 1054.40 | 1492.62      | 15.81          | 70.64          |
| R245fa        | 6451.65       | 26.36        | 782        | 48         | 734.44  | 1139.69      | 11.38          | 64.44          |

After the heat source temperature reaches 200℃, due to the employment of
R141b and the evaporation temperature of the working fluid is 175°C, so the thermal efficiency no longer changes, reaching a maximum of 15.81%. The maximum exergy efficiency of the whole cycle reaches 70.64% when the heat source temperature is 200°C, the working fluid is R141b while the evaporation temperature is 175°C. After 200 °C, with the increase in the heat source temperature, exergy efficiency began to decline. The reason is that, for subcritical cycles, the highest critical temperature limits the maximum evaporation temperature of working fluid at 175°C. The increase of heat source temperatures leads to bigger heat transfer temperature differences in the evaporator, the irreversible loss increases, and the exergy efficiency declines.

4.3. Comparison of simulation results of SRC, ORC and S-ORC power generation systems in fluid-low temperature

4.3.1 Comparison of thermal efficiency

Fig. 8 is thermal efficiency with the heat source variation for three kinds of cycle power generation systems. At 150 °C to 210 °C or so of low-temperature, the thermal efficiency of the ORC cycle is higher than that of the SRC and S-ORC cycles. The advantages of S-ORC begin to appear after the heat source temperature reaches 210°C. Its thermal efficiency is higher than that of the SRC and ORC cycles; in addition, the thermal efficiency of S-ORC is always higher than that of SRC.

Fig.8. The thermal efficiency of SRC, ORC and S-ORC power generation systems.

4.3.2 Comparison of exergy efficiency

Fig. 9 shows the exergy efficiency of the three kinds of power generation systems. At 150 °C to 210 °C or so of low-temperature, the exergy efficiency of
the ORC cycle is higher than that of the SRC and ORC cycles. Above about 210°C, the exergy efficiency of the ORC cycle was lower than that of the S-ORC cycle. The exergy efficiency of the SRC cycle at 150-350°C is the lowest stage for two reasons. On one hand, water is a kind of wet working fluid with high boiling point (100 °C) under normal pressure; to ensure the dryness of working fluid at the turbine outlet, the working fluid is in the negative pressure section in the turbine outputs. Considering the safety of the equipment (greater than 9kPa), the outlet temperature reaches a minimum of 45 °C and is higher than that of the ORC cycle of 37 °C. On the other hand, the thermal efficiency of SRC at low temperature is lower than that of ORC.

The exergy efficiency of SRC, ORC and S-ORC power generation systems.

4.3.3 The comparison of power generation

The system power output of SRC, ORC and S-ORC is shown in Fig.10. When the heat source temperature is around 150 °C to 210 °C, the ORC system generating capacity is greater than that of the S-ORC and SRC systems; after the heat source temperature exceeds 210°C, the S-ORC system has more power than ORC and SRC. The power generation of the SRC system is always less than that of S-ORC. After the heat source temperature reaches 200°C, the generating capacity rising trend of the ORC system is clearly less than SRC, when the heat source temperature reaches 350°C, the power generation of ORC and SRC systems is very close.

In addition, in the picture it can be found that by regulating the export temperature of the first stage cycle of S-ORC system, the slope of the power generation line of the S-ORC system is closer to that of the ORC system in the
low temperature section, while it’s relatively close to the SRC system in the middle and high temperature section. It indicates that S-ORC system has a good matching with the fluid, high and low temperature heat source by adjusting primary circulation outlet temperature of S-ORC system.

Fig.10. The power generation of SRC, ORC and S-ORC systems.

5. Simulation results and analysis of S-ORC power generation system

We calculated steam and organic working fluid combined cycle (S-ORC) under the condition of 150-350 °C steam heat source. According to the output temperature of the first cycle, this text selected a variety of organic substances for each temperature of the heat source, combined with the characteristics of the organic substance and the selection principle of organic working fluid. At the same time, we used the VBA module in Excel to program, calling libraries REFPROP 8.0 and analyzed the simulation results of thermal power generation model for each of the thermal sources. We compared the optimal primary circulation outlet temperature of 150-350 °C S-ORC simulation results and obtained the variation of thermal efficiency and exergy efficiency with heat source temperature of S-ORC, which is within the temperature range.

5.1. Thermal efficiency of S-ORC

The first law of thermodynamics illuminates the basic principle of the energy conservation and conversion. Energy cannot be generated or consumed by itself. Thermal efficiency is an evaluating index based on the first law of
thermodynamics, which reflects the ratio of the evaporator absorbed heat from the low temperature heat source to the output power. The thermal efficiency of the S-ORC with first cycle turbine outlet temperature changes is shown in Fig.11, thermal efficiency with the heat source temperature changes is shown in Fig.12 and the optimal thermal efficiency diagram of S-ORC is shown in Fig.10.

**Fig.11.** S-ORC thermal efficiency varies with the first stage turbine outlet temperature chart.

Fig. 11 shows that the thermal efficiency increases with the increase of the outlet temperature of the first stage under the condition of different temperature heat source and the thermal efficiency of the high-temperature heat source was obviously higher than that of the low-temperature heat source. When the organic working fluid and primary circulation outlet temperature were maintained, the thermal efficiency of the system increased gradually with the increase of heat source temperature.

**Fig.12.** S-ORC thermal efficiency with heat source temperature chart.
Fig. 12 shows that the thermal efficiency of the system increases with the source temperature under the condition that the working fluid and primary circulation outlet temperature are constant.

![Chart](image)

**Fig.13.** 150-350 °C heat S-ORC maximum thermal efficiency chart.

Fig. 13 shows that, as the heat source temperature increases, maximum thermal efficiency increases gradually, but the rising trend slows down. The reason is that, as the heat source temperature rises, due to the limitation of highest critical temperature of organic matter in the second stage ORC, the first stage SRC gradually occupies the dominant position of the whole system and the thermal efficiency of the system was mainly influenced by the first-order SRC cycle. When the heat source temperature is higher than 250°C, R141b, whose critical temperature is the highest, was used as the working fluid in the second stage cycle where the thermal efficiency was a maximum.

### 5.2. Exergy efficiency of S-ORC

The second law of thermodynamics is described as the quality of energy, thermodynamically, which refers to the change in the quality of energy during the phase change in the processes. According to the second law of thermodynamics, exergy is the measurement of the maximum useful work that can be obtained from the system. Moreover, the exergy can be called as irreversibility in thermodynamic point of view. Exergy efficiency is based on the exergy analysis method, which is more comprehensive than the energy analysis and it can specify the nature of the energy loss more deeply. The lower the irreversible loss of the
system, the greater the exergy efficiency. S-ORC exergy efficiency with the primary cycle turbine outlet temperature changes is shown in Fig.14.

![Fig.14. S-ORC exergy efficiency varies with the first stage turbine outlet temperature chart.](image)

Fig. 14 shows that the exergy efficiency increases with turbine outlet temperature of the primary cycle under different heat source temperatures and they all reached the maximum value at the highest point of the primary cycle turbine outlet temperature.

![Fig.15. S-ORC exergy efficiency with the heat source temperature chart.](image)

Fig. 15 shows that the S-ORC exergy efficiency decreases at first and then increases. This is because the exergy efficiency of the ORC is very high at low temperature while SRC’s exergy efficiency is low. With the increase of
temperature, the advantages of SRC in the high temperature section can be reflected.

![Fig.16. 150-350 °C heat S-ORC maximum exergy efficiency chart.](image)

Fig. 16 shows that the S-ORC maximum exergy efficiency increases at first and then decreases with the increase of the heat source temperature. At 250 °C thermal energy obtained by the second stage ORC cycle accounts for the largest proportion of the whole power system and the exergy efficiency of this point is a maximum of 66.46%.

5.3. S-ORC generating capacity

Through the simulation and calculation of power generation, we know that the power generation of the system increases with the increase in heat source temperature. With the increase in heat source temperature, the heat energy obtained by the system is increasing and the system generating capacity gets larger.

Fig. 17 is the S-ORC system maximum power generation graph under the condition of 150-350 °C heat source temperature. It can be seen that the maximum system generating capacity of each temperature increases with the increase of heat source temperature. In the picture, there is a significant change in the rising trend of the system. In the picture, the rising trend of system capacity decreased significantly, the reason is that the highest critical temperature (204.35℃) of organic working fluid limits the first stage turbine outlet temperature up to 190 °C. After the heat source temperature reaches 250°C, the heat energy obtained by the
first SRC cycle and generating capacity increased gradually while the thermal efficiency of the SRC cycle in the 250-350 °C heat source section is lower than that of the ORC cycle in the 120-250°C heat source section, leading to the phenomenon of upward trend of system power generation gradually becoming small above 250°C.

Fig.17. 150-350 °C heat S-ORC system maximum power output figure.

5.4. Experimental study of S-ORC power generation system

5.4.1 Experimental background and content

At present, there are many thermodynamic models and theoretical studies on S-ORC waste heat power generation system, but the experimental research and practical application are still few. In view of the first stage of S-ORC cycle is more mature, and the second stage is still in the theoretical stage with a small amount of experimental research, therefore, we will study the second stage ORC to verify the feasibility of S-ORC in this portion. The experimental address is located in Tianjin City, a certain workshop building of Jiang Tianyi heavy factory district, meanwhile, the factory is located in the north. On the one hand, the local waste heat steam is used for heating in the life, on the other hand, for the heat of the process. After the transition season and summer part of the steam will be remained. Due to there exist a certain range of temperature fluctuation in waste heat steam, therefore, in order to effectively use this part of steam power generation, the hot water storage tank installed in the factory. Water is heated to 120 °C under the action of waste heat steam and then it was send in a generating
unit for generating electricity, the power is directly supplied to the plant. The design power is 280kW capacity, we expect to verify the feasibility of S-ORC for waste heat power generation through experiments, improve the effectiveness of waste heat utilization technology in practical applications, reduce the power consumption of the plant from the grid and reduce the thermal pollution of the environment.

5.4.2 Introduction of experimental system

Schematic diagram of S-ORC power generation system second stage cycle is shown in Fig.18. Hot water at 120°C (corresponding to the S-ORC primary circulation outlet) enters the evaporator from the hot water storage tank. Before organic working fluid enters the turbine to work, it will absorb heat at a constant pressure in the evaporator, then enter into the condenser. Working fluid condensates to the saturated liquid state will enter into the evaporator to absorb heat again by the working fluid pump. At this time, the whole cycle has been completed. The actual devices of the test system are shown in Fig. 19. The whole power system is divided into three parts: water circulation, ORC circulation and cold water circulation. With regard to the heat source of 120°C, we chose R245fa as the working fluid. The working fluid using in the ORC cycle is R245fa which has good thermodynamic properties such as low specific heat and viscosity, low toxicity, low ozone depletion potential, low flammability. Most of the ORC systems are using R245fa as the working fluid have moderate global warming potential of 950, power density, while its lower critical pressure at higher temperature allows for reasonable system thermal efficiency [35]. Owing to the above mentioned properties and the favorable economic conditions, R245fa is a convenient option as working fluid. The properties of the R245fa are obtained from Ref. [36]
Fig. 18. Schematic diagram of S-ORC power generation system second stage cycle

Fig. 19. Actual devices of S-ORC power generation system second stage cycle

5.4.3 Comparison of test results and simulation results

After the system runs stably, the state parameters of the 19 groups are listed in Table 6. However, in actual operation, the output power can only reach
250-260kW, and the thermal efficiency is 9.44%-9.60%. Compared with the thermal efficiency 11.12%, which is calculated in accordance with the mathematical model mentioned above, the thermal efficiency of the actual operation is lower than the design, less about 2%.

It is worth mentioning that, in this paper, we have not employed a single parameter criterion as the objective function as already mentioned, but here we only use the thermal efficiency to reflect the feasibility of the S-ORC system. The main reason is that if we want to conduct more comprehensive and detailed verification, more experimental instruments and equipment are needed to support it. Subjecting to the economic constraints, so it cannot be completed at present. Furthermore, we observed that in all simulated cases, the variation trend of the thermal efficiency, exergy efficiency and generating capacity are the same. So, the thermal efficiency can be used as an indicator of whether the second stage power generation system is feasible.

Table 6
The thermal efficiency table of actual operation

| Time  | The pressure of working fluid in evaporator (MPa) | Working fluid temperature at the outlet of evaporator (K) | Working fluid temperature at the condenser outlet (K) | Working fluid temperature at the inlet of the condenser (K) | The pressure of working fluid in condenser (MPa) | Generated power (kW) | Thermal efficiency (%) |
|-------|-----------------------------------------------|----------------------------------------------------------|-----------------------------------------------------|----------------------------------------------------------|-----------------------------------------------|-----------------------|------------------------|
| 8:00  | 0.922                                         | 91.05                                                   | 25.95                                               | 45.95                                                    | 0.184                                         | 255                   | 9.60%                  |
| 9:00  | 0.932                                         | 91.35                                                   | 25.95                                               | 46.05                                                    | 0.184                                         | 258                   | 9.60%                  |
| 10:00 | 0.951                                         | 91.95                                                   | 26.15                                               | 46.75                                                    | 0.185                                         | 260                   | 9.44%                  |
| 11:00 | 0.951                                         | 91.95                                                   | 26.15                                               | 46.75                                                    | 0.185                                         | 260                   | 9.44%                  |
| 12:00 | 0.951                                         | 91.95                                                   | 26.15                                               | 46.75                                                    | 0.185                                         | 260                   | 9.44%                  |
| 13:00 | 0.932                                         | 91.35                                                   | 25.95                                               | 46.05                                                    | 0.184                                         | 258                   | 9.60%                  |
| 14:00 | 0.932                                         | 91.35                                                   | 25.95                                               | 46.05                                                    | 0.184                                         | 258                   | 9.60%                  |
| 15:00 | 0.932                                         | 91.35                                                   | 25.95                                               | 46.05                                                    | 0.184                                         | 258                   | 9.60%                  |
| 16:00 | 0.922                                         | 91.05                                                   | 25.95                                               | 45.95                                                    | 0.184                                         | 255                   | 9.60%                  |
| 17:00 | 0.922                                         | 91.05                                                   | 25.95                                               | 45.95                                                    | 0.184                                         | 255                   | 9.60%                  |
| 18:00 | 0.932                                         | 91.35                                                   | 25.95                                               | 45.95                                                    | 0.184                                         | 258                   | 9.60%                  |
| 19:00 | 0.951                                         | 91.95                                                   | 26.15                                               | 46.05                                                    | 0.185                                         | 260                   | 9.44%                  |
| 20:00 | 0.951                                         | 91.95                                                   | 26.15                                               | 46.05                                                    | 0.185                                         | 260                   | 9.44%                  |
| 21:00 | 0.922                                         | 91.05                                                   | 25.95                                               | 45.95                                                    | 0.184                                         | 255                   | 9.60%                  |
| 22:00 | 0.932                                         | 91.35                                                   | 25.95                                               | 46.05                                                    | 0.184                                         | 258                   | 9.60%                  |
It’s not difficult to find that the experimental data is not consistent with the theoretical value. Since presently little or no published experimental data are available on such combined power cycle, so the calculation results in this paper cannot be compared. As far as the comparison is concerned, influential factors are the following: (1) The system heat dissipation on the environment is not considered in the design; (2) the flow resistance of the working fluid in the pipeline is not considered; (3) The theoretical pump power and actual pump power can’t match completely. Although the test results are not ideal, but the results show the feasibility of the S-ORC power system and provide a reference for the further study.

6. Conclusions

The main conclusions of this paper are as follows:

- At 150-210 °C heat conditions, in the three cycles, ORC has the highest thermal efficiency and exergy efficiency and power generation; at 210-350 °C heat conditions, the performance of S-ORC has a clear advantage where its thermal efficiency and exergy efficiency are higher than the SRC and ORC power systems. This means that the combination of low-boiling point fluid and turbine can make full use of the low-grade waste heat energy below 200 °C. For low-grade enthalpy heat, the efficiency of the low-boiling point technology of the Rankine cycle is higher than in the conventional steam Rankine cycle in terms of recovering the latent heat of steam when it condenses.

- The S-ORC power generation system has better heat source temperature matching. By regulating the output temperature of the first stage in the S-ORC power generation system, the thermal efficiency and exergy efficiency of S-ORC is higher than that of SRC in the low-temperature heat source section by using the second stage ORC cycle. The thermal efficiency, exergy efficiency and power generation of S-ORC is higher than that of ORC and SRC in the high-temperature heat source section. Therefore, the S-ORC power generation system can be flexibly adapted to the change of heat source temperature.
• At 150-350 °C, for each heat source temperature, the maximum thermal efficiency and maximum power generation capacity of the S-ORC power generation system increases with source temperature while the maximum exergy efficiency first increased and then decreased. This indicates that as the temperature of the heat source increases, the whole S-ORC power generation system is inclined to the first SRC cycle, and the influence of the SRC cycle becomes more and more.

The thermal efficiency, exergy efficiency and power generation of the SRC power generation system are all lower than those of ORC and S-ORC power generation systems for 150-350°C waste heat steam. However, above 200 °C, the upward trend of thermal efficiency and power generation of the SRC system is significantly greater than that of the ORC power system; when the heat source temperature reaches 350 °C, thermal efficiency and power generation of two cycles becomes very close. This indicates that as the heat source temperature increases, the thermal efficiency and power generation of SRC will exceed that of the ORC.

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HIGHLIGHTS

- SRC, ORC and S-ORC three kinds of power systems are proposed.
- The advantages and disadvantages of three models under different conditions are compared.
- ORC is more efficient than SRC in recovering the latent heat of condensation of water vapor.
- S-ORC power generation system has a better matching of heat source temperature.
- The characteristics of the organic working fluids have been considered.