INTRODUCTION

The excessive use of energy will lead to a shortage of nonrenewable energy worldwide; in addition, the efficiency of energy utilization is very low during production. The organic Rankine cycle (ORC) can recycle low-temperature waste heat from a production process and use it to generate electricity, and this approach is a very effective technical method for energy conservation. The ORC is widely used when the heat source is low-temperature waste heat that is difficult to recycle. Nguyen et al.\(^1\) studied six kinds of working fluids, such as water, ammonia, and propane, to analyze the performance of the ORC for a recycling industrial waste heat system with temperatures of 100-250°C and showed that the ORC was more efficient than the steam Rankine cycle under low-temperature heat source conditions. To examine the performance of the ORC system, many scholars have set up experimental devices to simulate the power generation performance of an ORC system driven by different heat sources. Based on experimental data obtained in Lesotho, Sylvain et al.\(^2\) optimized a low-temperature ORC system with...
The ORC combines the characteristics of the Rankine cycle with the evaporation at low temperatures of organic working fluids. The thermal properties of the working fluids are important factors affecting the efficiency of the ORC. For a low-temperature waste heat ORC power generation system, the screening of a working fluid needs to not only pay attention to the economics of the system but also, with the increase in environmental awareness, to consider the environmental protection, safety, and stability of the working fluid. These factors must be considered comprehensively to select a suitable working fluid for an ORC system. Many scholars have conducted in-depth research on optimal working fluid selection for ORC systems. Wang et al. compared the thermodynamic properties of R134a, dry working fluid was very suitable for the system. Lakew with R245fa as the working fluid and found that this pure working fluid could improve the performance of a solar ORC power generation system. Heberle et al. studied mixtures of R245fa, a mixture working fluid could improve the thermal efficiency of the ORC. The conclusion showed that the second-law efficiency of these zeotropic mixtures increased by 7.1% to 14.2% compared to that of pure working fluids. In 2018, Zhai et al. developed an active design method for zeotropic mixture working fluids. First, the method requires the composition of the mixture working fluid to meet the critical temperature within a suitable range; second, the temperature glide of the mixture working fluid can be matched with a circulating cold source; and finally, the mixed working medium is required to have certain environmental protection and safety. A method that satisfies the first requirement is called the boiling point difference method. The disadvantage of this method is that the calculation process is cumbersome and requires a certain amount of experience, resulting in a low accuracy. Compared with the exhaustive method commonly used for pure working fluids and the boiling point difference method used for binary zeotropic mixture optimization, this paper proposes a method to explain the relationship between the physical properties of the working fluid and the net power output. It does not require detailed calculations or experiments for every pure working fluid like the exhaustive method does and has no need to perform performance calculations for each alternative working fluid like the boiling point difference method requires.

The low-temperature heat source of the ORC power generation system studied in this paper comes from the waste heat between the stages of a multistage centrifugal compressor. To achieve the best work efficiency of the multistage centrifugal compressor, the gas at each stage outlet needs to be cooled to the same temperature. A four-stage air separation centrifugal compressor is used as an example; the compressor outlet temperature of each stage is approximately 100°C and needs to be condensed to 45°C. The ORC system is used to recover part of this heat and use it to generate electricity. The main work of this paper is as follows. (a) A new standard for finding the best pure working fluid and binary zeotropic mixture is presented. By using this method, the optimal pure working fluid and binary zeotropic mixture and mixing ratio can easily be found when the net power output is selected as the objective function. (b) A model is established for the ORC to recover the cooling heat between compressor stages. Thirteen different pure working fluids and binary zeotropic mixtures of two of those fluids are studied. The names and critical temperatures of the 13 pure...
working fluids are given in Table 1. The optimal evaporation temperature of the ORC, the pinch temperature difference, and the evaporation and condensation temperature glide of the mixture working fluid are found to obtain the maximum net power output of the ORC.

2 | MODELS AND CALCULATION ASSUMPTIONS

2.1 | System description

The principle of the ORC recycling compressor for waste heat is shown in Figure 1, and the gas parameters of the compressor inlet and outlet are shown in Table 2. As shown in Figure 1, the ORC is composed of a condenser, working fluid pump, evaporator, and expander. The temperature and pressure of the saturated liquid organic working fluid are raised through the pump, and this process corresponds to the 3-4 process shown in Figure 2. The following 4-5 and 5-1 processes correlate with the sensible heat rise and vaporization of the working fluid in the evaporator, respectively. Then, as shown by the 1-2 process in Figure 2, the high-pressure saturated gas releases energy in the expander and converts it to work, which drives the generator to generate electricity. The exhaust vapor out of the expander is first cooled to the saturated gas phase and then condensed into the saturated liquid phase in the condenser, as represented by the 2-2 and 2-3 processes, respectively.

2.2 | Mathematical model of the system

Before deriving the calculation model, the following assumptions need to be made:

1. The system works under steady-state conditions.
2. Pressure loss in pipes and equipment and heat loss from the system to the environment are not counted.

| TABLE 2  | State parameters of the compressor |
| Position                                           | Pressure/MPa | Temperature/°C |
| Cooling water                                     | 0.3          | ≤32            |
| 1st stage inlet of compressor                     | −0.003       | 32             |
| 1st stage outlet of compressor                     | 0.064        | 98             |
| 2nd stage inlet of compressor                     | 0.058        | 45             |
| 2nd stage outlet of compressor                     | 0.161        | 102            |
| 3rd stage inlet of compressor                     | 0.156        | 45             |
| 3rd stage outlet of compressor                     | 0.306        | 100            |
| 4th stage inlet of compressor                      | 0.302        | 45             |
| 4th stage outlet of compressor                     | 0.51         | 93             |

![FIGURE 1](Schematic diagram of the ORC.)

![FIGURE 2](T-s diagram of the ORC.)
3. The working fluid is saturated at the outlet of both the condenser and evaporator.
4. The pump and expander work at a constant efficiency.

2.2.1 | Energy balance for the ORC

There is a preheating process 4-5 and an evaporation phase change process 5-1 in the evaporator. The energy balance equation in the evaporator is as follows:

\[ Q_{\text{eva}} = m_w \cdot (h_6 - h_8) = m_{\text{wf}} \cdot (h_1 - h_4) \]  

(1)

where \( Q_{\text{eva}} \) is the heat transfer power of the evaporator, kW; \( m_w \) and \( m_{\text{wf}} \) mean the mass flow rates of the heat source and working fluid, respectively, kg s\(^{-1}\); \( h \) dictates the specific enthalpy, kJ kg\(^{-1}\); and a subscript number indicates the position of the working fluid on the T-S diagram.

The energy loss in the evaporator can be divided into two parts, one of which is the heat that cannot be absorbed by the evaporator as follows:

\[ I_{\text{eva1}} = m_w \cdot [ (h_8 - h_0) - T_0 \cdot (s_8 - s_0)] \]  

(2)

where \( I_{\text{eva1}} \) is the heat not absorbed by the evaporator, kW; \( T \) represents the temperature, K; \( s \) means the specific entropy, kJ kg\(^{-1}\)K\(^{-1}\); and subscript 0 indicates the reference state, which is the working fluid at an environment temperature of 20°C.

The other part is the irreversible loss in the evaporator, and this part can be described by:

\[ I_{\text{eva2}} = m_h \cdot [ (h_6 - h_8) - T_0 \cdot (s_6 - s_8)] - m_{\text{wf}} \cdot [ (h_1 - h_4) - T_0 \cdot (s_1 - s_4)] \]  

(3)

The sum of \( I_{\text{eva1}} \) and \( I_{\text{eva2}} \) is the energy loss in the evaporator.

The theoretical output power of the expander can be expressed as:

\[ W_{\text{th}} = m_w \cdot (h_1 - h_{2s}) \cdot \eta_{\text{th}} = m_{\text{wf}} \cdot (h_1 - h_2) \]  

(4)

where \( W_{\text{th}} \) represents the theoretical output power of the expander, kW; \( h_{2s} \) and \( h_2 \) are the specific enthalpies of the working fluid after the isentropic expansion process and actual expansion process, respectively; and \( \eta_{\text{th}} \) is the isentropic efficiency of the expander. The loss in the expander can be calculated as follows:

\[ I_{\text{th}} = m_{\text{wf}} \cdot T_0 \cdot (s_2 - s_1) \]  

(5)

In the condenser, there are two processes of precooling 2-2, and condensation phase change 2-3. The energy balance equation in the condenser can be described as follows:

\[ Q_{\text{con}} = m_w \cdot (h_{11} - h_9) = m_w \cdot (h_2 - h_3) \]  

(6)

where \( Q_{\text{con}} \) is the heat transfer power of the condenser, kW, and \( m_w \) is the mass flow rate of the cooling water, kg s\(^{-1}\). The energy loss in the condenser is as follows:

\[ I_{\text{con}} = m_{\text{wf}} \cdot ((h_2 - h_5) - T_0 \cdot (s_2 - s_3)) \]  

(7)

The power calculation equation of the working fluid pump can be calculated as follows:

\[ W_p = m_{\text{wf}} \cdot (h_{4s} - h_3) \cdot \eta_p = m_{\text{wf}} \cdot (h_4 - h_3) \]  

(8)

where \( W_p \) is the power of the working fluid pump, kW, and \( \eta_p \) is the isentropic efficiency of the working fluid pump. The energy loss in the working fluid pump can be calculated as follows:

\[ I_p = m_{\text{wf}} \cdot T_0 \cdot (s_4 - s_3) \]  

(9)

The mass flow rate of the working fluid can be calculated by the following equation:

\[ m_{\text{wf}} = \frac{m_h \cdot (h_6 - h_1)}{h_1 - h_5} \cdot \frac{Q_m}{T_3 \cdot \Delta s_5} \]  

(10)

where \( Q_m \) means the heat absorbed by the working fluid during the evaporation phase change, kW. The entropy difference of this process is \( \Delta s_{5,1} = s_{5} - s_{1} \), kJ kg\(^{-1}\)K\(^{-1}\).

2.2.2 | Performance criterion

The net power output of the ORC is regarded as the useful recovery of waste heat. A higher net power output indicates a better thermal performance of the ORC, and this output can be calculated by:

\[ W_{\text{net}} = W_{\text{th}} - W_p \]  

(11)

In this paper, we aim to determine the maximum net power output \( W_{\text{net}} \) to find the optimal pure working fluid, mixture working fluid, evaporation, and condensation pinch temperature difference and evaporation temperature of the ORC.

3 | CHOICE OF PURE WORKING FLUIDS AND ORC PARAMETERS

3.1 | Basis of calculation

The ORC recovery heat source thermodynamic parameters in this paper take the third-stage outlet of the compressor in
Table 2 as an example. In addition, the basic parameters of the ORC are described in Table 3.

When a working fluid and working conditions are selected for calculation, the values of the condensation and evaporation pinch temperature difference and evaporation temperature must be assumed first. Then, we determine the evaporation pressure and latent heat of vaporization of the working fluid, and the mass flow rate of the working fluid is also calculated at the same time. After that, the condensing pressure \( P_c \) of the ORC can be calculated by an iterative method, which is shown in Figure 3. The iterative process first calculates the cooling water mass flow rates \( m_{w1} \) of the 2-2\(_j\) process and \( m_{w2} \) of the 2\(_j\)-3 process separately. When the two values are very close, the iteration calculation is ended. Finally, the power of the expander, power of the working fluid pump, and net power output can be calculated.

### 3.2 Pure working fluid optimization

Thirteen pure working fluids are selected in this paper for study and are shown in Table 1. The working fluid must be environmentally friendly and safe, needs to have good flow and heat transfer performance, and should have a critical pressure and temperature within a reasonable range.

Exhaustive methods are usually used for the optimization of pure working fluids. This paper intends to construct a new parameter to approximate the net power output of different pure working fluids. In Figure 2, the area enclosed by 1-2-3-4-5-1 is the net output work of the pure working fluid per unit mass flow rate, which is represented by \( w_{\text{net}} \), kJ·kg\(^{-1}\). For the liquid phase of the working fluid, the power consumption of the pump is very low, and the area 3-4-5-3 is very small and can be ignored. Taking R227ea as an example, \( W_{\text{net}} \) is 80.91 kW for the first stage heat recovery of the compressor, and \( W_p \) is 5.8 kW, which can be calculated by Equation (8). The power consumption of the pump requires only 7.2%, which has little impact on the approximate calculations. When the expander efficiency is close to isentropic, the line 1-2 is close to vertical, and the area 1-2-3-2\(_j\)-1 can be regarded as a right triangle. After approximate simplification, we convert the net output work of the pure working fluid per unit mass flow rate into the area of a trapezoid 1-2-3-5-1, which is shown in Figure 4.

Thus, the approximate unit net power output can be described as follows:

\[
w'_{\text{net}} = \Delta s_{5-1} \cdot (T_5 - T_{2j}) + \frac{1}{2} \Delta s_{3-5'} \cdot (T_5 - T_{2j})
\]

where \( w'_{\text{net}} \) is an approximate value of \( w_{\text{net}} \), kJ·kg\(^{-1}\), and \( \Delta s_{5-1} \) = \( I_{s3-1} \) is the entropy difference of the 3-5' process, kJ·K\(^{-1}\).

The approximate net power output can be calculated by:

\[
W'_{\text{net}} = \left[ \Delta s_{5-1} \cdot (T_5 - T_{2j}) + \frac{1}{2} \Delta s_{3-5'} \cdot (T_5 - T_{2j}) \right] \cdot m_{\text{wf}}
\]

Combined with Equation (10), Equation (13) can be transformed to:

\[
W'_{\text{net}} = \left[ \Delta s_{5-1} \cdot (T_5 - T_{2j}) + \frac{1}{2} \Delta s_{3-5'} \cdot (T_5 - T_{2j}) \right] \cdot \frac{Q_{\text{in}}}{T_5 \cdot \Delta s_{5-1}}
\]

\[
= \frac{Q_{\text{in}}}{T_5} \cdot (T_5 - T_{2j}) + \frac{Q_{\text{in}}}{2} \cdot (T_5 - T_{2j})^2 \cdot \frac{1}{k \cdot \Delta s_{5-1}}
\]

where \( k \) is the slope between the saturated liquid phase points of the working fluid at the evaporating pressure and condensing pressure.

From Equation (14), \( Q_{\text{in}} \) is only affected by the heat source temperature, working fluid vaporization temperature, and pinch temperature difference. Before the optimal selection of pure working fluids, these values are already given. The condensation temperature of the working fluids can also be kept constant by controlling the flow rate of cooling water. Therefore, \( W'_{\text{net}} \) only changes with \((k \cdot \Delta s_{5-1})^{-1}\), and the relationship is linear. \( k \) and \( \Delta s_{5-1} \) are only related to the physical properties of working fluids.

Figure 5 verifies the accuracy of the approximation using an exhaustive method, and the specific calculation results are shown in Table 4. A pinch temperature difference of 5°C is given for both the condenser and evaporator, and the net power output of 13 pure working fluids with evaporation temperatures of 65°C and 60°C is calculated. In Figure 5, each point corresponds to a pure working fluid, and \( W_{\text{net}} \) and \((k \cdot \Delta s_{5-1})^{-1}\) are approximately linear, and according to the linear regression fits of the two sets of data, the coefficient of determination \( R^2 \) values are .99 and .98, respectively, which prove the goodness of the fit. Consequently, after the determination of the heat source temperature and the pinch temperature difference, the ORC can achieve a better net power output with a pure working fluid that has a higher \((k \cdot \Delta s_{5-1})^{-1}\) value.
3.3 | Optimal evaporation temperature

This paper studies the effect of increasing the evaporation temperature on the 13 pure working fluids. It is assumed that the pinch temperature difference of both the evaporator and condenser is 5°C and that the net power output varies with the evaporation temperature, as shown in Figure 6. The net power output $W_{\text{net}}$ has a maximum value, and the evaporation temperatures corresponding to the maximum net power output of the 13 pure working fluids are shown in Table 5. The same conclusion can be drawn from Table 4, whereby using R227ea as the pure working fluid of the ORC can yield a higher net power output.

In Figure 7, R227ea is used as an example to analyze the variation in exergy loss of each device with evaporation temperature. An increase in the evaporation temperature has little effect on the losses of the working fluid pump and the expander, and these losses account for a small percentage of the total losses. The exergy losses of the condenser and evaporator gradually decrease, but the waste heat not recovered by the ORC increases exponentially with the evaporation temperature. Before the evaporation temperature rises to the optimum value, the losses from heat exchangers are dominant, so the net power output increases. After that point, the heat that cannot be recovered by the ORC increases rapidly, and the net power output decreases.
3.4 Optimal pinch temperature difference

Taking the pure working fluid R227ea as an example, we compared the pinch temperature differences of 5, 10, and 15°C for the condenser and the evaporator. Figure 8 reflects the effect of the evaporation pinch temperature difference on the net power output. A higher pinch temperature difference results in a lower output power. Figure 9 shows that an increase in the pinch temperature difference will significantly increase the heat loss that the ORC cannot recover, and the other losses in the system do not change substantially. To obtain a higher cycle net output work and to satisfy the heat transfer temperature difference of the heat exchanger, 5°C is chosen as the pinch temperature difference of the evaporator for the calculation.

Figure 10 shows the effect of the condensation pinch temperature difference on the net power output, which is similar to the evaporator. However, as shown in Figure 11, a higher condensation pinch temperature difference will also lead to an increase in the loss of the condenser, in addition to increasing the amount of heat that the ORC cannot recover. Therefore, this paper also chooses 5°C as the condensation pinch temperature difference for the calculation.

4 Optimization of binary zeotropic mixture working fluids

A mixture composed of two or more kinds of pure working fluids with different boiling points is called a zeotropic mixture working fluid. The physical properties of mixtures composed of more than two kinds of pure working fluids have many uncertainties, so binary zeotropic mixtures are the main research objects of scholars. Because zeotropic mixtures have the thermodynamic characteristic of temperature glide, the thermal performance curve of the working fluid changes in the heat exchanger, as shown in Figure 12. $\Delta T_{ge}$ and $\Delta T_{gc}$ represent the temperature glide in the evaporator and condenser, respectively. Because the waste heat temperature between the compressor stages is approximately 100°C, the saturated evaporation temperature and condensation temperature of the ORC are very close, so it can be assumed that $\Delta T_{ge}$ is equal to $\Delta T_{gc}$.

In this system, the temperature difference of the heat source changes significantly more than that of the cooling water, and the evaporation temperature glide plays a small role in the optimization of mixture working fluids; thus, the condensation temperature glide $\Delta T_{gc}$ is a key point of our research.

As concluded in a previous study, the maximum net power output can be obtained by using the pure working fluid R227ea. A mixture of R227ea/isopentane is used

| Working fluid | $T_s = 60°C$ | $T_s = 65°C$ |
|---------------|--------------|--------------|
|               | $(k\Delta s_{5.1})^{-1} \times 10^3$ | $W_{net}/W$ | $(k\Delta s_{5.1})^{-1} \times 10^3$ | $W_{net}/W$ |
| R227ea        | 14.78        | 81 296.51    | 15.8          | 85 759.12     |
| Isobutane     | 9.37         | 76 693.65    | 9.76          | 79 605.16     |
| R114          | 9.45         | 77 278.80    | 9.79          | 80 094.22     |
| Butane        | 8.3          | 76 052.12    | 8.59          | 78 678.87     |
| R245fa        | 8.49         | 76 410.36    | 8.79          | 79 083.32     |
| R245ca        | 8.02         | 76 192.32    | 8.25          | 78 700.05     |
| R123          | 6.98         | 75 055.99    | 7.17          | 77 356.63     |
| Isopentane    | 7.85         | 76 234.95    | 8.08          | 78 676.40     |
| Pentane       | 7.46         | 75 896.74    | 7.66          | 78 259.32     |
| R113          | 6.96         | 75 419.63    | 7.12          | 77 655.16     |
| Hexane        | 7.15         | 75 801.38    | 7.32          | 78 055.55     |
| R236ea        | 9.97         | 77 793.64    | 10.35         | 80 754.94     |
| R236fa        | 10.88        | 78 311.64    | 11.37         | 81 569.33     |

| TABLE 4 | Calculation of $W_{net}$ for different pure working fluids |

![Figure 6](image.png) Net power output changes with evaporation temperature
as an example to analyze the effect of condensation temperature glide on the ORC performance by changing the mass fraction of R227ea, which is represented by \( X \). It is assumed that the evaporation temperature is 60°C and the pinch temperature difference is 5°C. The effect of the mass fraction of R227ea on the net power output and condensation temperature glide is shown in Figure 13. When \( X \) is 0-1, the working fluid is a pure component. A higher net power output can be obtained using the mixture working fluid than that with each pure component. The net power output has two maxima on the curve and reaches the maximum value when \( X = 0.9 \). The condensation temperature glide first increases and then decreases with increasing proportion of the R227ea component. A comparison of the two curves shows that the maximum value of the net power output does not occur in areas where the condensation

| Working fluids | \( T_e/°C \) | \( W_{net}/W \) |
|----------------|-------------|-------------|
| R227ea         | 67.73       | 86 403.76   |
| R236fa         | 66.25       | 81 701.69   |
| R236ea         | 65.89       | 80 820.09   |
| R114           | 65.73       | 80 136.88   |
| Isobutane      | 65.90       | 79 669.89   |
| R245fa         | 65.60       | 79 109.54   |
| R245ca         | 65.40       | 78 708.84   |
| Butane         | 65.57       | 78 701.34   |
| Isopentane     | 65.33       | 78 680.14   |
| Pentane        | 65.24       | 78 259.06   |
| Hexane         | 65.11       | 78 051.39   |
| R113           | 65.10       | 77 650.89   |
| R123           | 65.20       | 77 354.89   |

FIGURE 7 Exergy losses of each device varying with evaporation temperature

FIGURE 8 Effect of the evaporation pinch temperature difference on the net power output

FIGURE 9 Effect of the evaporation pinch temperature difference on the exergy loss of each device

FIGURE 10 Effect of the condensation pinch temperature difference on the net power output
The condensation temperature glide is high. The condensation temperature glide values corresponding to the two maximum points are close to the cooling water temperature rise of 10°C. This finding means that the optimal condensation temperature glide will lead to the same temperature difference between the inlet and outlet of the condenser, and this value is equal to the condensation pinch temperature difference. The heat exchange process is shown in Figure 14, where $\Delta T_w$ means the temperature rise of the cooling water.

Figure 15 shows the effect of the mass fraction of R227ea on the exergy loss of each device. The losses of the expander and the working fluid pump are still small; in addition, the waste heat not recovered by the ORC decreases with the increase in the R227ea component, and the minimum value is obtained when $X = 0.9$. Combining the analysis with Figure 13 shows that the exergy of the condenser is smaller when the condensation temperature glide is higher, but the exergy loss of the evaporator follows the opposite trend. As a result, there must be an optimal value of the condensation temperature glide that can achieve the highest net power output of the ORC.

All the other binary zeotropic mixtures have been calculated, and the same conclusion can be reached for each: There are two maximum points of the net power output with changes in the mixing ratio. One of those points has a larger value on the side of the pure component that has a better performance, and the corresponding condensation temperature glide is approximately equal to the cooling water temperature rise.

Another key point in studying binary zeotropic mixtures is the choice of two pure working fluids for mixing and their proportions. The parameter $(k \cdot \Delta s_{5-1})^{-1}$ is considered as a measure of the choice of pure working fluid. In the study of binary zeotropic mixture working fluids, we can also simplify the $T$-$s$ diagram of the ORC’s thermal process, as shown in Figure 16.

Compared to the simplified $T$-$s$ diagram of a pure working fluid, the diagram of a mixed working fluid has two more areas, 1-5-1’-1 and 3-2-3’-3, formed by the temperature glide, and these areas are also the reason why the mixture working fluid can obtain a higher net power output. The approximate unit net power output can be described as follows:

$$W'_{\text{net}} = \Delta s_{5-1} \cdot \left( T_5 - T_{3.j} \right) + \frac{1}{2} \Delta s_{3'-5'} \cdot \left( T_5 - T_2 \right) + \frac{1}{2} \Delta s_{3-5} \cdot \Delta T_{ge}$$

(15)

where $w'_{\text{net}}$ is the approximate unit net power output of the ORC with a binary zeotropic mixture working fluid, kJ kg$^{-1}$, and $\Delta T_{gc}$ and $\Delta T_{ge}$ are the condensation and evaporation temperature glide, respectively, K. In this paper, the entropies of 1 and 1’ are very close, and the assumptions that the 1, 2t, and 1’ points are on the same isentropic line and $\Delta T_{gc} = \Delta T_{ge}$ are made before giving the following equation of $W'_{\text{net}}$:

$$W''_{\text{net}} = \frac{Q_{in} \cdot (T_5 - T_{2j})}{T_5} + \frac{Q_{in} \cdot (T_5 - T_{2j})^2}{2T_5} \cdot \frac{1}{k \cdot \Delta s_{5-1}} + \frac{Q_{in} \cdot (T_5 - T_{2j}) \cdot \left( \frac{1}{2k \cdot \Delta s_{5-1}} + 1 \right)}{T_5} \cdot \Delta T_{ge}$$

(16)

From Equation (16), the approximate net power output of the ORC with a binary zeotropic mixture working fluid is affected by the parameters $\Delta T_{ge}$ and $(k \cdot \Delta s_{5-1})^{-1}$. According to previous conclusions, the optimal condensation temperature glide needs to be equal to the cooling water temperature rise, which is 10°C in this paper. When this condition is satisfied, the mixture working fluid with the largest $(k \cdot \Delta s_{5-1})^{-1}$ value is optimal.
Two binary zeotropic mixtures of R227ea/R236fa and R227ea/R123 are taken as examples. Table 6 shows the relationship between the \((k \cdot \Delta s_{5-1})^{-1}\) parameter of a mixture working fluid and the mixing ratio. It can be discovered that the \((k \cdot \Delta s_{5-1})^{-1}\) value of a zeotropic mixture working fluid is between the values of the two pure working fluids. To obtain the mixture working fluid with a higher \((k \cdot \Delta s_{5-1})^{-1}\) value, it is necessary to choose the pure working fluid with the largest \((k \cdot \Delta s_{5-1})^{-1}\) value. Therefore, the pure working fluid R227ea, which has the largest \((k \cdot \Delta s_{5-1})^{-1}\) value, is chosen and can form 11 binary zeotropic mixtures with the remaining pure working fluids. The changes in the condensation temperature glide of the 11 binary zeotropic mixtures with the \((k \cdot \Delta s_{5-1})^{-1}\) value are shown in Figure 17. The \((k \cdot \Delta s_{5-1})^{-1}\) value of each mixture working fluid is calculated in Table 6.
value of the R227ea/hexane mixture is the largest when the condensation temperature glide is equal to the cooling water temperature rise of 10°C, and the corresponding mixing ratio can achieve the maximum net power output.

After determining the optimal binary zeotropic mixture working fluid, the optimal evaporation temperature can be found by iterative calculation. A previous study showed that the maximum net power output of the ORC exists with changes in the saturated evaporation temperature \( T_e \), and the iterative process is shown in Figure 18.

Table 7 shows a comparison of the multistage centrifugal compressor waste heat recovery between using the optimal pure working fluid and the binary zeotropic mixture working fluid. The best mixture working fluid is R227ea/hexane (0.98/0.02), and the net power output is 424.63 kW, which is 29.3% higher than that of the best pure working fluid R227ea.

### TABLE 6 Distribution of \( (k \cdot \Delta s_{5.1})^{-1} \) with increasing of R227ea mass fraction

| \( X \) | R227ea/R236fa | R227ea/R123 |
|---|---|---|
| 0 | 10.88 | 6.98 |
| 0.1 | 11.21 | 7.31 |
| 0.2 | 11.56 | 7.67 |
| 0.3 | 11.94 | 8.08 |
| 0.4 | 12.34 | 8.56 |
| 0.5 | 12.76 | 9.11 |
| 0.6 | 13.19 | 9.77 |
| 0.7 | 13.62 | 10.58 |
| 0.8 | 14.04 | 11.61 |
| 0.9 | 14.43 | 12.96 |
| 1 | 14.78 | 14.78 |

### FIGURE 17 Condensation temperature glide changes with the \( (k \cdot \Delta s_{5.1})^{-1} \) value for the mixture working fluids

### FIGURE 18 Optimal evaporation temperature iterative calculation process

### TABLE 7 Comparison of multistage centrifugal compressor waste heat recovery between pure working fluid and mixed working fluid

| Parameter | 1st stage | 2nd stage | 3rd stage | 4th stage | Total |
|---|---|---|---|---|---|
| Outlet | | | | | |
| \( T_e/°C \) | 98 | 102 | 100 | 93 |
| \( P/kPa \) | 164 | 261 | 406 | 610 |
| \( h/kJ \cdot kg^{-1} \) | 497.96 | 501.87 | 499.65 | 492.26 |
| Inlet | | | | | |
| \( T_e/°C \) | 45 | 45 | 45 | 45 |
| \( P/kPa \) | 158 | 256 | 402 | 610 |
| \( h/kJ \cdot kg^{-1} \) | 444.46 | 444.26 | 443.97 | 443.56 |
| Heat power | | | | | |
| \( \Delta h/kJ \cdot kg^{-1} \) | 53.5 | 57.61 | 55.68 | 48.7 |
| \( Q/kW \) | 1632.82 | 1758.26 | 1699.35 | 1486.32 | 6576.75 |
| Pure working fluid | R227ea | | | | |
| \( m_{w}/kg \cdot s^{-1} \) | 9.98 | 10.81 | 10.39 | 8.97 | 40.16 |
| \( T_e/°C \) | 66.57 | 69.26 | 67.9 | 63.41 |
| \( W_{net}/kW \) | 80.91 | 93.79 | 87.19 | 66.51 | 328.40 |
| Mixed working fluid | R227ea/hexane (0.98/0.02) | | | | |
| \( m_{w}/kg \cdot s^{-1} \) | 9.88 | 10.87 | 11.24 | 8.6 | 39.59 |
| \( T_e/°C \) | 60.3 | 63.51 | 61.75 | 57.62 |
| \( W_{net}/kW \) | 105.06 | 118.79 | 111.87 | 88.91 | 424.63 |

### CONCLUSIONS

Tremendous waste heat generation occurs during the interstage cooling of multistage centrifugal compressors. In this paper, a model using an ORC to recover this waste heat for electric power generation is established. To obtain the
maximum net power output, 13 pure working fluids are selected to study the effect of their parameters on the performance of the ORC, and the optimal binary zeotropic mixture is studied based on these fluids. A parameter, \((k \cdot \Delta s_{5,1})^{-1}\), is built as the standard for the selection of the best pure working fluid and binary zeotropic mixture working fluid. In addition, the condensation temperature glide must be close to the cooling water temperature rise to obtain a larger net power output.

The main conclusions are as follows:

1. Among the 13 pure working fluids, R227ea has the largest \((k \cdot \Delta s_{5,1})^{-1}\) value and can make the ORC recover the highest net power output of 328.4 kW in this system.
2. The net power output of the ORC has a maximum value with increasing evaporation temperature, and the pinch temperature differences of the evaporator and condenser are chosen as 5°C to obtain a higher net power output and satisfy the heat transfer temperature difference of the heat exchanger.
3. When the condensation temperature glide is equal to the cooling water temperature rise of 10°C, the binary zeotropic mixture R227ea/hexane (0.98/0.02) has the largest \((k \cdot \Delta s_{5,1})^{-1}\) value and a net power output of 424.63 kW in this system; this value is 29.3% higher than that of R227ea. As a result, using a binary zeotropic mixture as the working fluid is more efficient than using a pure working fluid in this system.

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