Performance optimization of high-temperature heat pump system for staged heating under large temperature span

Zhang Hao\textsuperscript{a}, Zhang Yanting\textsuperscript{a}\textsuperscript{*}, Wang Lin\textsuperscript{b}, Xu Jingyu\textsuperscript{c}, Huang Lumeng\textsuperscript{a}, Huang Zheng\textsuperscript{a}, Zhang Guangzhia

Dr. Zhang Hao, Prof. Zhang Yanting, Huang Zheng
Department of Mechanical and Electrical Engineering, China University of Petroleum (East China), Qingdao 266580, Shandong, China;
Email: ytzhang@upc.edu.cn
Prof. Wang Lin
Henan University of Science and Technology, Luoyang 471000, Henan, China
Dr. Xu Jingyu
Shanghai Haomu Energy Saving Technology Co., Ltd., Shanghai 200335, China;

Abstract: With the rise in the widespread application of high-temperature heat pump (HTHP) systems, the system temperature across the large volume, high condensing temperature, and sink unbalanced proportion of heat supply, and other characteristics of the HTHP system need to be optimized as the primary solution. To improve the thermodynamic performance of the system at high temperatures across the process, the effects of the intermediate cooling structure circulation mode and hierarchical heating on the HTHP system were analyzed in this work. Furthermore, the HTHP system was optimized by adjusting the appropriate diversion coefficients in order to improve its coefficient of performance (COP). The optimum value of COP and the corresponding coefficients of diversion for the different process requirements were calculated using R245fa as the refrigerant and perturbing the coefficients of diversion in combination with a simulated annealing algorithm. When the condensing temperature was between 120–140 °C, the COP of the optimized system was observed to be 15.93–20.48% higher than that of the traditional two-stage compression system, and the system was found to exhibit a significant unit heat production and refrigeration capacity.

1. Introduction
The high-temperature heat pump (HTHP) technology is one of the most promising technologies used for replacing calcining boilers to achieve high-temperature thermal product processing. This technology is not only environmentally friendly but also has high heating efficiency. Bergamini et al. [1] compared the thermodynamic performance of various vapor compression cycle systems with natural gas boilers based on HTHP. They pointed out that the HTHP system is a promising alternative to the traditional boiler calcination technology. In their research, Pan L et al. [2] showed that the HTHP system consumed less amount of energy and caused less pollution as compared to the traditional calcining boiler process. Further, HTHPs have a relatively rich heat source foundation in real applications. Fitó et al. [3] studied the benefits of HTHP technology at three industrial waste heat temperatures and achieved favorable results in different applications. Chaturvedi et al. [4] studied the two-stage direct-expansion solar-assisted heat pump systems, using solar energy as the heat source to obtain hot water at 90 °C. Zhang et al. [5] used geothermal energy as the heat source to analyze the performance of an HTHP system with R245fa and R152a azeotropic working fluids and conducted actual tests.
Nevertheless, in warming up the hot water, the high-temperature heat pump maintains a stable condensation temperature in the condenser by consuming latent heat. When the thermal product's initial temperature is low, the system has a heat exchange process with a large temperature difference, which also causes a large exergy loss in this process.

Considering the process application range of HTHPs, in this study, in response to the process requirements of the HTHP without phase change of the heat sink, an HTHP system structure with parallel double condenser characteristics has been established. The parallel compressors classify the heating temperature area so that the heat sink can achieve a step-by-step heat exchange under multiple temperature gradients. By setting up two-stage splits, the influence of the inter-cooling method on the HTHP system and the optimal heat supply ratio can be verified by studying the HTHP system within different temperature thresholds. In addition, the relationship between the suction superheat, and the exhaust superheat before and after refrigerant compression was established in order to avoid the liquid hammer condition and the supercritical heat exchange in the compressor. As a result, the system was observed to exhibit superior thermodynamic performance.

2. The system model
Due to the large temperature rise process, the heat pump system has a larger temperature span of the heated object in the cold utility. In addition, the main energy of the refrigerant in the heat pump system comes from its latent heat, which makes the temperature difference between the refrigerant and the heated working fluid have a large initial temperature difference, resulting in a large heat exchange loss. Therefore, it is hoped to use the graded heating method to heat the cold public works.

![System Diagram](image.png)

Fig 1. Schematic showing the structure of the multistage sink heating HTHP system.

The principle of the design system is as follows. Two different condensing temperatures, $T_{\text{cond1}}$ and $T_{\text{cond2}}$, are formed in parallel by the two compressors of different powers, and the refrigerant is split via the split valve group to control the heat exchange ratio in the different temperature ranges. A flash tank (FT) is used as the intermediate air supplement device, in which the saturated liquid refrigerant can preheat the heat sink and supercool itself. So far, a three-stage heat exchange process in different temperature areas has been realized in the system. The structure of the multistage sink heating HTHP system has been shown in Fig 1.

2.1. Model assumptions
The following assumptions have been made while building the model of the multistage HTHP system:
1. The pressure loss during the refrigerant cycle has been assumed to be negligible.
2. There is no energy loss during heat transfer within the system.
3. The temperature of the heat source is constant at 60 °C and the heat supply has a continuous value.
4. To analyze the relationship between the degree of subcooling clearly, subcooler-1 and subcooler-2 have a degree of subcooling of 5 °C, and condenser-1 and condenser-2 make the refrigerant have a degree of subcooling of 5 °C.
5. The isentropic efficiency of the compressor is a linear empirical model proposed by Brown et al.

6. The diverter valve group has a relatively accurate diverging performance.

2.2. Analysis of the parameters of the system model

In the system shown in Fig 2, the coupling strength of the system is adjusted via two shunt coefficient variables, x and y. x is used for adjusting the heating ratio of the system at temperatures Tcon1 and Tcon2. The function of the shunt coefficient y is to adjust the coupling strength of the intermediate cooling of the refrigerant. Fig 4 shows the pressure–enthalpy diagram of the system for different splitting coefficients.

Let the mass flow rate of the refrigerant flowing through compressor-1 be $m_1$ and that flowing in the flash evaporator be $m_2$. Let the gas ratio be $a$. Then, according to the conservation of mass, the relationship between $m_1$ and $m_2$ is $m_1 = (1-a)m_2$. Further, let $m_3$ represent the total mass flow of the refrigerant before the refrigerant is divided into compressor-2 and compressor-3. Thus, it can be expressed as:

$$m_3 = a \cdot m_2 + (1-a) \cdot m_1 = \left(\frac{1}{1-a} - y\right) \cdot m_1 \tag{1}$$

According to the conservation of energy, the gas ratio, $a$, in the FT can be calculated by combining $x$ and $y$ with the specific enthalpy of the refrigerant at the inlet of the flash evaporator as follows:

$$a = \frac{(x - xy) \cdot h_{\text{sub-co1}} + (1-x-y+xy) \cdot h_{\text{sub-co2}} + y \cdot h_{\text{comp1-out}} - h_{\text{FTl}}}{h_{\text{FTv}} - h_{\text{FTl}} - (xy \cdot h_{\text{sub-co1}} + (y-x-y) \cdot h_{\text{sub-co2}} - y \cdot h_{\text{comp1-out}})} \tag{2}$$

where $h_{\text{sub-co1}}$ is the specific enthalpy of subcooler-1 after subcooling (in kJ/kg), $h_{\text{sub-co2}}$ is the specific enthalpy of subcooler-2, (in kJ/kg) after subcooling, $h_{\text{comp1-out}}$ is the specific enthalpy of the discharged refrigerant at compressor-1 (in kJ/kg), $h_{\text{FTv}}$ is the specific enthalpy of the saturated gaseous refrigerant in the flash evaporator (in kJ/kg), and $h_{\text{FTl}}$ is the specific enthalpy of the saturated liquid refrigerant in the flash evaporator (in kJ/kg).

According to assumption 3, when the heat supply is constant, $m_1$ only changes with a change in the flash pressure, $p_{\text{FT}}$, which is expressed as follows:
\[ m_i = m_{Hs} \frac{h_{Hs-in} - h_{Hs-out}}{h_{comp1-in} - h_{HE}} \]  

(3)

where \( h_{Hs-in} \), \( h_{Hs-out} \), \( h_{comp1} \), and \( h_{HE} \) are the specific enthalpies of the heat source inlet, heat source outlet, compressor-1 suction refrigerant, and the refrigerant after passing through the heat exchanger (HE) respectively (in kJ/kg).

The above system structure has three compressors with different pressure levels. Compressor-1 compresses the refrigerant from the evaporation pressure, \( p_{eva} \), to the flash pressure, \( p_{FT} \), and the pressure ratio in this case is \( \lambda_1 \). Compressor-2 compresses the refrigerant using the flash pressure, \( p_{FT} \), to the second condensing pressure, \( p_{con2} \), leading to a pressure ratio of \( \lambda_{21} \). Compressor-3 compresses the refrigerant from the flash pressure, \( p_{FT} \), to the first condensing pressure, \( p_{con1} \), and the pressure ratio in this case is \( \lambda_{22} \). The relationship between the four pressure systems can be expressed as \( p_{con1} > p_{con2} > p_{FT} > p_{eva} \). The work done by the three compressors is \( W_1 \), \( W_2 \), and \( W_3 \), which is expressed by Equations (4)–(6):

Compressor-1:  
\[ W_1 = m_i \cdot (h_{comp1-out} - h_{comp1-in}) \]  

(4)

Compressor-2:  
\[ W_2 = (1 \cdot x) \left( \frac{1}{1-a} - y \right) \cdot m_i \cdot (h_{comp2-out} - h_{comp2-in}) \]  

(5)

Compressor-3:  
\[ W_3 = \left( \frac{1}{1-a} - y \right) \cdot m_i \cdot (h_{comp3-out} - h_{comp3-in}) \]  

(6)

The heat of the three heat exchanges is \( Q_1 \), \( Q_2 \), and \( Q_3 \), which is expressed by Equations (7)–(9).

HE preheating:  
\[ Q_1 = m_i \cdot (h_{HE} - h_{FT}) \]  

(7)

Two-stage heat transfer:  
\[ Q_2 = (1 \cdot x) \left( \frac{1}{1-a} - y \right) \cdot m_i \cdot (h_{comp2-out} - h_{sub-co2}) \]  

(8)

Three-stage heat transfer:  
\[ Q_3 = \left( \frac{1}{1-a} - y \right) \cdot m_i \cdot (h_{comp3-out} - h_{sub-co1}) \]  

(9)

According to model assumption 4, the suction superheat of compressor-1 can be controlled by the evaporator. The suction superheat of compressor-2 and compressor-3 is adjusted according to the split coefficient, \( y \). Combined with the conservation of mass and energy, the specific enthalpy, \( \Delta h_{sub-in-23} \), of the suction superheat of compressor-2 can be expressed as:

\[ \Delta h_{sub-in-23} = (a + y + ay - 1) \cdot h_{FT} + (1-a-y+ay) \cdot h_{comp1-out} \]  

(10)

The main characteristic parameters of the new system that distinguish it from the traditional system are the two diversion coefficients \( x \) and \( y \). These coefficients affect the overall COP of the system, the optimal flash pressure, \( p_{FT} \), and the exhaust superheat. The two shunt coefficients themselves have a mutual coupling effect.

3. Results and discussion

3.1. Heating ratio and process analysis

Depending on the heat pump process involved, the system involves three heat exchange processes for the heat sink. Thus, the heat exchange interval of the three heat exchange processes is defined as \( T_{FT} - T_{sink1} \) (the first heat exchange), \( T_{con2} - T_{sink2} \) (the second heat exchange), \( T_{con1} - T_{sink3} \) (the third heat exchange), where \( T_{sink1} \), \( T_{sink2} \), and \( T_{sink3} \) are the upper limits of the temperature of the heat sink in the three heat transfers. \( T_{con2} \) was considered to be a variable, \( T_{con1} \) was set to 130 °C, and \( T_{eva} \) was set to 50 °C. From the above analysis, the shunt coefficient, \( y \), was taken to be 0, and the optimal COP solution that can be achieved by the HTHP system with \( x \) in the range of 0–1 was calculated using the model of the system (the COP only considers the amount of heat exchanged by the system and does not consider the heat exchange temperature limit for the time being). The changing trend of the three
heat exchanges $Q_1$, $Q_2$, and $Q_3$ was as shown in Fig 3.

Fig 3. Effect of the shunt coefficient, $x$, on the COP and heating ratio of the system.

From the Fig 3, it can be seen that as the shunt coefficient, $x$, increases, the COP of the system decreases from 7.09 to 3.23 from the perspective of the overall energy consumption. Simultaneously, the heat supply of the system at different heating intervals will change significantly. As $x$ increases from 0 to 1, the saturated liquid-phase refrigerant temperature in the flash evaporator increases from 67.9 °C to 87.84 °C, which dramatically reduces the limit on the initial temperature of the heat sink.

Taking $x$ to have a fixed value of 0.8 and setting the maximum condensation temperature of the system to be variable, the resulting heat supply ratio of the three heat exchanges and the change in the COP of the system as the condensation temperature increases are as shown in Fig 4.

Fig 4. Effect of the condensing temperature, $T_{con1}$, on the COP and the heating ratio of the system.

With an increase in the maximum condensing temperature, $T_{con1}$, the three-stage heat exchange of the system shows a slight upward trend. The increments of $Q_1$, $Q_2$, and $Q_3$ are 19.32%, 22.15%, and 9.87%, respectively. From this, it can be inferred that under the same working conditions, as the condensation temperature increases, the increasing rate of heat exchange in $Q_1$, $Q_2$, and $Q_3$ will affect the heat supply ratio of the system to the heat sink.

To further enhance the applicability of the system, it is necessary to consider the influence of the different types and temperatures of the heat sources on the system. To investigate this, the maximum condensing temperature, $T_{con1}$, of the system was set to a constant value of 130 °C, and the shunt coefficient, $x$, was taken to have a constant value of 0.8. The evaporation temperature was gradually increased from 30 °C to 60 °C and the changing trend of the three-time heat supply of the system was obtained, as shown in Fig 5.

As can be seen from the Fig 5, as $T_{eva}$ increases, the total pressure ratio in the system decreases, and the COP increases from 2.68 to 4.4. The corresponding values of the heat supply, $Q_2$, and $Q_3$, show a decreasing trend, decreasing by 25.4% and 24.2%, respectively.

Thus, it can be inferred that reducing the shunt coefficient, $x$, can increase the COP of the system.
3.2. Analysis of the performance of the optimized system

From the above partial analysis of the relationship between the superheat and the heating ratio of the system, it is observed that the optimization of the system involves three limiting conditions:

1. The refrigerant compression having the wet working fluid characteristics must have a certain degree of inhalation superheat to prevent liquid shock during the process.
2. The exhaust superheat should be controlled in order to avoid supercritical heat exchange.
3. A reasonable shunt coefficient should be chosen so that the heat supply ratio of the three heat exchanges and the heating temperature range of the system meet the requirements of the multistage heat absorption process of the heat sink.

Under the premise of these restrictive requirements, the thermodynamic performance of the system should be improved as much as possible. In order to produce a better COP solution, combined with the simulated annealing algorithm, the values of the range of $x$ and $T_{comp1-sub-out}$ were taken to be variables. Furthermore, by adding a specific range of disturbance to the variable to calculate the system COP, the COP solutions in multiple systems were compared to find the optimal solution.

Taking the production of high-pressure hot water without a phase change as the process goal, since the pressure does not change, the specific enthalpy required for hot water heating has a linear relationship with the rise in temperature of the hot water. Therefore, the proportional relationship between Q1, Q2, and Q3 must satisfy the linear relationship of $C_p$ to meet the process requirements. The trend of the change in $x$ was obtained for the systems with different values of $T_{con1}$, as shown in Fig 6.

From the Fig 6, it can be seen that as $T_{con1}$ increases, the optimal COP that satisfies the requirement of the working of the system is reduced from 4.53 to 3.49. When $T_{con1}$ is in the range of 120–130 °C, the decreasing trend of $x$ is relatively slow. When $T_{con1}$ is in the range of 130–140 °C, the decreasing trend of $x$ increases sharply due to the small change in the heat ratio required to produce hot water under the no phase change condition.

By comparing the unit cooling and unit heating capacities of the traditional and the optimized systems at different condensing temperatures, a comparison of the effects of the staged heating structure on the compressor selection and the output of thermal products was obtained, as shown in Fig 7.

Fig 6. Values of COP of the optimized system and the corresponding values of $x$ as a function of the condensing temperature.

Fig 7. Changes in unit cooling and unit heating capacities as a function of the condensation temperature.

From the Fig 7, it can be observed that an increase in $T_{con1}$ leads to a decrease in the unit cooling and unit heating capacities of the two systems. However, the magnitudes of both capacities for the optimized system are always higher than the traditional system. When $T_{con1}$ increases from 120 °C to 140 °C, the unit cooling capacity of the optimized system decreases from 66.0 kJ/kg to 54.7 kJ/kg, which is a decrease that is 11.06–16.98% higher than the traditional system. As a result, the HTHP system consisting of a staged heating structure requires a lower quantity of circulating refrigerant as compared to the traditional system under the same cooling capacity condition and reduces the rated power of the compressor.
4. Conclusion
In this study, a multistage sink heating HTHP system was used for classifying the heating temperature. The ratio of the heat supply at different temperatures was controlled by the shunt valve group, which reduced the phenomenon of low utilization energy efficiency. The following are the key points of this work:

1. An investigation of the influence of the inter-cooling mode and staged heating on the performance of the HTHP system was carried out using the split coefficients $x$ and $y$. The shunt coefficient $x$ determined the heat supply ratio for the different temperature gradients of the system. The main parameter $y$ determined the degree of cooling of the refrigerant in the system and expanded the applicability of the system under conditions.

2. When R245fa was used to produce hot water, the circulation method with an incomplete cooling structure enabled to achieve a high COP of the system. This method enabled the refrigerant to enter the high-pressure stage compressor with a certain degree of suction superheat; thus, effectively reducing the compression risk of the hydraulic shock of the machine.

3. The ratio of the three heat exchanges of the system was adjusted via the shunt coefficient $x$. This ratio must meet the heat absorption ratio of the sink in the different temperature ranges when obtaining the optimal COP. This ratio is only available under the constant pressure condition. It was observed to have a linear relationship with the temperature gradient. The optimized system was compared with the traditional system at different condensing temperatures. The results thus obtained showed that the COP of the optimized system increased by 15.93–20.48%, and the unit cooling and unit heating capacities increased by 11.06–16.98% and 7.54–13.76%, respectively under the same working conditions.

Acknowledgments
Funding for projects are jointly funded by the National Natural Science Foundation of China (51804333) and Graduate Innovation Project Funding Project (YCX2019064).

References
[1] Bergamini R, Jensen JK, Elmegaard B: Thermodynamic competitiveness of high temperature vapor compression heat pumps for boiler substitution. Energy 2019, 182:110-121.
[2] Pan L, Wang H, Chen Q, Chen C: Theoretical and experimental study on several refrigerants of moderately high temperature heat pump. Applied Thermal Engineering 2011, 31(11-12):1886-1893.
[3] Fitó J, Hodencq S, Ramousse J, Wurtz F, Stutz B, Debray F, Vincent B: Energy- and exergy-based optimal designs of a low-temperature industrial waste heat recovery system in district heating. Energy Conversion and Management 2020, 211.
[4] Chaturvedi SK, Abdel-Salam TM, Sreedharan SS, Gorozabel FB: Two-stage direct expansion solar-assisted heat pump for high temperature applications. Applied Thermal Engineering 2009, 29(10):2093-2099.
[5] Zhang SJ, Wang HX, Guo T, Chen C: Performance Simulation and Experimental Testing of Moderately High Temperature Heat Pump Using Non-azeotropic Mixture for Geothermal District heating. 2010 Asia-Pacific Power and Energy Engineering Conference (Appeec) 2010.