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Performance simulation and exergy analysis on multi-stage compression high temperature heat pumps with R1234ze(Z) refrigerant

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Abstract

In this paper, the system simulation and exergy analysis of multi-stage compression high temperature heat pump (HTHP) systems with R1234ze(Z) working fluid are conducted. Both the single-stage and multi-stage compression cycles are analysed to compare the system performance with 120 ℃ pressurized hot water supply based upon waste heat recovery. Energy and exergy efficiencies of the systems are calculated to study the influences of operating conditions on the system performance. Variation of the HTHP performance with increasing waste heat source temperature is investigated. The exergy destruction ratios of every component for different stage compression systems are compared. The results showed the exergy loss ratios of compressor is big than that of evaporator and condenser for single-stage compression system. The multi-stage compression system has better energy and exergy efficiencies with the increase of compression stage number. Compared with single-stage compression system, the COP improvements of two-stage and three-stage compression system are 9.1% and 14.6%, respectively. When the waste heat source temperature is 60 ℃, the exergy efficiency increase about 6.9% and 11.8% for two-stage and three-stage compression system respectively.

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Keywords: multi-stage compression; high temperature heat pump; heat recovery; exergy destruction; R1234ze(Z) refrigerant

1. Introduction

A heat pump system uses some amount of external power to accomplish the work of transferring energy

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from a low temperature heat source (waste heat for example) to a high temperature heat sink (user side). According to the various low temperature heat sources, heat pumps can be classified as air source, ground source, water source, hybrid source and solar-assisted heat pump. Due to its feature of temperature lift and high energy efficiency, heat pump system has been extensively used for indoor space heating and also hot water heating. With the development of science and technology, the new applications of heat pump are explored and validated. Recently industry heating has been considered very important, as a heat pump could be even used to upgrade thermal energy, industrial waste heat could be possibly turned as useful thermal energy for industry processes.

Water-source heat pump (WSHP) technologies such as the geothermal water-source heat pumps and seawater water-source heat pumps are energy-saving methods that have been used in commercial and residential heating applications [1]. Also, the WSHP with waste heat recovery has better prospects in industrial applications for its high efficiency and high temperature output. Survey results of temperature range requirements in some industries revealed that most of the temperature required in industry are above 60 ℃, for example, wood drying, food and beverage production, dyeing process, district heating and crude oil heating [2, 3]. Watanabe [4] summarized the temperatures of heat demand in different types of industrial processes and divided them into air, water and steam applications. The heat sinks with temperature from 50 ℃ to 125 ℃ are required in general machinery, electric components, transport equipment, food and so on. In the most industrial fields, coal-fired boilers, gas-fired boilers, oil-fired boilers and electric boilers are often used for hot water supply and process heating. However, these heating equipment waste more primary energy and cause environmental pollution problems for the low efficiency and combustion emissions. Meanwhile, large amounts of low grade heat can be recovered to satisfy the requirements of production and living for useful purposes. HTHP offers a most practical solution to those problems. They recover heat from industrial waste and produce high temperature hot water. The most important of all, high temperature heat pumps have proven to be an effective way to reduce greenhouse gases emission.

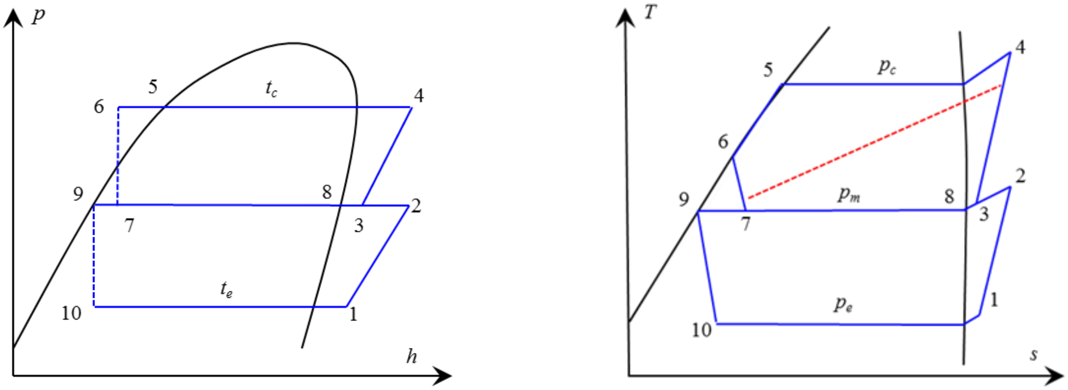
Li et al. [5] studied the system performance of a water-to-water HTHP unit with R22/R141b mixture to provide hot water with temperature of 80 ℃. The highest hot water temperature could reach 85 ℃. Liu et al. [6] developed a water-to-water heat pump system with HTR01 refrigerant to recover waste heat with temperature of 30-60 ℃ and provide moderately high-temperature hot water with temperature of 70-90 ℃. Brown et al. [7] proposed fluorinated olefin R-1234ze(Z) as a possible high-temperature refrigerant for HTHP and estimated the refrigerant thermal properties and potential system performances. Wang et al. [8] developed a heat pump system with heat recovery in building heating and experimentally studied the system performance. The hot water temperature was reported around 85 ℃ in most of the test conditions. Also, Wang et al. [9] further investigated a HTHP unit using parallel cycles with serial heating on the water side at temperatures ranging from 75 ℃ to 85 ℃. Pan et al. [10] evaluated several moderately high-temperature refrigerants and found that a zeotropic mixture refrigerant HC600/HFC245fa was a good option for the HTHP. The system achieved a COP of 3.83 at an evaporating temperature of 55 ℃ and condensation temperature of 100 ℃. Chamoun et al.[11] developed a dynamic model for an industrial heat pump using water as refrigerant and the reported temperature range was around 120-130 ℃.

In order to achieve the large temperature lift with higher temperature output, the exergy loss in compression and expansion process for the single-stage compression system is very high, which lowers the compression efficiency and degrades the system performance. As a result, a multi-stage compression design, for example, multi-stage heat pump system is developed to overcome those problems.

In this paper, multi-stage compression heat pump systems with R1234ze(Z) refrigerant are investigated to improve the thermal performance and system exergy efficiency. Thermodynamic simulation approach of multi-stage compression heat pumps is conducted to evaluate the system performance. Based on exergy analysis, the energy destruction at each separated component is developed to get the better performance as well as improvement potential of proposed systems. Thermodynamic analysis is conducted based on the variation of waste heat source temperature. The power consumption, system COP and exergy efficiency under different working conditions are analysed. The exergy destruction ratio of each process is compared and validated with reference results.

2. Multi-stage Compression Heat Pump Systems

Figure 1 shows the p-h and T-s diagrams of the two-stage vapour compression heat pump system. The working principle is as follows: the liquid refrigerant in the evaporator absorbs heat from the waste water and vaporizes. The super-heated refrigerant vapour is compressed to an intermediate temperature and intermediate pressure gas by the lower-stage compressor. Then the discharge gas from the lower-stage compression is mixed with the intermediate pressure vapour refrigerant from the flash tank. The mixed vapour enters the upper-stage compressor for the second-stage compression. The high temperature and pressure refrigerant gas from upper-stage compression flows into the condenser where it exchanges heat with the water from the water supply system and becomes liquid refrigerant. The water in the condenser is heated up after absorbing heat from the refrigerant gas and circulates back to the water supply system. And then the saturated liquid refrigerant is further cooled down in the subcooler. The high pressure refrigerant from the subcooler is throttled by the upper-stage expansion valve and becomes a liquid-gas mixture of intermediate pressure. The refrigerant mixture is then separated into liquid phase and vapour phase in the flash tank with intermediate pressure. The liquid refrigerant is further throttled by the lower-stage expansion valve and becomes a liquid-gas mixture of low pressure and low temperature. The vapour refrigerant of intermediate pressure is mixed with the discharge gas from the lower-stage compression. Finally, the low pressure refrigerant mixture flows back to the evaporator where it absorbs heat from waste water and vaporizes for next cycle.



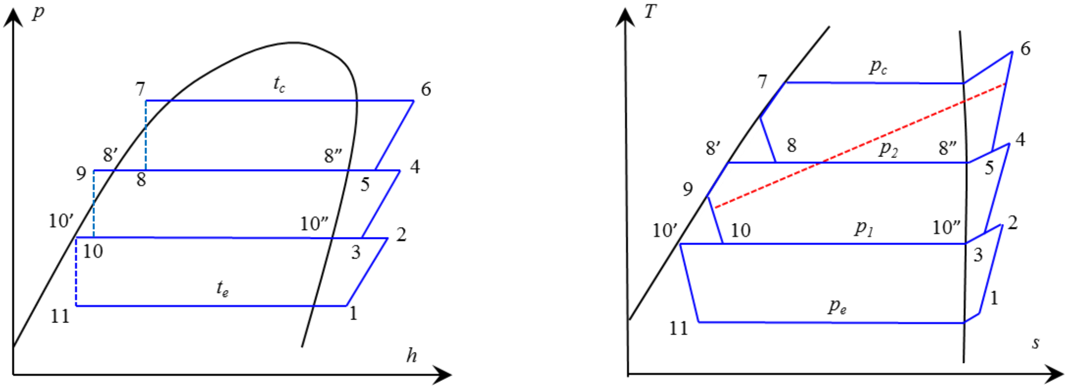
(a) P-h diagram

(b) T-s diagram

Fig. 1. p-h and T-s diagrams of two-stage compression heat pump system

From Figure 1 it also can be seen that due to the two-phase separation in the flash tank, the liquid entering the evaporator has lower enthalpy compared to that of a single-stage cycle. Thus the enthalpy difference across the evaporator is greater than that of a single-stage cycle. The saturated vapour from the flash tank also has lower temperature than that of the vapour in the compressor, which helps to reduce the compressor discharge temperature. Reduced compressor power consumption leads to higher system COP.

Figure 2 shows the p-h and T-s diagrams of the three-stage compression heat pump system. The working principle is as follows: the refrigerant from evaporator is compressed in the first-stage and then mixed with the vapour refrigerant of the first-stage pressure. The mixed vapour enters the compressor for the second-stage compression. After mixing with the vapour refrigerant of the second-stage pressure, the mixed refrigerant is further compressed in the third-stage. And then the refrigerant is cooling down in the condenser and flows through the third-stage expansion valve. The refrigerant is separated into liquid phase and vapour phase in the flash tank II. The vapour refrigerant of second-stage pressure is mixed with the discharge gas from the second-stage compression. The liquid refrigerant is further cooling down in the subcooler and enters the second-stage expansion valve. The refrigerant is separated again in the flash tank I. The vapour refrigerant of first-stage pressure is mixed with the discharge gas from the first-stage compression. The liquid refrigerant enters the first-stage expansion valve and then is heated by the waste heat in the evaporator.



(a) p-h diagram

(b) T-s diagram

Fig. 2. P-h and T-s diagrams of three-stage compression heat pump system

From Figure 1(a) and 2(a), it can be seen that the pressure difference of each compression stage and expansion process become small for multi-stage compression heat pump. The irreversible loss can be greatly reduced to improve the exergy efficiency. Taking into account the pressure ratio and discharge temperature, the temperature difference of evaporating and condensing can be even larger based on the compression stage increasing. For the low grade industrial waste heat, it is a promising approach to develop high-temperature heat pumps with 120 ℃ pressurized hot water supply.

3. System Modelling and Exergy Analysis

The following assumptions are made for the development of simulation model:

(1) The system reaches a steady state, and pressure drop in pipes, gas cooler, condenser and evaporator are neglected; (2) The heat losses to the environment in the gas cooler, condenser, evaporator and compressor are neglected.

3.1 System modelling

Multi-stage compression heat pump systems with R1234ze(Z) working fluid are modelled using Engineering Equation Solver (EES) (Software, F-chart, 2012). Multi-stage compression (two-stage and three-stage) are compared with the single-stage compression cycle. The compressor model based on the suction refrigerant properties and compressor geometric parameter has been adopted. It can calculate the mass flow rate and the discharge specific enthalpy of refrigerant, and then the energy consumption of compressor can be determined. The mass flow rate of refrigerant through the compressor is given by:

$$\dot{m} = \rho_s n V_{dis} \eta_v \tag{1}$$

The power consumption of compressor can be calculated by:

$$\mathrm{W} = \frac{\dot{m}(h_{d,is} - h_s)}{\eta_{is}} \tag{2}$$

where ρ_s is the density for the suction-side refrigerant, n is the compressor speed, V_{dis} is the compressor displacement, $h_{d,is}$ is the discharge isentropic enthalpy, h_s is the suction enthalpy, and η_{is} is the isentropic efficiency ($\eta_{is}=0.8014-0.0484(P_d/P_s)$) [12].

The heating capacity is defined as:

$$\mathrm{Q} = \dot{m}(h_{con,in} - h_{con,out}) \tag{3}$$

The COP of multi-stage compression heat pump is defined as the ratio of heating capacity to the total power consumption. i.e.

$$COP = \frac{Q}{W} \tag{4}$$

3.2 Exergy analysis

In a vapour compression heat pump system, there are usually four major processes: evaporation, compression, condensation, and expansion. External energy (power) is supplied to the compressor and heat is added to the system in evaporator, whereas in condenser heat rejection is occurred from the system. Heat rejection and heat addition are dissimilar to different refrigerants, which cause a change in energy efficiency for the systems. Exergy losses in various components of the system are not the same. For the multi-stage compression heat pump, heat transfer process is also existed in the subcooler. Exergy is consumed or destroyed due to entropy created depending on the associated processes. To specify the exergy losses or destructions of the multi-stage compression heat pump, exergy analysis is necessary.

According to the second law of thermodynamics, a practical process is always irreversible. In a heat pump system, irreversible losses are caused by different factors, such as nonisentropic compression and temperature difference of heat transfer. According to the second law of thermodynamics, exergy analysis equations are showed as follows:

$$E_{heat,in} + E_{mass,in} + E_{work} = E_{heat,out} + E_{mass,out} + I_{rr} \tag{5}$$

$$\Psi = (h - h_0) - T_0(s - s_0) \tag{6}$$

where I_{rr} is irreversible loss, E_{heat} , E_{mass} and E_{work} are exergy during heat transfer, mass transfer and working process, Ψ is specific exergy in any state, T_0 is the surrounding temperature, h_0 and s_0 are enthalpy and entropy of working fluid under temperature of T_0 and pressure of 0.1 MPa. There is no mass transfer between hybrid source heat pump and the surrounding environment, hence $E_{mass} = 0$.

And theoretical exergy losses in different components are calculated according to the following formulas [13-15].

For the compression process:

Compressor work, $$W_c = m(h_{out} - h_{in}) \tag{7}$$

Power consumption, $$W_{el} = W_c / \eta_m \cdot \eta_e \tag{8}$$

So, exergy loss, $$I_{comp} = m(\psi_{in} - \psi_{out}) + W_{el} = m[(h_{in} - h_{out}) - T_0(s_{in} - s_{out})] + W_{el} \tag{9}$$

where I_{comp} is the theoretical exergy destruction of compressor; m is the mass flow rate of working fluid; s_{in} and s_{out} are specific entropy of working fluid at inlet and outlet of compressor. η_m is the mechanical efficiency of the compressor. η_e is the electrical efficiency of the motor. With reference to cited literatures [12], it is assumed that mechanical efficiency of the compressor is 95% and the electrical efficiency of the motor is 98%.

For condensation process:

$$I_{\mathrm{cond}} = m_{\mathrm{R1234ze}}(\psi_{\mathrm{R1234}ze,in} - \psi_{\mathrm{R1234}ze,out}) + m_{hw}(\psi_{hw,in} - \psi_{hw,out}) \\ = m_{R1234ze}[(h_{\mathrm{R1234}ze,in} - h_{\mathrm{R1234}ze,out}) - T_0(s_{\mathrm{R1234}ze,in} - s_{R1234ze,out})] + m_{hw}[(h_{hw,in} - h_{hw,out}) - T_0(s_{hw,in} - s_{hw,out})] \tag{10}$$

For evaporation process:

$$I_{evap} = m_{\mathrm{R1234}ze}(\psi_{\mathrm{R1234}ze,in} - \psi_{\mathrm{R1234}ze,out}) + m_{ww}(\psi_{ww,in} - \psi_{ww,out}) \\ = m_{R1234ze}[(h_{\mathrm{R1234}ze,in} - h_{\mathrm{R1234}ze,out}) - T_0(s_{\mathrm{R1234}ze,in} - s_{\mathrm{R1234}ze,out})] + m_{ww}[(h_{ww,in} - h_{ww,out}) - T_0(s_{ww,in} - s_{ww,out})] \tag{11}$$

For subcooling process:

$$I_{\mathrm{sub}} = m_{R\mathrm{1234ze}}(\psi_{\mathrm{R1234}ze,in} - \psi_{\mathrm{R1234}ze,out}) + m_{hw}(\psi_{hw,in} - \psi_{hw,out}) \\ = m_{R\mathrm{1234}ze}[(h_{\mathrm{R1234}ze,in} - h_{\mathrm{R1234}ze,out}) - T_0(s_{\mathrm{R1234}ze,in} - s_{R\mathrm{1234}ze,out})] + m_{hw}[(h_{hw,in} - h_{hw,out}) - T_0(s_{hw,in} - s_{hw,out})] \tag{12}$$

For expansion process:

$$I_{exp} = m_{R\mathrm{1234ze}}(\psi_{in} - \psi_{out}) = m_{R\mathrm{1234ze}}T_0(s_{out} - s_{in}) \;\text{[Throttling, } h_{exp,in} = h_{exp,out}\text{]} \tag{13}$$

Total exergy destruction:

$$I_{total} = I_{comp} + I_{cond} + I_{evap} + I_{\mathrm{exp}} + I_{sub} \tag{14}$$

Compared to conventional energy analysis, exergy analysis can quantitatively characterize the thermodynamic imperfection of heat transfer process and the possibility for thermodynamic development for heat pump system. Exergy efficiency is defined as the ratio of the total exergy increasement to the total power consumption of multi-stage compression heat pump.

$$\eta_x = \left(E_{hw,out} - E_{ww,in}\right) / W_{el} \tag{15}$$

Where the total exergy output Ehw,out and the exergy input Eww,in of the multi-stage compression heat pump can be given by:

$$E_{hw,out} = m_{hw}(\psi_{hw,out} - \psi_{hw,in}) = m_{hw}[(h_{hw,out} - h_{hw,in}) - T_0(s_{hw,out} - s_{hw,in})] \tag{16}$$

$$E_{ww,in} = m_{ww}(\psi_{ww,in} - \psi_{ww,out}) = m_{ww}[(h_{ww,in} - h_{ww,out}) - T_0(s_{ww,in} - s_{ww,out})] \tag{17}$$

4. Results and Discussion

The system performances of multi-stage compression heat pump systems are evaluated under the operating conditions for 120 ℃ pressurized hot water supply. The waste heat recovered in evaporator is supposed as a constant of 420 kW. The respective simulation results are discussed in the following subsections.

4.1 Variation of system performance

The variation of total power consumption with waste heat source temperature for different stage compression is showed in Figure 3. As the waste heat source temperature increased from 50 to 90 ℃, the total power consumption was found to decrease from 180 kW to 70.5 kW for single-stage compression system. For the two-stage and three-stage compression system, the total power consumption decrease from 170 kW to 68.4 kW and from 163 kW to 65.7 kW, respectively. This was mainly due to the decrease of compression ratio caused by the evaporating temperature rise. It also can be seen that three-stage compression system always has the smallest compressor power consumption. With the waste heat source temperature increasing, the power saving of multi-stage compression became less and less significant.

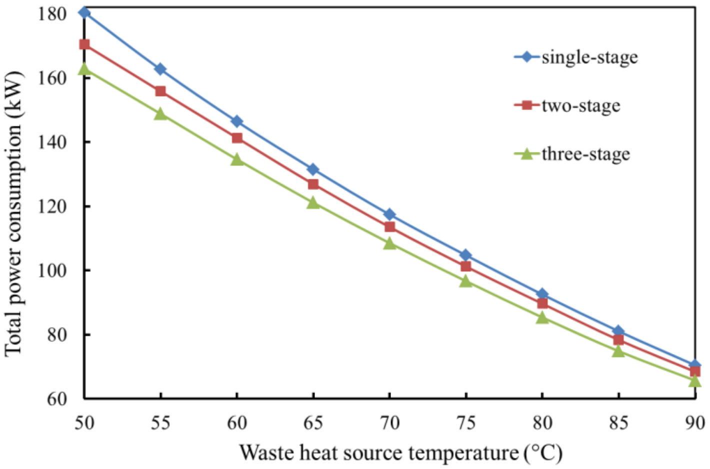


Fig. 3. Variation of total power consumption with waste heat source temperature

Based on the simulation results of total power consumption and heating capacity, the variation of system COP with waste heat source temperature is analyzed in Figure 4. It can be seen that system COP increased with the waste heat source temperature rise. As the waste heat source temperature increased from 50 to 90 ℃, the system COP increased from 3.1 to 6.7 for single-stage compression system and from 3.5 to 6.98 for two-stage compression system, respectively. The system COP of three-stage compression system was 3.74 for 50 ℃ waste heat source temperature and 7.14 for 90 ℃ waste heat source temperature. Under the same waste heat source temperature condition, the COP improvement of multi-stage compression system was obvious. Compared with single-stage compression system, the COP improvements of two-stage compression system are 12.2% and 6.3% for 50 ℃ and 80 ℃ waste heat source temperature, respectively. The COP improvements of three-stage compression system are 19.8% for 50 ℃ waste heat source temperature and 9.4% for 80 ℃ waste heat source temperature.

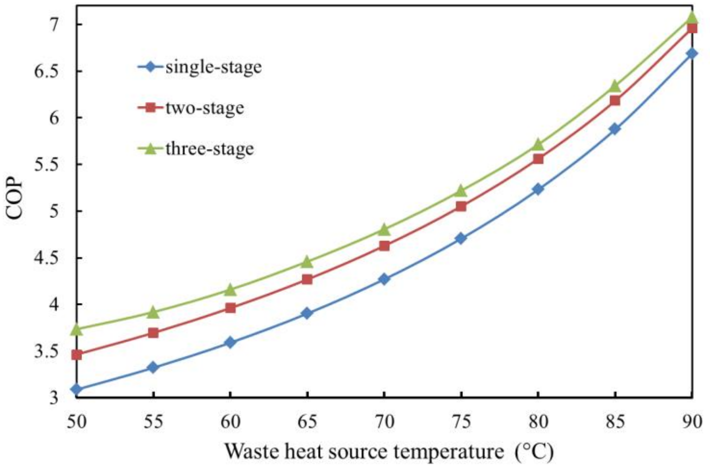


Fig. 4. Variation of COP with waste heat source temperature

Figure 5 reveals the variation of exergy efficiency with waste heat source temperature for different stage compression systems. As the waste heat source temperature increased, the exergy efficiency decreased gradually. For the single-stage and two-stage compression systems, the exergy efficiency decreased 18.3% and 19.2% when the waste heat source temperature increased from 50 to 90 ℃. For the three-stage compression system, the exergy efficiency decreased 20.2% when the waste heat source temperature increased from 50 to90 ℃. With the evaporating temperature increasing, the exergy destruction of compression process and expansion process became less and less serious. The total exergy destruction of the system is decreased while the exergy loss of condensation and evaporation process is a constant because of the fixed heat transfer approach temperature. Taking all the above factors into consideration, the relatively irreversible loss increased with the waste heat source temperature rise. So the exergy efficiency of all the single-stage and multi-stage compression systems decreased. For the waste heat source temperature of 60 ℃, the improvements of exergy efficiency were 6.9% and 11.8% for two-stage and three-stage compression systems when compared with single-stage compression system.

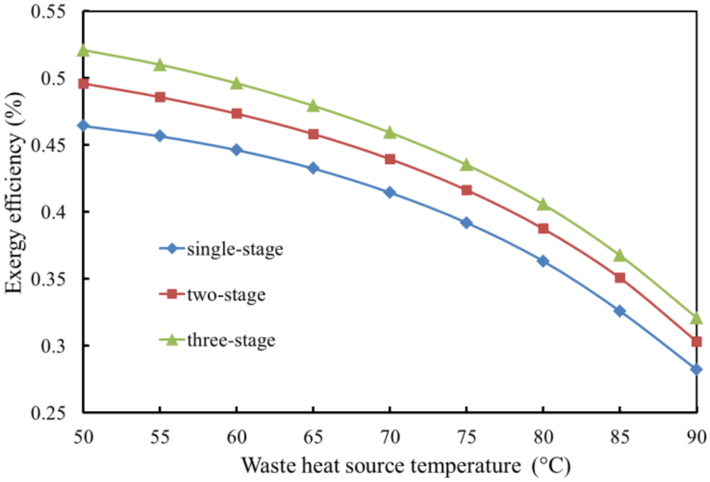


Fig. 5. Variation of exergy efficiency with waste heat source temperature

Table 1 shows simulation results of R1234ze(Z) heat pumps evaluated under 60 ℃ waste heat source temperature condition. As the stage number is increased, compressor work is reduced and heating capacity is increased. This results in an increase in COP. This table also shows the mass flow rate and pressure of each stage in the compression process. The reduced pressure ratio results in compressor work reduction, and finally results in COP improvement. When the three-stage compression heat pump is applied, the COP improvement is 16.4% under 60 ℃ waste heat source temperature conditions.

Table 1 Simulation results of R1234ze(Z) heat pumps.

Items	Unit	Single-stage	Two-stage	three-stage
W_{com}	kW	146.1	141.8	137.2
Q_{con}	kW	524.6	545.7	561.8
Q_{eva}	kW	420	420	420
COP	—	3.59	3.93	4.18
m_{total}	kg s^{-1}	2.96	3.11	3.25
$m_{in,1}$	kg s^{-1}		0.496	0.276
$m_{in,2}$	kg s^{-1}			0.322
P_{suc}	Bar	3.91	3.91	3.91
$P_{in,1}$	Bar		8.92	6.76
$P_{in,2}$	Bar			11.72
P_{dis}	Bar	20.29	20.29	20.29

4.2 Variation of exergy destruction ratio

The exergy destruction ratio reflects the percentage of exergy loss for every process in heat pump systems. Figure 6 shows the variation of exergy destruction ratio of compression process with waste heat source temperature. It can be seen that exergy destruction ratio of compression process decreased with the waste heat source temperature rise. As the waste heat source temperature increased from 50 to 90 ℃, the exergy destruction ratio of compression process was found to decrease from 44% to 36% for single-stage compression system. For the two-stage and three-stage compression system, the destruction ratio decreased from 42% to 31% and from 40% to 28%, respectively. This is mainly because the exergy destruction in the compressors decreased with the rise of evaporating pressure.

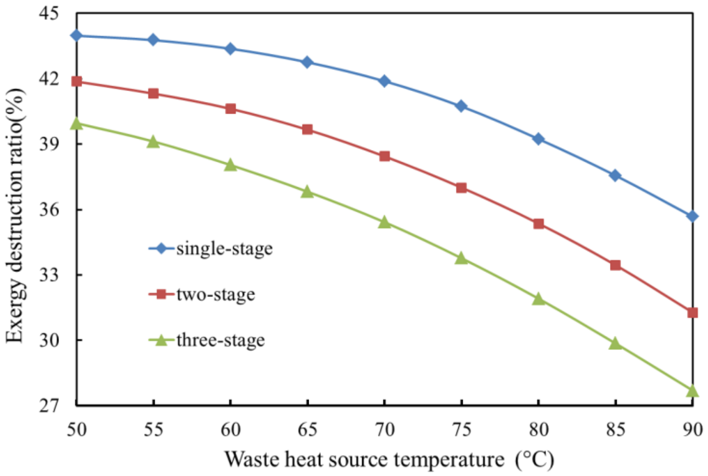


Fig. 6. Exergy destruction ratio of compression processes

Figure 7 indicates the variation of exergy destruction ratio of condensation processes with waste heat source temperature. As the waste heat source temperature increased from 50 to 90 ℃, the exergy destruction ratio of condensation processes increased from 29% to 43% for two-stage compression system and from 32% to 45% for three-stage compression system. For single-stage compression system, the exergy destruction ratio of condensation processes has the same increment (about 14%) with multi-stage compression system. This is mainly due to the decrease of the total exergy destruction and the constant exergy loss of condensation with waste heat source temperature increasing.

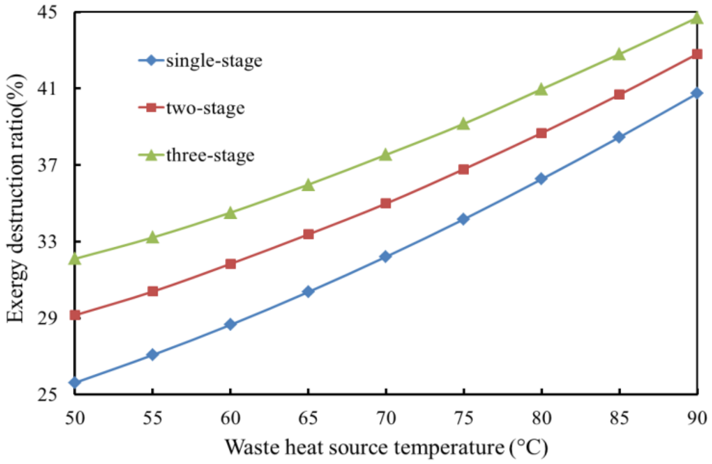


Fig. 7. Exergy destruction ratio of condensation processes

Figure 8 reveals the variation of exergy destruction ratio of expansion process with waste heat source temperature. As the waste heat source temperature increased from 50 to 90 ℃, the exergy destruction ratio of expansion process was found to decrease from 21.4% to 12.3% for single-stage compression system. For the two-stage and three-stage compression system, the destruction ratio decreased from 17.3% to 10.1% and from 14% to 9%, respectively. The reason is that the increase of waste heat source temperature can effectively reduce the pressure difference between evaporating and condensing, and then reduce the exergy loss of expansion process.

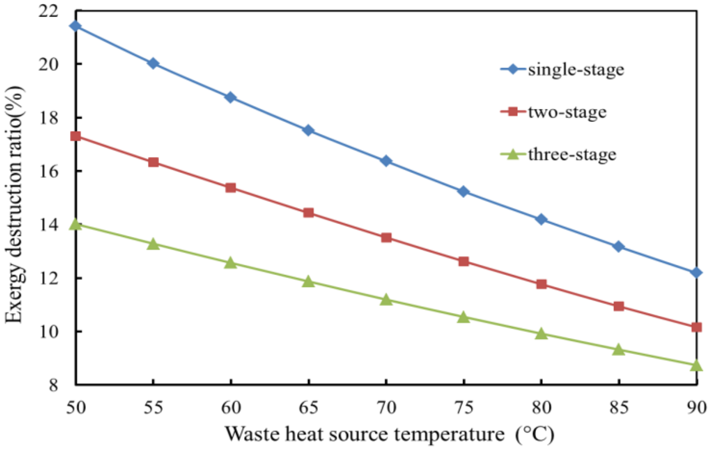


Fig. 8. Exergy destruction ratio of expansion processes

Figure 9 shows the variation of exergy destruction ratio of evaporation processes with waste heat source temperature. It can be seen that the exergy destruction ratio of evaporation processes increased 5.2% and 3.7% for two-stage and three-stage compression system, respectively. With the waste heat source temperature increasing from 50 to 90 ℃, the exergy destruction ratio of evaporation processes increased from 8.4% to15% for single-stage compression system. The mainly reason for evaporation processes was similar with that for condensation process. The decrease of the total exergy destruction may take an important part in the rise of the exergy destruction ratio for evaporation processes.

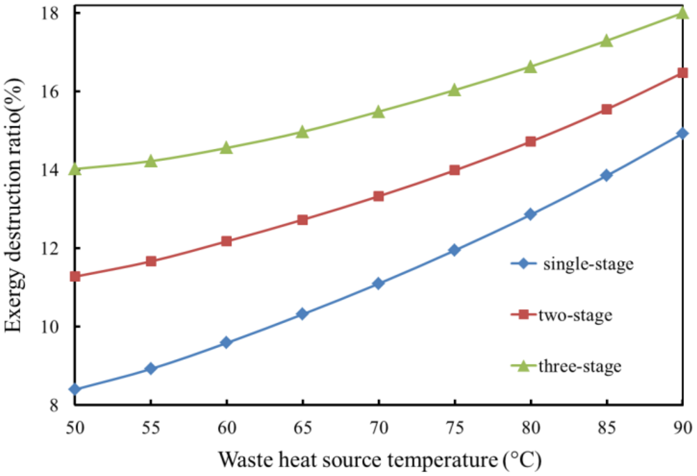


Fig. 9. Exergy destruction ratio of evaporation processes

5. Conclusions

Multi-stage compression heat pump systems with R1234ze(Z) refrigerant are investigated to recover the waste heat in industrial process. The pressured water is gradually heated by subcooler and condenser to a high-temperature of 120 °. Simulation approach is conducted to analyze the performance of multi-stage compression heat pump systems under different waste heat source temperature conditions. Besides, exergy analysis is also carried out to investigate the exergy destruction and exergy loss ratio. Main conclusions are as follows.

1) The waste heat source temperature has a great influence on total power consumption and system COP of the multi-stage compression heat pumps. As the waste heat source temperature increased from 50 to 90 ℃, system COP increases from 3.5 to 6.98 for two-stage compression system and from 3.74 to 7.14 for two-stage compression system, respectively. As the stage number increased for the same waste heat recovered, multi-stage compression heat pump has less power consumption.

2) Multi-stage compression heat pump improved the system COP and exergy efficiency obviously. When the three-stage compression heat pump is applied, the COP improvement is 16.4% under 60 ℃ waste heat source temperature conditions. The improvements of exergy efficiency are 6.9% and 11.8% for two-stage and three-stage compression systems when compared with single-stage compression system.

3) The exergy destruction ratio of each process is investigated for multi-stage compression heat pumps. With the waste heat source temperature increasing, the exergy destruction ratio of compression and expansion processes decreased, while that of condensation and evaporation processes increased. For the same operating conditions, three-stage compression heat pump has the minimum exergy destruction of compression and expansion process. The results indicate that the three-stage compression heat pump system has obvious advantage of exergy efficiency with heat recovery.

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