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Research Article

Research on High Temperature Cascade Heat Pump System for Vapor Production Employing Scroll Compressors and Multi-stage Preheat Cycle

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Abstract

Water vapor production is an essential process in industrial fields such as dyeing, food processing, and pharmaceutical manufacturing, yet it is characterized by substantial power consumption. Against the backdrop of global carbon neutralization initiatives, high-temperature heat pumps have emerged as a promising alternative to traditional boilers, owing to their environmental friendliness and cost-saving advantages. However, developing and manufacturing such vapor production systems poses significant technical challenges: compressors must withstand extreme high pressures, heat exchangers suffer from low effectiveness during high-temperature condensation, suitable working fluids balancing low evaporation and high condensation temperatures are scarce, and components face issues like heat exchanger incrustation and insufficient compressor lubrication at high discharge temperatures. In this study, a cascade high-temperature heat pump system integrated with a flashing circulation and multi-stage preheating cycle was developed for water vapor production. The system comprises three core cycles: an R134a low-temperature heat pump cycle, an R245fa high-temperature heat pump cycle, and a two-stage preheat-evaporation water cycle. Theoretical simulations were conducted to evaluate component performance (compressors, heat exchangers, etc.) and optimize system parameters. An experimental testbed was constructed with scroll compressors (3.5kW for R134a, 5kW for R245fa) and plate heat exchangers, verifying system reliability under varied operating conditions. Results demonstrate the system efficiently produces 135°C/0.3MPa water vapor, with a Coefficient of Performance (COP) ranging from 2.1 to 3.5. COP increases with evaporation temperature and decreases with condensation temperature, with an optimal intermediate-stage temperature of 70–75°C. This system proves to be an efficient, eco-friendly industrial alternative to conventional boilers.

Keywords

High-temperature Heat Pump, Steam Vapor Production, Cascade System, Simulation Analysis, Testbed Development

1. Introduction

Heating is widely required in industries such as dyeing, food processing, and pharmaceuticals, where high-temperature water vapor serves as the primary heating medium. Traditionally, steam supply relied on boilers, which are

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environmentally unfriendly and have therefore been banned or restricted in many urban areas. Electric boilers, though cleaner, are inefficient and have high operating costs. Recently, high-temperature heat pumps have emerged as a promising substitute for boilers in industrial steam production due to their superior heating performance, long-term reliability, and strong potential for waste-heat recovery [1, 2].

Heat pumps were originally used in air conditioning and ventilation where the heating temperature was around 45-65°C. However, in water vapor producing process, the heating temperature was required to be more than 120°C [3]. Thus, a series of problems appear in the development and operation of such high temperature heat pump. First, the key component should be able to operate effectively and efficiently in such high temperature, while the heat exchanger might have incrustation risks and the compressor could be insufficiently lubricated as such high discharge temperature [4, 7]. Secondly, there are no such working fluid which could satisfy the low evaporation temperature and high condensation temperature at reasonable operating pressure [8, 9]. Thirdly, the heat conduction effectiveness of water-refrigerant heat exchanger might not be able to fully evaporate such high temperature vapor so that a circulation cycle with flash tank must be made in the condenser [10-12].

In this study, a cascade high-temperature heat pump system with flash circulation and a multi-stage preheating cycle was developed for producing 135°C/0.3 MPa water vapor. Theoretical simulations were first conducted to predict system performance and aid in the selection of components such as compressors, heat exchangers, pumps, and other auxiliary equipment. An experimental testbed was subsequently constructed to investigate and validate system reliability. Data analysis and optimization were performed to enhance steam production performance.

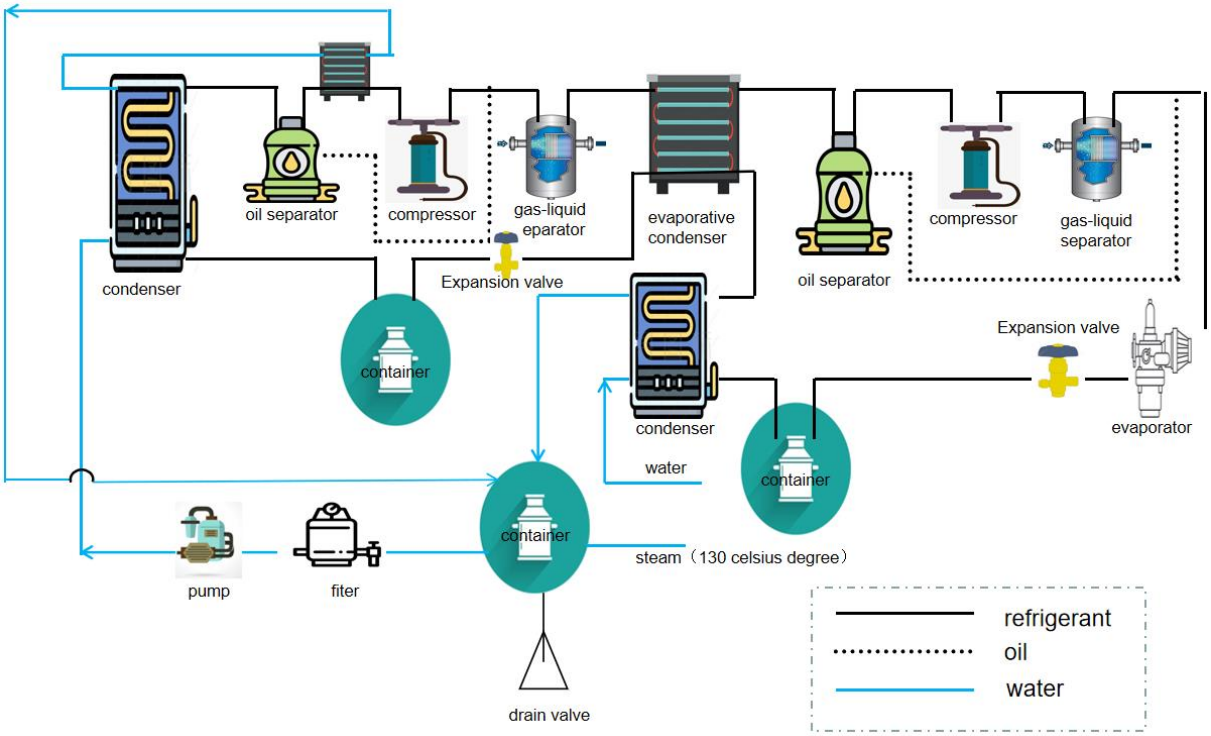

Figure 1. *Configuration of cascade heat pump system.*

2. System Description and Simulation Models

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2.1. System Description

As shown in Figure 1, a cascade heat pump system is built to satisfy the requirement of 135°C water vapor producing. This system was made up mainly of three cycles with different working fluid, which are the R134a low temperature heat pump cycle, the R245fa high temperature heat pump cycle and the two stage preheat-evaporation water cycle.

The low-temperature R134a heat pump cycle includes a

compressor, an evaporator, an evaporative condenser, an expansion valve, and auxiliary components such as an oil separator, a gas-liquid separator, and a reservoir. R134a is used as the working fluid in this cycle. It evaporates in the evaporator by absorbing heat from the ambient environment. The working fluid then flows into the gas-liquid separator, where the refrigerant vapor is directed to the compressor. After compression, the vapor is separated from the oil and condenses in the evaporative condenser, transferring heat to the R245fa liquid. The heat from the R134a fluid is further recovered in a preheater to raise the temperature of the incoming water. Finally, the liquid R134a returns to the reservoir before beginning the next cycle.

The high temperature R245fa heat pump cycle also consists primarily of a compressor, an evaporative condenser, a condenser, and an expansion valve, along with similar supporting components like oil separators, gas-liquid separators, and reservoirs. R245fa serves as the working fluid in this cycle. After being heated by the condensed R134a in the evaporative condenser, it enters the evaporator where it absorbs heat from the R134a liquid. The R245fa gas then passes through gas-liquid separator before reaching the compressor and the oil separator where the pressure is further increased, and any remaining oil is separated prior to condensation in the condenser. The condensed R245fa releases heat to the water stream, raising its temperature to saturation point. Subsequently, the cooled R245fa liquid returns to the reservoir for another round of circulation.

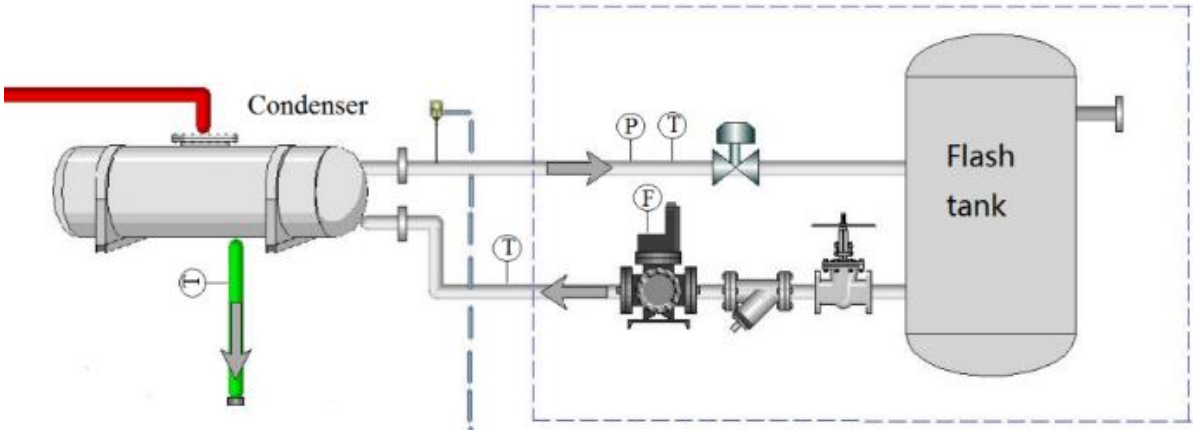

Figure 2. *Configuration of the flash tank system.*

Finally, in the two-stage preheat-evaporation water cycle, water from the main system is initially preheated using the latent heat from both the R134a and R245fa cycles. This preheating step raises the water's temperature before it enters the primary evaporation stage. In this stage, the water is brought to its saturation point, evaporating into steam at 130 degrees Celsius. This steam is then used to serving various applications or processes as efficient supply of high temperature heat.

In the flash tank system configuration, a heating circulation is maintained at an appropriate cycling ratio. Water from the preheating cycle is pumped to a pressure of 0.2–0.5 MPa, depending on the steam supply requirements. The high-pressure water exchanges heat with R245fa in the condenser and with the superheated discharge from the high-temperature compressor (after the oil separator), raising its temperature to 135°C. This high-pressure water is then pressure-reduced in the flash tank according to vapor supply needs, with the flow rate controlled by an electromagnetic valve at the flash tank inlet based on the outlet temperature

2.2. Mathematical Model

A series of mathematical models was developed to evaluate the performance of the steam production system. Power consumption is primarily attributed to the compressors, while heating is supplied by the condenser, superheater, and preheater. Models for compressor efficiency, heat exchangers, and expansion valves are essential for these calculations. The compressor model predicts isentropic efficiency, while the heat exchanger model estimates heat transfer capacity and the logarithmic mean temperature difference (LMTD) between fluids. The optimal condensing temperature is determined through simulation. The following assumptions were made to simplify the thermodynamic analysis [13-15].

1) All throttling processes are viewed as isenthalpic process.
2) The pressure losses within heat exchangers are neglected.
3) The power consumption of pumps is neglected

In the heat pump system, each component could be viewed as a control volume with working fluid flowing in or out, as well as heat and power consumed. The basic equations describing mass balance and energy balance are given as:

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{1}$$

$$\dot{Q} - \dot{W} + \sum_{in} \dot{m}h - \sum_{out} \dot{m}h = 0 \tag{2}$$

The power consumption of compressors depends on the theoretical power consumption according to the operating condition and the isentropic efficiency of the compressor.

$$W_{comp} = \frac{W_{th}}{\eta_a} \tag{3}$$

The suction volumetric flow rate of working fluid depends on the discharged mass flow rate, the volumetric efficiency and thermodynamic properties of working fluid at the suction side.

$$\dot{V}_{suc} = \frac{\dot{m}_{dis}}{\rho_{suc}\eta_v} \tag{4}$$

In the simulation of heat exchanger, the calculation is made adopting the Effectiveness-NTU (Number of Transfer Unit) method, in which the effectiveness of heat exchanger and the number of transfer units are calculated and correlated at the given type of heat exchanger, whether it is for evaporator, condenser preheater or over-heater [16].

With the definition of ε, the actual heat exchanging rate could be calculated by

$$q = \varepsilon C_{\min}(T_{h,i} - T_{c,i}) \tag{5}$$

On the other hand, the number of transfer units (NTU) refers to a dimensionless parameter defined as

$$\mathrm{NTU} = \frac{UA}{C_{\min}} \tag{6}$$

where U refers to the heat exchange coefficient, A refers to the heat exchange area. plate heat exchanger is calculated by [17, 18]

$$\varepsilon = 1 - exp[(\frac{1}{C_r})(NTU)^{0.22}\{exp[-C_r(NTU)^{0.78}] - 1\}] \tag{7}$$

With the acknowledgement of the inlet thermodynamic properties, heat exchanger type and geometry parameters and mass flow rate of each fluid, the outlet operating condition could be calculated. Furthermore, the thermodynamic properties of refrigerant and water are determined with Nist Refprop 9.1 while Matlab R2024a is employed as the solver.

The calculation of compressor shaft power directly determines the power consumption and COP. It is found that for scroll compressor, the modeling for shaft power is more accurate compared with merely calculating isentropic efficiency because the heat transfer, mass inter-leakage and mechanical losses are affecting it in different mechanism. Calculation of compressor shaft power as [5, 6]

$$W = P_{suc}V_{suc}a_1\left[\left(\frac{P_{dis}}{P_{suc}}\right)^{a_2+\frac{k-1}{k}} + \frac{a_3}{P_{dis}}\right] + W_{loss} \tag{8}$$

where the a1, a2, a3 and Wloss should be calculated by fitting experimental data. The isentropic efficiency could also be calculated based on the practical shaft power consumption of compressor.

In this investigation, the compressor efficiency is also calculated through experimentally measuring the power consumption and actual flow rate at various operating conditions. Therefore, the validated parameters is fit and then employed in the simulation of the system.

The performance of the steam production systems includes the COP (Coefficient of Performance) and the capacity. The former is illustrated by the power consumption at a given amount of steam produced, while the latter is illustrated by the massive flow rate of steam at the inlet of users.

$$COP = \frac{Q_{cond} + Q_{preheat} + Q_{overheat}}{\sum W_{comp}} \tag{9}$$

Figure 3. *Experimental facility.*

3. Experimental Investigation

An experimental testbed was constructed based on the system configuration described above. Two scroll compressors were employed: a 3.5 kW compressor for the R134a low-temperature cycle (max pressure: 3 MPa) and a 5 kW compressor for the R245fa high-temperature cycle (max pres-

sure: 3 MPa). Four plate heat exchangers and one tube-in-tube heat exchanger, with capacities ranging from 4 kW to 13 kW, were used. Thermostatic expansion valves were installed in both heat pump stages. Temperatures were measured using Pt-100 thermocouples, and pressure meters were installed at the inlets and outlets of the compressors and the flash tank. The evaporation temperature was varied from 5°C to 30°C, and the condensation temperature was set between 110°C and 135°C.

4. Result Analysis

4.1. Experimental Test Result

The experimental tests demonstrated the system's effective production of high-temperature water vapor under various operating conditions. Initially, the evaporator did not operate at maximum capacity due to an imbalance in the low-stage cycle, where the R245fa heating was insufficient to cool the R134a working fluid in the gas-to-gas heat exchanger. Consequently, the water recycle temperature was initially around 100°C.

As the high-stage cycle stabilized, the discharge pressure of R134a reached 2.6 MPa (saturation temperature ~80°C). Subsequently, water injection helped control the temperature and prevent over-pressurization. The water from the superheater reached 135°C, producing discharged vapor at 120–125°C, as shown in Figure 4.

It should be illustrated that the flash tank in this system was somewhat too large for such low capacity compressors and heat exchangers, so that it was time consuming for the operation of the system to supply vapor for the first time. It is recommended that a less amount of water restored in the flash tank, which both increase the efficiency of low staeg system at dynamic parameters and shorten the time elapse for the steam supply.

Figure 4. *Test result.*

4.2. Influence of Operating Temperature on COP

The Coefficient of Performance (COP) generally increases with an increase in evaporator temperature (tcooling). This system was experimentally proved to be effective at supplying water vapor continuously at a minimum ambient temperaturea at 10°C, so that a minimum 5°C evaporating temperature was settled. At this temperature, no significant frost is observed at the surface of evaporator, while a full load of evaporator fan is set to fulfill the requirement for heat exchange at such low temperature of evaporation. On the other hand, the maximum operating temperature of this system was set at a maximum room temperature of 40°C, where high running temperature could also be made at such evaporating temperature.

At a condenser temperature (T_cond) of 130°C, the COP increased from approximately 2.17 at an evaporator temperature of 5°C to about 2.77 at 35°C. The increase in COP was more significant at lower evaporator temperatures and less pronounced at higher temperatures. This improvement is primarily attributed to the reduction in compression power at higher suction pressures, especially in the low-temperature cycle. Furthermore, higher evaporation temperatures increase the mass flow rate, enhancing both heat exchange efficiency and the overall heating capacity of the system.

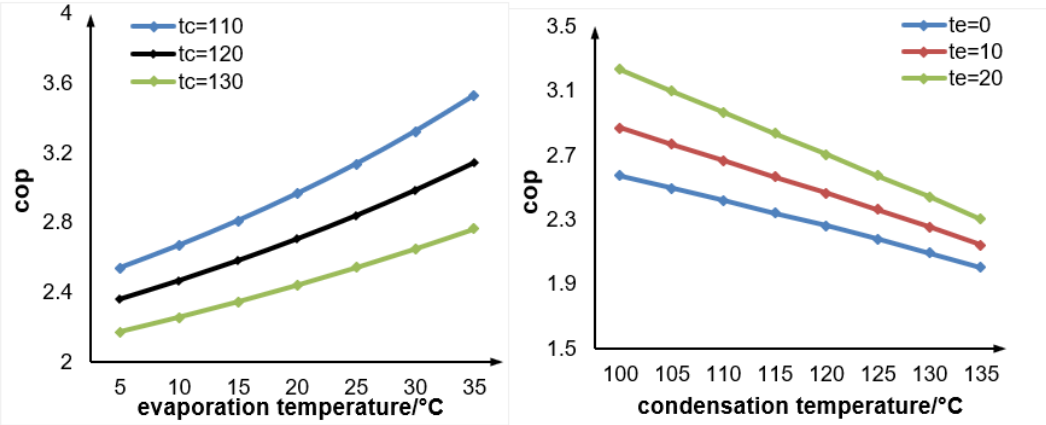

Figure 5. Influence of operating temperature on performance of heat pump system.

4.3. Optim al Inter-stage Temperature

At the evaporator temperature range considered and a condenser temperature of 130°C, the COP initially increases and then decreases, indicating the presence of an optimal intermediate stage temperature (tmid). This optimal behavior is observed specifically when the intermediate temperature is set at 72°C.

The choice of 72°C as the optimal inter-stage temperature carries critical system control implications: it necessitates a suitable operating condition between the R134a low-temperature cycle and R245fa high-temperature cycle. Specifically, real-time monitoring of tmid via Pt-100 thermocouples (as used in the testbed) should trigger dynamic adjustments of compressor frequencies or thermo-expansion valve openings, stabilizing tmid near 72°C to maintain the system at peak COP, even amid fluctuations in ambient temperature or steam demand.

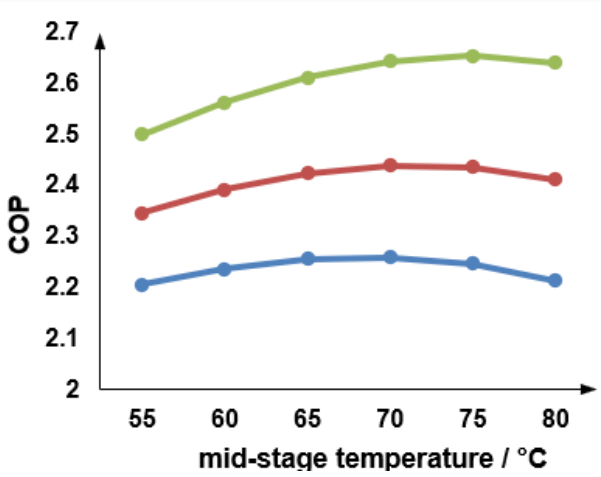

Figure 6. Influence of mid-stage temperature on performance of heat pump system.

For an evaporator temperature starting at 15°C, the COP increases up to an evaporator temperature of 20°C, where it reaches its maximum value of 2.4414. Beyond this point, the COP values are not provided for higher evaporator temperatures, but based on the trend, we can infer that the COP would start to decrease after reaching the maximum at an evaporator temperature of 20°C.

Therefore, the maximum COP is 2.4414, which occurs at an evaporator temperature of 20°C when the intermediate temperature (tmid) is 72°C and the condenser temperature (ta) is 130°C.

5. Conclusion

In this paper, a novel high temperature cascade heat pump system is developed for 130°C vapor production. Both theoretical and experimental analysis is made on the performance of the system. This configuration was proved to be effective and efficient in vapor steam production as substitution of boiler. Its COP increase with increase in evaporation temperature and decrease in steam temperature, ranging from 2.1 to 3.5. The best fit mid-stage temperature lies at around 70-75°C according to operating conditions. *Practically, compared with traditional boilers, the system cuts annual energy consumption by ~45% and reduces CO_2 emissions by over 60% per ton of steam; under typical industrial energy pricing, the investment payback period is estimated at 1.5-2.5 years, strongly validating its economic and environmental feasibility for industrial adoption.*

Abbreviations

NTU	Number of Transfer Unit
COP	Coefficient of Performance

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Conflicts of Interest

The authors declare no conflicts of interest.

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Research Field

Haihan Yu: Applied Physics
Zhaorui Zhao: Compressor and Refrigeration