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Analysis of a modified single-effect heat pump desalination system

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ABSTRACT

Heat pumps offer an effective heat recovery way for small-capacity thermal desalination plants. A modified single-effect heat pump desalination system is proposed in this paper. The auxiliary condenser for the condensing of high temperature refrigerant in a conventional system is adjusted to condense the low temperature steam. Aspen Plus software is employed to develop simulation models, which are subsequently validated. With the simulation results, performances of the modified system and the conventional system are compared. It is concluded that modified system produces more freshwater for a constant heat pump operation condition and seawater feed condition. Furthermore, the circulating refrigerant mass together with the compression power can be reduced by the modification when the freshwater production is fixed, resulting in a growth rate of 7% in both freshwater production and performance ratio (PR). However, the modification sacrifices the heat transfer area of auxiliary condenser. For both systems, an increase in heat pump evaporating temperature leads to a decrease in freshwater production but an increase in the PR. Conversely, elevating the heat pump condensing temperature results in a reduction of both freshwater production and PR for both systems. Findings offer reference for the further optimal design of small-scale heat pump desalination systems.

Keywords: Desalination; Single-effect; Heat pump; Auxiliary condenser; Aspen plus

1. Introduction

The paucity of freshwater resources is emerging as the primary impediment to the sustainable advancement of the global economy. The utilization of desalination has demonstrated to be an exceptionally dependable alternative to fulfill both the civil and industrial freshwater demands [1–3]. Desalination plants have gained significant traction in regions facing freshwater scarcity and in coastal countries. According to statistics, more than 18,000 desalination plants worldwide produced an estimated 38 billion cubic meters of freshwater in 2016, with an anticipated increase to 54 billion cubic meters by 2030 [4].

Distillation is a well-established and effective desalination methodology, that employs the basic principle of heating seawater to produce steam and then condensing it to obtain freshwater [5]. Multi-effect distillation (MED) is one of the most widely used desalination technologies that relies on this method. However, the primary challenge associated with this technique is its substantial energy consumption due to the significant latent heat of water evaporation. To address this issue, the recovery of condensing heat from low-temperature steam is considered an efficient approach for enhancing the energy efficiency of these systems. Mechanical vapor recompression or heat pump technologies can be employed to achieve this objective. The former technique compresses low-temperature steam directly using a mechanical steam compressor, which then enables the reuse of high-temperature compressed steam as a heat source for seawater evaporation. The stringent technical

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requirements for the oil-free steam compressor must be ensured to guarantee the quality of condensed freshwater, while the discharge temperature limits the pressure ratio of the compressor [6]. Moreover, steam compressors are generally priced exorbitantly.

The compression heat pump is a mature and energy-saving heat recovery technology, employs the same principle as the compression refrigeration cycle. Specifically, it absorbs the condensation heat of low temperature steam with the refrigerant liquid in the evaporator and then compresses the resulting refrigerant gas to higher pressure, which is directed to the condenser. Through this process, the condensation heat of the refrigerant gas is reclaimed for seawater evaporation, thereby recovering the low temperature steam condensing heat. The application of compression heat pump for seawater desalination can date back to 1983 [7]. Slesarenko [8] conducted a thermal analysis of a heat pump desalination system, indicating that the heat pump was well suited for small-capacity thermal desalination plants. The utilization of heat pump can increase the system's performance index by 2–3 times.

Up to now, numerous investigations have been conducted on heat pump desalination systems. Siqueiros et al. [9] performed the energy and availability analysis for heat pump desalination systems. The results showed that no matter the compression heat pump or the absorption heat pump could improve the performance of desalination plants. The compression heat pump is driven by electricity and the absorption heat pump is driven by heat. The selection of heat pump technology should consider the price and form of energy resources. Yang at el. [10] carried out an experimental study on a wastewater desalination system with a low-temperature heat pump. R134a was selected as the refrigerant. According to the results, the performance ratio (PR) of the system reached 7.2 with a freshwater production rate of 3 kg/h. Lin et al. [11] designed a double-stage forced-circulation heat pump desalination system and performed an experimental study with R22 as the heat pump refrigerant. It was concluded that the system consumed more energy than reverse osmosis and mechanical vapor recompression desalination system. But the heat pump desalination plant could be cheaper and more compact.

A part of scholars also combined the heat pump with the membrane distillation (MD) unit or humidification–dehumidification (HDH) desalination unit. Tan et al. [12] coupled the heat pump with a sweep-gas MD system and built a corresponding experimental setup for the experimental study. Results showed that the freshwater production per unit of energy consumption was doubled. Zhang et al. [13] proposed an HDH heat pump desalination system and conducted an experimental study on its performance. The maximum freshwater production of the proposed system was 22.26 kg/h which was higher than that of any other HDH system. Based on the first law of thermodynamics, Lawal et al. [14] performed a theoretical investigation of the HDH desalination system coupled with a compression heat pump. The results indicated the gain output ratio (GOR) of the system was as high as 8.88 with an optimum mass flow rate ratio of 0.63.

The heat transfer medium employed in a heat pump system, namely the refrigerant, significantly impacts the performance of heat pump desalination systems. As such, the characteristics and properties of the refrigerant assume paramount importance. In addition to traditional refrigerants, the research and application of natural refrigerants have been paid more and more attention. Farsi et al. [15] combined the transcritical carbon dioxide refrigeration with Boosted-MED system. According to the results of energy and economic analysis, it was concluded that the specific heat transfer area would be reduced by 135% compared to conventional MEDs. Attia [16] evaluated the performance of passive vacuum generation heat pump seawater distillation system with different refrigerants including R22, R12, R134a and R718 (pure water). Using water as refrigerant was recommended once the technical problems of steam compressor were overcome.

It is widely recognized that heat pumps typically exhibit a higher release of heat from the condenser than absorption of heat by the evaporator. In certain heat pump-based applications, such as desalination or drying, the attainment of system stability necessitates the incorporation of an auxiliary condenser, as mandated by the principles of energy conservation. At present, the auxiliary condensers in the reported systems were added at the outlet of the refrigerant compressor [17,18] or downstream of the main condenser [19], resulting in a loss of high-grade thermal energy released by the high-pressure refrigerant. In addition to reducing the refrigerant condensing heat, the auxiliary condenser can also facilitate heat absorption of the heat pump evaporator, thereby dissipating part of the low temperature steam condensing heat. However, there is a lack of in-depth research on the impact of the location of the auxiliary condenser.

The present study introduces a modified single-effect heat pump desalination system, achieved through relocating the auxiliary condenser from the refrigerant pipeline to the steam pipeline. The modification can optimize the energy balance scheme of the single-effect heat pump desalination system to reduce the waste of high-grade thermal energy. Models for the system performance simulation are established with the Aspen Plus software, which is a widely adopted tool for theoretical investigations in desalination systems research [20–22]. The test data of a heat pump desalination product is used to verify the reliability of the simulation method. Moreover, a comparative performance analysis of the two systems is conducted to investigate the advantageous impact of the modification in various operating conditions. The study aims to establish the feasibility of the modified system, thus contributing to the development of high-efficiency heat pump desalination products and providing a valuable reference for future research.

2. System description

Fig. 1 is the schematic diagram of the conventional single-effect heat pump desalination system, wherein the air-cooled auxiliary condenser is positioned at the outlet of the refrigeration compressor, and the heat exchange coils of the heat pump condenser and evaporator are respectively installed in the seawater evaporator and steam condenser. The working process of the system is described as follows:

The feed seawater is introduced to the preheater through a feed pump and then is mixed with recycled brine. The

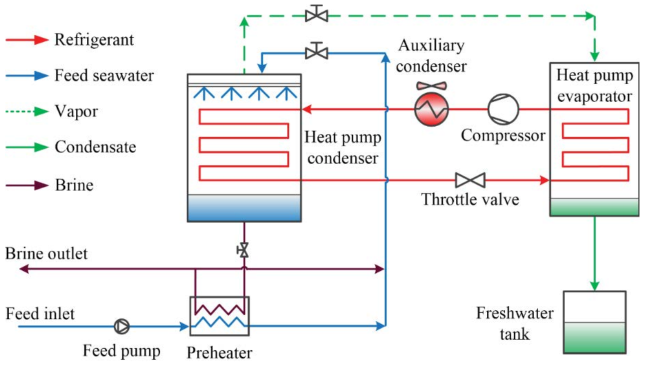

Fig. 1. Schematic diagram of the conventional single-effect heat pump desalination system.

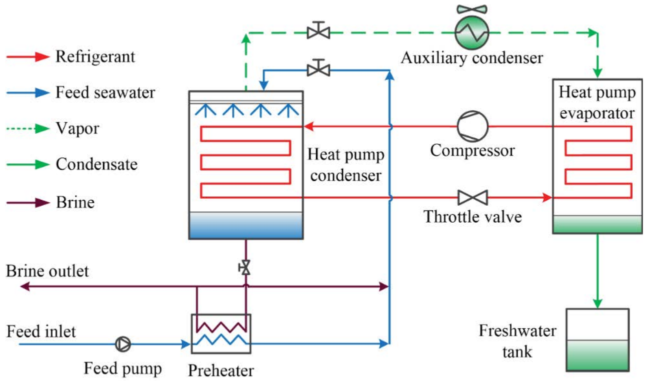

Fig. 2. Schematic diagram of the modified single-effect heat pump desalination system.

resulting mixture is subsequently sprayed into the seawater evaporator and subjected to heating by the high-pressure refrigerant present within the condensing coil, thereby inducing evaporation of seawater and consequent production of low-temperature steam. The resultant brine exhibits increased concentration due to the evaporative process. The low-temperature steam is channelled into the heat pump evaporator, where it undergoes condensation into freshwater. All the brine from the seawater evaporator flows into the preheater, aiding in the preheating of the feed seawater. Subsequently, partial cold brine is mixed with the feed seawater and reintroduced into the seawater evaporator. The remaining brine is discharged.

For the heat pump cycle, the compressed refrigerant gas first flows into the auxiliary condenser to dissipate partial high-grade heat. Following this, the refrigerant is introduced into a condensing coil and condenses into the refrigerant liquid with a certain degree of subcooling. The released heat is utilized for seawater evaporation. The resulting liquid refrigerant is subsequently passed through a throttle valve and introduced into an evaporator coil, where low-pressure liquid refrigerant evaporates by absorbing the steam condensing heat. The evaporated refrigerant gas is then compressed into high-pressure refrigerant gas, ready for the next cycle.

In the pursuit of enhanced energy efficiency and reduction of high-grade heat waste, modifications have been made to the conventional single-effect heat pump desalination system by relocating the auxiliary condenser, as shown in Fig. 2. The basic working principle of the modified system is the similar with that of conventional system shown in Fig. 1. The difference is that the compressed refrigerant gas now directly flows into the seawater evaporator, thereby enabling the utilization of all high-grade heat released by refrigerant for seawater evaporation. In light of the variance between the evaporator and condenser of the heat pump regarding heat transfer capacity, the location of the auxiliary condenser has been adjusted to address the excessive low-temperature steam condensing heat and ensure the stable performance of the desalination system. Notably, retention of an air-cooled heat exchanger as the auxiliary condenser will significantly reduce the heat transfer temperature difference and necessitate that the steam temperature be at least 5°C~10°C higher than ambient temperature, which represents the sole drawback of the modified system. This limitation, however, can be overcome by substituting the air-cooled heat exchanger with a water-cooled plate heat exchanger.

The control strategy of the auxiliary condenser is adjusted based on its location to maintain a stable evaporating/condensing temperature of the heat pump under varying seawater feed conditions. In the conventional system, the heat dissipated by the auxiliary condenser is regulated according to the degree of refrigerant subcooling, whereas in the modified system, the regulation is based on the superheat of refrigerant gas at the evaporator outlet. For different operation conditions, the refrigerant circulation amount is adjusted to maintain the desired refrigerant gas superheat for the conventional system and the refrigerant liquid subcooling for the modified system. Because of not any waste of refrigerant condensing heat, the modification offers the advantage of reducing the refrigerant circulation amount and compression power for a fixed desalination target or increasing the freshwater production for a specific heat pump operation condition.

3. Methods

The performance evaluation of two single-effect heat pump desalination systems is carried out through theoretical research in this paper. The present section presents a comprehensive account of the employed research methodologies, which includes assumptions, simulation models, parameters settings, and performance definitions.

3.1. Basic assumptions

The following basic assumptions are made for the process simulation [23]:

- The system operates under stable conditions;
- Ignore the heat loss and fluid pressure drop of each component;
- The condensate produced is pure water;
- Ignore the influence of non-condensable gas.

3.2. Models

The introduction of models for computing physical properties and simulating system performance, which are essential for theoretical studies, is presented below, alongside the validation of these models.

3.2.1. Boiling point elevation of seawater

The boiling point elevation (BPE) phenomenon is a well-documented property of salt solutions. In the present article, seawater is examined as a prototypical example of a sodium chloride solution. The study's main objective is to elucidate the relationship between salinity and BPE, which is expressed by the following equations [24]:

$$\mathrm{BPE} = \mathrm{A}X + \mathrm{B}X^2 + \mathrm{C}X^3 \tag{1}$$

$$A = 8.325\times10^{-2} + 1.883\times10^{-4}T + 4.02\times10^{-6}T^2 \tag{2}$$

$$B = -7.625\times10^{-4} + 9.02\times10^{-5}T - 5.2\times10^{-7}T^2 \tag{3}$$

$$C = 1.522\times10^{-4} - 3\times10^{-6}T - 3\times10^{-8}T^2 \tag{4}$$

where X is the salinity, ‰; T is the temperature of seawater, °C.

The BPE value of seawater with a salinity of 35‰ and temperature of 40°C is 0.35°C. The higher the seawater salinity and temperature, the higher the BPE value. As a result, the condensing temperature or the pressure of compressed refrigerant has to increase to ensure the proper heat transfer temperature difference.

3.2.2. Heat transfer coefficient

The heat transfer coefficient of the heat pump condenser and the heat pump evaporator is calculated by Eqs. (5) and (6) respectively according to the literature [25]:

$$U_e = 1.9394 + 1.40562\times10^{-3}T_b - 2.0752\times10^{-4}T_b^2 + 2.3186\times10^{-6}T_b^3 \tag{5}$$

$$U_c = 1.6175 + 0.1537\times10^{-3}T_v + 0.1825\times10^{-3}T_v^2 - 8.026\times10^{-8}T_v^3 \tag{6}$$

where U_e is the heat transfer coefficient of the evaporator, kW/(m²·°C); U_c is the heat transfer coefficient of the condenser, kW/(m²·°C); T_b is the boiling temperature of seawater, °C; T_v is the condensing temperature of steam, °C.

3.2.3. Heat transfer area

The heat transfer area of the auxiliary condenser is calculated by Eq. (7):

$$A = \frac{Q_a}{U_a \Delta T} \tag{7}$$

where A is the heat transfer area, m²; Q_a is the heat dissipating capacity of the auxiliary condenser, kW; U_a is the total heat transfer coefficient of the auxiliary condenser, kW/(m²·°C); ΔT is the heat transfer temperature difference of the auxiliary, °C.

3.2.4. Simulation model

Simulation models for two single-effect heat pump desalination systems are established with the Aspen Plus Software. In particular, a modified desalination system is employed to exemplify the simulation model as depicted in Fig. 3. The construction of the simulation model involves the implementation of the HeatX, Valve2, Aircooler, FSsplit, and Mixer modules. The simulation of the system is carried out utilizing the ELECNRTL physical property method, which is appropriate for the simulation of inorganic electrolytes such as the simplified NaCl solution used in this study.

3.3. Performance definitions

The coefficient of performance (COP) is used as the performance parameter of the heat pump. Its definition is as follows:

$$\mathrm{COP} = \frac{Q_{\mathrm{hot}}}{W_{\mathrm{comp}}} = \frac{m_r\left(h_{c,\mathrm{in}} - h_{c,\mathrm{out}}\right)}{W_{\mathrm{comp}}} \tag{8}$$

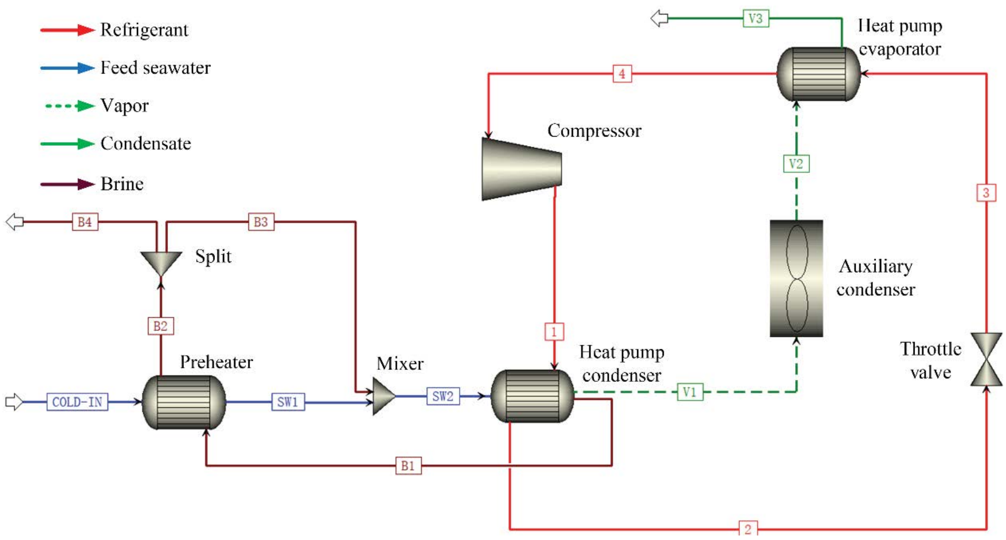

Fig. 3. Simulation model of the modified single-effect heat pump desalination system.

$$W_{\text{comp}} = \frac{m_r\left(h_{\text{rc,out}} - h_{\text{rc,in}}\right)}{\eta_s} \tag{9}$$

where Q_{hot} is the heating capacity that is the heat absorbed by the seawater, kW; m_r is the circulating refrigerant mass, kg/s; $h_{c,\text{in}}$ and $h_{c,\text{out}}$ is the enthalpy of refrigerant in and out of the condenser respectively, kJ/kg; W_{comp} is the refrigerant compression power, kW; $h_{\text{rc,out}}$ is the enthalpy of refrigerant out of the refrigeration compressor, kJ/kg; $h_{\text{rc,in}}$ is the enthalpy of refrigerant into the refrigeration compressor inlet, kJ/kg; η_s is the isentropic efficiencies of compressor.

Performance ratio (PR) and freshwater production are used to evaluate the performance of desalination system. According to the conservation of energy, the mass of freshwater production with a fixed circulating mass of refrigerant can be calculated by the following equation:

$$m_{\text{fw}} = \frac{m_r\left(h_{c,\text{in}} - h_{c,\text{out}}\right) - m_d\left(h_{d,\text{out}} - h_{d,\text{in}}\right)}{\gamma + \left(h_{s,\text{out}} - h_{s,\text{sa}}\right)} \tag{10}$$

Similarly, the circulating mass of refrigerant changing with the freshwater production is calculated by the equation as follows:

$$m_r = \frac{m_{\text{fw}}\left(\gamma + \left(h_{s,\text{out}} - h_{s,\text{sa}}\right)\right) + m_d\left(h_{d,\text{out}} - h_{d,\text{in}}\right)}{h_{c,\text{in}} - h_{c,\text{out}}} \tag{11}$$

where m_{fw} is the mass of freshwater, kg/s; m_d is the mass flowrate of seawater, kg/s; $h_{d,\text{in}}$ and $h_{d,\text{out}}$ is the enthalpy of seawater in and out of the condenser respectively, kJ/kg; γ is the latent heat of water evaporation, kJ/kg; $h_{s,\text{sa}}$ and $h_{s,\text{out}}$ is the enthalpy of saturate steam and superheat steam respectively, kJ/kg. Due to the reason of BPE, the steam out of the condenser has a certain value of superheat.

Fig. 4. Photograph of the tested single-effect heat pump desalination product.

PR is defined as the amount of distillate produce per 2,326 kJ of heat input or number of pounds of distillate produce per 1,000 Btu of heat. The defining formula is as follows [26]:

$$\text{PR} = \frac{2326 m_d}{W_{\text{comp}}} \tag{12}$$

3.4. Model validation

A single-effect heat pump desalination product developed by a company is presented in Fig. 4, which operates according to the conventional single-effect heat pump system as shown in Fig. 1, and utilizes refrigerant R134a. The product's test data were employed for the validation of simulation models. The testing condition is a mass flow rate of 80 kg/h for the feed seawater and evaporating/condensing temperatures of 50°C/10°C, respectively. A comparison between the test data and simulation results is illustrated in Table 1. Based on the error analysis presented in Table 1, the established models exhibited acceptable accuracy for performance simulation of heat pump desalination systems within an acceptable error range.

3.5. Parameter setting

The required basic input parameters for the system simulation are summarized in Table 2. Refrigerant of the heat pump is R134a. Base on the values applied in most of the literature reports and heat pump products, the superheat of refrigerant liquid and the subcooling of refrigerant gas are both set as 5°C. The circulation mass of refrigerant is coincidence with the freshwater production according to Eqs. (10) and (11).

Table 1
Comparison between Aspen Plus simulation results and experimental data

Parameter	Test data	Simulation results	Error (%)
T_v (°C)	39	40.35	3.46
W_{comp} (kW)	6.29	6.33	0.64
m_d (kg/h)	39.78	39.84	0.15
PR	14.71	14.64	0.48
COP	7.66	7.62	0.52

Table 2
Basic parameters

Parameter	Value
Feed seawater temperature °C	10~25
Feed seawater salinity, ‰	35
Feed seawater mass flow rate, kg/h	40~100
Seawater circulation ratio, %	60
Degree of superheat or super-cooling, °C	5

Due to the reason that the heat exchanger is not prone to scaling when the seawater temperature is below 70°C, the condensing temperature of the heat pump is set to be lower than 70°C. The concentration ratio defined as the ratio of the concentration of concentrated brine to the concentration of feed seawater is controlled to be less than 2.5 according to reference [23]. The heat pump desalination system is suitable for small capacity desalination. Thus the mass flowrate of the feed seawater is not exceeding 100 kg/h under all simulated conditions. To investigate the location effect of the auxiliary condenser, the heat dissipating capacity of auxiliary condenser in each system is the same, which is 2.5 kW for most simulation conditions. When the feed seawater mass flow increases, the dissipated heat will also increase to guarantee the concentration ratio. In the present study, the control strategy pertaining to the heat dissipation capacity of the auxiliary condenser has not been taken into consideration.

4. Results and discussions

According to the simulation results, a study on the performance difference between the conventional and modified systems operating under the same condition together with the influence of operation conditions is carried out. Firstly, the impact of the heat pump's evaporating/condensing temperature is analyzed while keeping the seawater feed condition constant and refrigerant circulation amount consistent. Secondly, the system's performance is evaluated by varying the mass of feed seawater while maintaining fixed brine concentration and feed seawater temperature, representing different freshwater production. Then, the influence of the feed seawater temperature is analyzed. Finally, variation of the auxiliary condenser's heat transfer area is discussed. It aims to validate the feasibility of the modified system and provide valuable insights for the optimal design of single-effect heat pump desalination systems.

4.1. Influence of heat pump evaporating/condensing temperature

This analysis maintains the feed seawater temperature and mass flowrate at 20°C and 80 kg/h, respectively, and the refrigerant circulation amount at 625 kg/h. Figs. 5 and 6 depict the performance variation of two systems when the heat pump evaporating temperature and condensing temperature are altered. Fig. 5 presents the changes in compression power and COP of the heat pump, whereas Fig. 6 shows the changes in freshwater production and PR of the desalination system. Despite the different placement of the auxiliary condenser in each system, the trends in performance variation with heat pump evaporating/condensing temperatures are consistent.

As depicted in Fig. 5, the COP value increases as the evaporating temperature increases or the condensing temperature decreases. The refrigerant compression power and the condensing heat of refrigerant gas remain the same under each heat pump operation condition due to the constant refrigerant mass flowrate. However, in the conventional single-effect system, a portion of the condensing heat is wasted by the auxiliary condenser. Therefore, COP of the modified system is higher than that of the conventional system under all simulated conditions. By relocating the auxiliary condenser, the COP increases from 5.79 to 6.24 when the evaporating/condensing temperature of the heat pump is 20°C/50°C.

From the results presented in Fig. 6, it can be found that the freshwater production exhibits a gradual decline while the PR demonstrates an increase as the evaporating temperature rises. The *p-h* diagram in Fig. 7 elucidates that the heat absorbed by the refrigerant in the evaporator increases a bit with a rise in evaporating temperature, accompanied by a bigger decrease in refrigerant compression power. As a result, heating capacity supplied by the condenser decreases and less seawater evaporates. Besides, the rate of reduction of freshwater production is lower than that of compression power, resulting in a higher PR for the heat pump desalination system. For instance, upon elevating the evaporating temperature from 10°C to 20°C at a condensing temperature of 50°C, both the conventional system and the modified system display an increase of 0.5 kg/h in freshwater production. Further, the PR of the conventional system and the modified system are enhanced by 5.0 and 5.4, respectively.

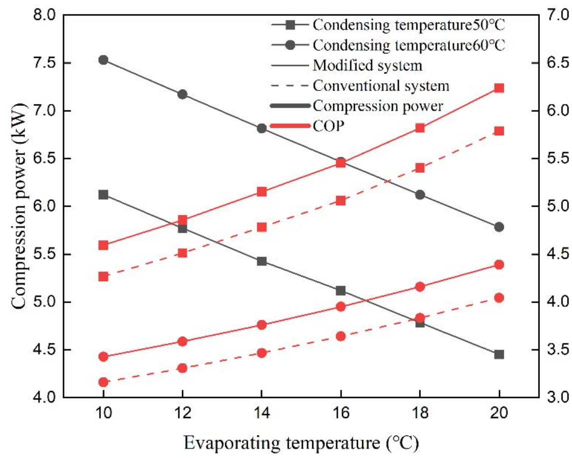

Fig. 5. Variation of refrigerant compression power and COP with the heat pump evaporating/condensing temperature.

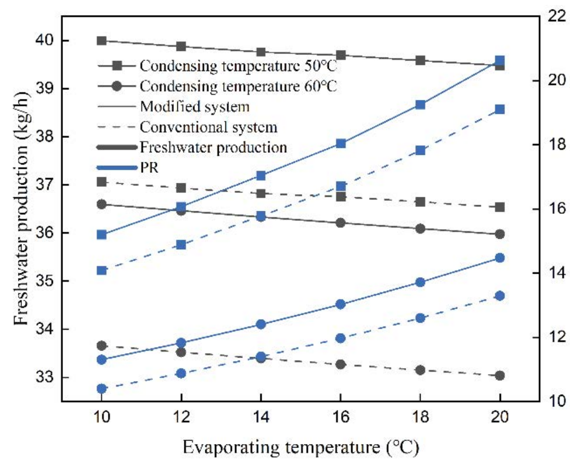

Fig. 6. Variation of freshwater production and PR with the heat pump evaporating/condensing temperature.

Fig. 7 demonstrates that the refrigerant compression power escalates with the surge in condensing temperature. However, with constant subcooling, the heat absorbed by the evaporator experiences a more substantial decline than the rise in compression power, causing a reduction in the heating capacity supplied by the condenser. Therefore, both the freshwater production and PR depict a decline as the heat pump condensing temperature increases, as shown in Fig. 6. For instance, upon increasing the condensing temperature from 50°C to 60°C at an evaporating temperature of 20°C, both systems show a decrease of 3.5 kg/h in freshwater production. Further, the PR of the conventional system and the modified system decrease by 5.8 and 6.2, respectively.

The results shown in Fig. 6 reveal that the modified system exhibits higher freshwater production and PR than the conventional system for all heat pump operating conditions. By modifying the location of auxiliary condenser, the freshwater production of the desalination system can be increased by 2.9 kg/h, independent of the heat pump evaporating temperature or condensing temperature. Moreover, the value of PR increases from 19.1 to 20.6 at a heat pump evaporating/condensing temperature of 20°C/50°C.

4.2. Influence of feed seawater mass flowrate

In order to investigate the effect of feed seawater flowrate, a series of fixed parameters are employed, including a brine concentration ratio of 2, a heat pump evaporating temperature of 10°C, a condensing temperature of 50°C, and a feed seawater temperature of 20°C. Therefore, different feed seawater mass flowrate represents varying freshwater production. The circulating mass of refrigerant is regulated according to Eq. (11). While the freshwater production remained constant between the conventional and modified systems, the focus of differentiation is on the heating capacity and power consumption of the heat pump.

Fig. 8 shows the relationship between compression power and COP of the heat pump system as a function of feed seawater mass flowrate, while Fig. 9 depicts the relationship between freshwater production and PR with respect to feed seawater mass flowrate. The increase in feed seawater flowrate causes an increase in compression power but does not alter the COP since the heat pump's operating conditions remain constant. However, as the mass of steam produced by seawater evaporation increases with greater feed seawater flowrate, more heat is required, resulting in a rise in circulation mass of refrigerant and compression power. The modified system, which does not waste any condensing heat, experiences a smaller increase in refrigerant circulation mass and thus has a larger COP and lower compression power. COP of the modified system is 4.6, which represents a 7% increase over that of the conventional system which is 4.3. Additionally, the benefit of compression power reduction will increase as feed seawater mass flowrate increases.

The PR values of both heat pump desalination systems remain constant with varying feed seawater mass flowrates

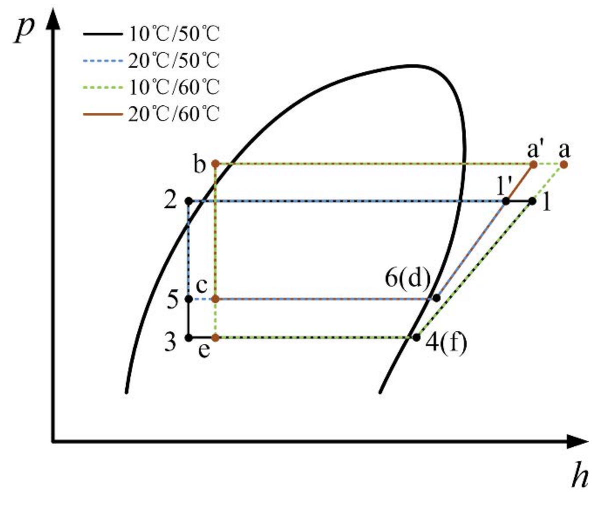

Fig. 7. The *p-h* diagram of a heat pump operating process: 1, a, actual compressor outlet status point; 4, 6, d, f, compressor inlet status point; 2, b, throttle valve inlet status point; 3, 5, c, e, throttle valve outlet status point.

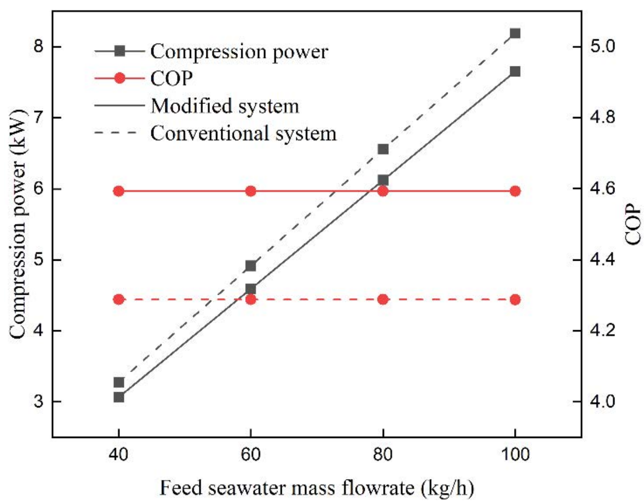

Fig. 8. Variation of compression power and COP with the feed seawater mass flowrate.

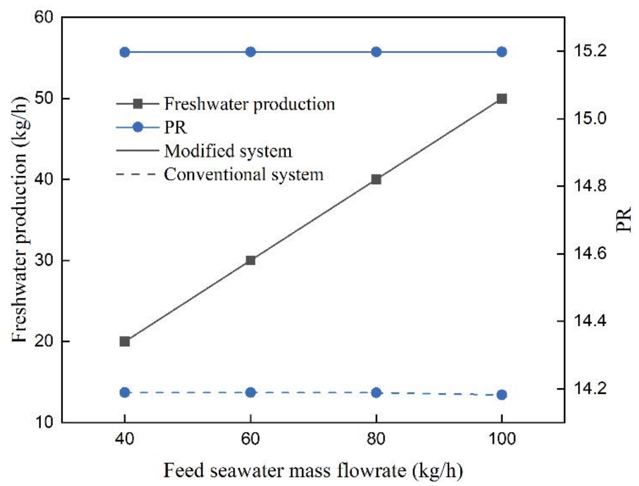

Fig. 9. Variation of freshwater production and PR with the feed seawater mass flowrate.

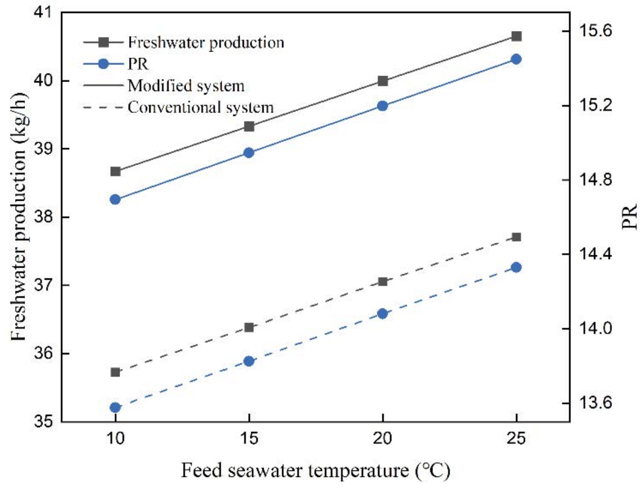

Fig. 10. Variation of freshwater production and PR with the feed seawater temperature.

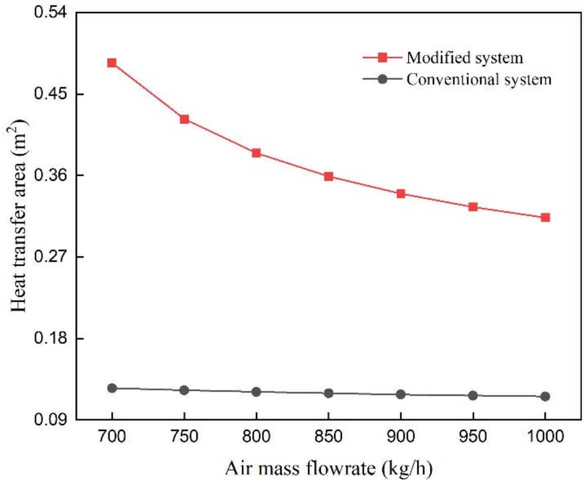

Fig. 11. Variation of heat transfer area with the air mass flow rate.

due to the concurrent enhancement of compression power and freshwater production. However, the modified system surpasses the conventional system in terms of PR value, as shown in Fig. 9. Specifically, relocating the auxiliary condenser leads to a noteworthy enhancement in PR, escalating from 14.2 to 15.2, corresponding to a remarkable growth rate of 7%. Based on these results, it can be inferred that the modified system outperforms the conventional system in terms of energy efficiency and heat pump equipment size at a given concentration ratio target and feed seawater flowrate.

4.3. Influence of feed seawater temperature

During the operation of a desalination plant, the temperature of feed seawater may vary depending on its origin. To examine the impact of this variation, fixed parameters are employed, including the evaporating/condensing temperature of the heat pump at 10°C/50°C, refrigerant circulation mass flowrate of 625 kg/h and feed seawater mass flowrate of 80 kg/h. For a constant heat pump operational condition, the alteration in feed seawater temperature does not affect the compression power or COP of the heat pump. Fig. 10 illustrates the change in freshwater production and PR corresponding to variations in feed seawater temperature.

As shown in Fig. 10, both the freshwater production and PR increase with the increase in feed seawater temperature, irrespective of any location of the auxiliary condenser. This trend is due to a decrease in sensible heat absorption of the feed seawater before evaporation as feed seawater temperature rises. As a result, mass of evaporated seawater increases for a fixed heating capacity supplied by the condenser. Moreover, Fig. 10 shows that the modified system produces higher freshwater production and PR than the conventional system, and the performance increase is unaffected by the feed seawater temperature. After modification of the auxiliary condenser location, the increments of freshwater production and PR are 2.9 kg/h and 1.1, respectively.

4.4. Heat transfer area of the auxiliary condenser

The location modification of the auxiliary condenser decreases the heat transfer temperature difference and consequently increases the required heat transfer area for a fixed heat transfer capacity. Heat transfer area is one of the primary factors affecting the size of the auxiliary condenser. Fig. 11 depicts the variation in heat transfer area difference under different air flowrates, considering the auxiliary condenser as an air-cooled heat exchanger with surrounding air temperature at 25°C. The refrigerant circulation mass flowrate of the heat pump system is 625 kg/h, and the values of heat transfer area represent the inner surface area of the heat exchange pipe.

It can be found that the heat transfer area of the auxiliary condenser is higher in the modified system. Meanwhile, the heat transfer area of each auxiliary condenser in the conventional and modified systems decreases with increase of air mass flowrate. However, the reduction rate of the auxiliary condenser in the modified system is greater. This is because of that the required heat transfer area is a ratio of the heat transfer capacity to the product of heat transfer coefficient and temperature difference. The heat transfer coefficient increases with the rise of air mass flowrate. And the heat transfer temperature difference of the auxiliary condenser in the conventional system is greater. Thus, the influence of the rise of heat transfer coefficient is smaller. At an air mass flowrate of 1,000 kg/h, the heat transfer temperature difference of the auxiliary condenser is 25.4°C for the conventional system and 9.4°C for the modified system. Therefore, the heat transfer area of the auxiliary condenser in the modified system is higher, and the influence of air flowrate changes is greater. When the air mass flowrate is 700 kg/h, the heat transfer area of the auxiliary condenser in the conventional system is 0.125 m^2, which increases to 0.484 m^2 after the location modification. However, the sacrifice of heat transfer area implies that the type of water-cooled plate heat exchanger would be more appropriate for the modified desalination system.

5. Conclusion

This paper proposed a modified single-effect heat pump desalination system, in which the auxiliary condenser is relocated to the low-temperature steam pipeline. Performances of both the conventional system and the modified system are simulated using Aspen Plus simulation software considering various parameters such as the heat pump evaporating and condensing temperatures, as well as the feed seawater mass flowrate and temperature. With the performance comparison, the influence of location of the auxiliary condenser is discussed. Base on the results, the following conclusions can be drawn:

- The modified single-effect heat pump desalination system exhibits superior values of COP and PR compared to the conventional system under all simulated conditions, regardless of the heat pump evaporating or condensing temperature. Moreover, both systems display similar trends: the freshwater production decreases while the PR increases with the increase of evaporating temperature, whereas both the freshwater production and PR decrease with the increase of condensing temperature.
- When the heat pump operates under a fixed condition and the target brine concentration ratio remains constant, the refrigeration compressor of the modified system exhibits a lower compression power consumption. Moreover, the aforementioned benefit escalates as the mass flowrate of treated feed seawater or the desired freshwater production increases. For different feed seawater mass flowrate conditions, the PR values of the modified system and conventional system are determined to be 14.2 and 15.2, respectively, which corresponds to a growth rate of 7%. Furthermore, through the location modification of the auxiliary condenser, the freshwater product and PR can be increased by 2.9 kg/h and 1.1, respectively, under conditions of constant feed seawater flowrate but differing feed seawater temperature.
- Location modification of the auxiliary condenser results in a decrease in the heat transfer temperature difference and, consequently, an increase in the required heat transfer area for the auxiliary condenser. The water-cooled plate heat exchanger would be a more appropriate for the modified system.

In conclusion, the location modification of the auxiliary condenser can lead to the increment in system performance when the heat dissipating capacity of the auxiliary condenser remains unchanged. However, design of the auxiliary condenser including its type selection and heat dissipating capacity determination should take the control strategy into consideration. Additionally, the modification also affects the system equipment cost, in view of the consequential changes in the heat transfer characteristics of both the condenser and evaporator. Consequently, further comprehensive investigation is warranted to develop the modified single-effect heat pump desalination system.

Symbols

γ — Latent heat of water evaporation, kJ/kg
η_s — Isentropic efficiencies of compressor
$h_{c,\text{in}}$ — Enthalpy of refrigerant into of the condenser, kJ/kg
$h_{c,\text{out}}$ — Enthalpy of refrigerant out of the condenser, kJ/kg
$h_{d,\text{in}}$ — Enthalpy of seawater into of the condenser, kJ/kg
$h_{d,\text{out}}$ — Enthalpy of seawater out of the condenser, kJ/kg
$h_{\text{rc,in}}$ — Enthalpy of refrigerant out of the refrigeration compressor, kJ/kg
$h_{\text{rc,out}}$ — Enthalpy of refrigerant into of the refrigeration compressor, kJ/kg
$h_{s,\text{sa}}$ — Enthalpy of saturate steam, kJ/kg
$h_{s,\text{out}}$ — Enthalpy of superheat steam, kJ/kg
m_d — Distillate production/freshwater production, kg/s
m_{fw} — Mass of freshwater, kg/s
m_r — Circulating refrigerant mass, kg/s
A — Heat transfer area, m^2
Q_a — Heat dissipating capacity of the auxiliary condenser, kJ
Q_{hot} — Heating capacity, kJ
T — Temperature of seawater, °C
T_b — Boiling temperature of seawater, °C
T_v — Condensing temperature of steam, °C
U_a — Total heat transfer coefficient of auxiliary condenser, kW/(m^2·°C)
U_c — Heat transfer coefficient of heat pump evaporator, kW/(m^2·°C)
U_e — Heat transfer coefficient of heat pump condenser, kW/(m^2·°C)
W_{comp} — Refrigerant compression power, kW
X — Salinity, ‰
ΔT — Heat transfer temperature difference of the auxiliary, °C

Acronym

COP — Coefficient of performance
PR — Performance ratio

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