Experimental investigation on an enhanced humidification-dehumidification solar desalination system with weakly compressed air and internal heat recovery

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Abstract

Humidification-dehumidification (HDH) process has proven to be a feasible method to obtain fresh water. On-going research on HDH desalination system has been carried out to improve the system performance and maintain steady fresh water production. However, it is difficult to guarantee the efficiency and stability of fresh water production at the same time. In this paper, an enhanced HDH desalination system with weakly compressed air and internal heat recovery is thus proposed, and the experimental setup is then designed and fabricated. Benefiting from a new-design evaporative-condenser, the latent heat released by the process air can be recycled completely, which can provide a continuous and steady heat input for the evaporation of seawater. Therefore, additional heat source and cooling seawater are no need any more during steady running. The experimental results show that when the spraying seawater temperature is heated to 70 ºC, and both evaporation pressure and condensation pressure are -1444 Pa and 943 Pa, respectively, the fresh water production and specific energy consumption (SEC) are 116 g h^{-1} and 1.01 kg $kW^{-1}h^{-1}$. By experiments, we have confirmed the feasibility of enhanced-HDH desalination. Meanwhile, the theoretical analysis of the system self-sustaining is also verified.

Keywords: humidification-dehumidification (HDH), desalination, moist air, fresh water,

1. Introduction

With the rapid development of society and quality improvement of people's living, the importance of fresh water has become more and more prominent. But at the same time, the shortage of fresh water resources and water contamination have also been paid much more attention by people (Li, 2013). Fortunately, seawater covers three-quarters of earth's surface, and it contains a huge amount of fresh water if the salt can be separated from the seawater. Therefore, removing most of salt from the seawater richly on earth to obtain fresh water, namely desalination technology, has proven to be a promising and feasible method to obtain fresh water from seawater for daily life, agriculture and industry (Elimelich, 2011; Shatat 2013).

Desalination technologies as we know such as reverse osmosis (RO), multi-stage flash (MSF), low-temperature multi-effect distillation (LT-MED) and mechanical vapor compression (MVC) are suitable for large-scale fresh water supply. Desalination units using above methods are both located near large chemical plants or power plants, in which electricity, steam or waste heat is particularly rich, and not suitable for the place with restricted in space and resources (Srithar, 2018). In other words, above methods are usually highly energy-intensive and driven by traditional fossil fuels, leading to global warming intensifies. Therefore, seeking for the low-carbon and clean as well as renewable energy to take place of the traditional fossil fuels in desalination system is imperative. However, the traditional desalination methods have serious requirement for working temperature and pressure. In this context, since 21st century, the process of HDH has attracted close attention by its outstanding merits, such as simple structure, low working temperature, mild evaporation, easy to control, atmospheric operation and can be directly drunk and so on. Furthermore, profit from the low working temperature and atmospheric operation, the process of HDH can be driven by renewable and sustainable energy such as solar energy and geothermal energy as well as energy-saving technologies such as heat pump technology. A combination of HDH process with clean and renewable energy has been recognized as a practicable method to obtain drinkable water in small-scale (Dai, 2000 and 2002; Zheng, 2017). Compared with traditional desalination methods, HDH process is susceptive to the stability of heat sources, especially when the heat source is solar energy. In this context, many researches on HDH

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desalination system aim to improve the efficiency of heat recovery or combine with other technologies such as heat pump, thus reducing the demand for the stable heat sources.

Kang et al (2015) presented a two-stage multi-effect solar desalination system based on HDH process, in which both the latent heat and partial sensible heat in the remaining seawater are reused for further improving the fresh water yield. The simulation results show that when the spray seawater temperature of the higher-temperature humidifier is 95 ºC, the fresh water yield can reach up to 72.60 kg h^{-1} and the corresponding GOR is 2.44. However, it is hard to obtain a temperature of 95 ºC in traditional solar collector. Meanwhile, when the spray seawater temperature is less than 70 ºC, the corresponding GOR will also be decreased to less than 2. Wu et al (2017a, 2017b) reported a three-stage solar HDH desalination system to further improve the system performance, including GOR and fresh water yield. Between higher temperature HDH and lower temperature HDH, a middle temperature HDH was added to improve energy cascade consumption. The maximum GOR can reach up to 2.65 only when the spray seawater temperature at higher temperature HDH is 85 ºC. However, from above literature review, we can see that a relatively high spray seawater temperature needs to be provided for above three-stage desalination systems to achieve the optimal system performance, especially when it is driven by the traditional solar collectors.

However, the aforementioned solar HDH desalination system is single-heat source system, which is susceptible to the solar irradiation intensity. Therefore, an effectiveness solution to this problem is to combine with a heat pump to provide heat source and cold source, respectively. Xu et al (2018) proposed a SAHP desalination based on HDH process to obtain fresh water efficiently and steadily, and a test rig was also designed. The experimental results indicate that the maximum fresh water yield and GOR are 12.38 kg $kW^{-1}h^{-1}$ and 1.24, respectively. However, the system performance can be further improved if the preheated seawater from the precooler is reused again. Xu et al (2019a) therefore proposed a two-stage humidification solar assisted heat pump desalination system with HDH process to further improve the fresh water productivity and GOR. It can be seen from the experimental results that the maximum fresh water productivity has increased from 12.38 kg $kW^{-1}h^{-1}$ to 14.35 kg $kW^{-1}h^{-1}$, and the value of GOR has also increased from 1.24 to 1.93. However, the system performance is constrained by a bottleneck: increasing dehumidifying capacity can result in a reduction in the performance of lower-temperature (LT) humidifier. Consequently, a modified system is then proposed by Xu et al (2019b) to solve this bottleneck effectively. And the maximum fresh water yield can be increased by 16.70% to 20.54 kg/h, and the corresponding maximum *GOR* is also increased by 18.05% to 2.42.

Strictly speaking, it usually loses more than you gain if a heat pump is introduced during HDH process due to its high-power consumption and higher initial cost. As a result, the system structure is no longer simple flexible. If a compressor instead of seawater heaters or/and process air heaters is used as the driving component of the system, and an expander or throttle valve is added between humidifier and dehumidifier to decrease the pressure of evaporation for achieving high effective of evaporation of seawater in humidifier, it is easy to identify that there are no heat sources directly involved in the operation of the HDH desalination system. In other words, the working principle of the pressure-driven HDH desalination system is derived from the heat pump cycle in which the evaporator absorbs directly heat produced by the condenser.

Based on working principle of pressure-driven HDH process, Ettouney (2005) proposed a humidification vapor compression desalination system, in which both humidifier and dehumidifier are placed in the same chamber. And the condensation of compressed process air in the dehumidifier provides the heat input for the evaporation of seawater in humidifier. However, the waste heat of seawater and process air is not reused. Therefore, Narayan et al (2011) proposed a novel variable pressure HDH desalination system, in which both humidifier and dehumidifier are operated at different pressures respectively and independent of each other. Therefore, the effects of pressure ratio (the ratio of condensation pressure to the evaporation pressure) and condensation pressure as well as the air side pressure drop on the system performance was investigated by the authors. Meanwhile, owing to the huge increase in entropy generation, a throttle valve instead of expander will lead to an increase in SEC. And GOR of the system can reach up to 6, which is higher traditional HDH desalination system. Besides, the compressor driven by high-temperature steam has also been analyzed by Narayan et al (2012), and the expander is coupled with the RO technology. As seen, the maximum GOR of the proposed system is 20, and the corresponding SEC is about 105.26 kg $kW^{-1}h^{-1}$, which both are much higher than those of the traditional HDH desalination system. According to the system proposed by Narayan et al (2012), the exergy analysis of above proposed system was carried out by Al-Sulaiman et al (2013) who also developed a novel parameter, the total true specific exergy lost, to evaluate the exergetic performance of the system. As found from the simulation results that the exergy destruction of the thermal vapor compressor is the largest, accounting for 50% of the total exergy destruction. And the similar work has also been performed by Siddiqui et al (2017).

From above recent investigations and attempts to increase the system performance or reduce the system energy consumption, it is not hard to find that cooling seawater is essential to the conventional HDH desalination system expect direct-contact condensation, especially to the process of dehumidification. However, it is impossible to recycle completely the heat released by the condensation of process air in the dehumidifier, and only a fraction of the heat recovered can be used owing to the efficiency and heat loss in the energy conversion process. If the condensation heat can be directly transferred to the evaporation side of seawater, not only the heat released in the condensation side of process air can be recycled completely, but also the cooling seawater is no longer required (Vlachogiannis 1999; El-Khatib, 2004; Xu, 2019c).

In this paper, an enhanced solar HDH (E-HDH) desalination system with weakly compressed air and internal heat recovery is thus proposed, and the experimental setup is then designed and fabricated. Benefiting from a new-design evaporative-condenser, the latent heat released by the process air can be recycled completely, which can provide a continuous and steady heat input for the evaporation of seawater. Therefore, additional heat source and cooling seawater are no need any more during steady running. The objective of this paper is to verify our proposed system and investigate the system thermodynamic characteristics. And the thermal performance of the system is studied under different operating conditions such as seawater temperature and pressure difference.

2. System description

A novel E-HDH solar desalination system is formed when compressor and vapor chamber in traditional mechanical vapor compression (MVC) desalination system are substituted by an air blower and an evaporative-condenser, respectively, which is shown in Fig. 1. An air blower instead of mechanical compressor is helpful to reduce the working pressure. Meanwhile, water vapor, as the work medium of a traditional MVC desalination system, is also replaced by the moist air in the novel system. Therefore, the novel E-HDH solar desalination system performance is not only determined by the working pressure of moist air and seawater temperature, but also by the moist air conditions.

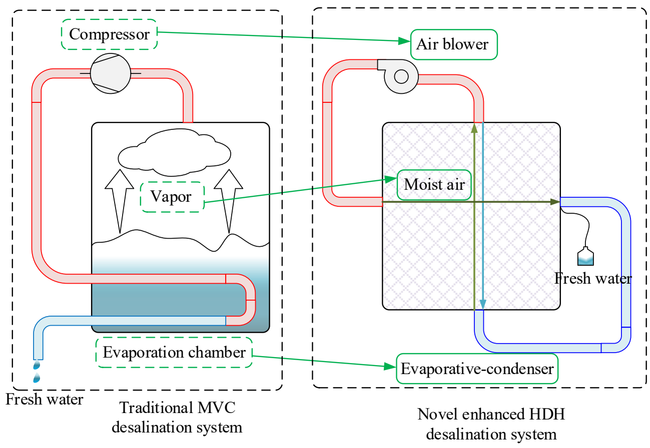

Fig. 1 Development of a novel E-HDH desalination system

Fig. 2 depicts the detailed schematic diagram of the novel E-HDH solar desalination system based on Fig. 1. After being heated to the setting temperature by the seawater heater, the hot seawater is uniformly sprayed on the top of the evaporation channel. The high temperature falling film is formed on the surface of the evaporation channel, which is greatly beneficial to the evaporation of seawater. Unsaturated moist air under evaporation pressure after throttling flows from bottom to top through the falling film surface of the evaporation channel for heat and mass transfer, thus reaching the nearly saturated moist air. Then the saturated moist air out of the evaporative channel is compressed adiabatically by air blower. The compressed moist air is unsaturated. Next, the compressed moist air passes through the condensation channel, where partial water vapor is condensed into fresh water. Better yet, the condensation latent heat can be completely recycled for seawater evaporation in the evaporative-condenser for reheating seawater again. The compressed moist air under condensation pressure contributes to promote the condensation of water vapor. After condensation, the saturated moist air is depressurized to unsaturation by throttle valve. At last, the unsaturated moist air is sucked into the evaporation channel by air blower to increase the moisture content again. The remaining seawater drawn from the evaporative-condenser is still warm, and its concentration is not changed much. Hence, it is pumped into the hot seawater sprayer again.

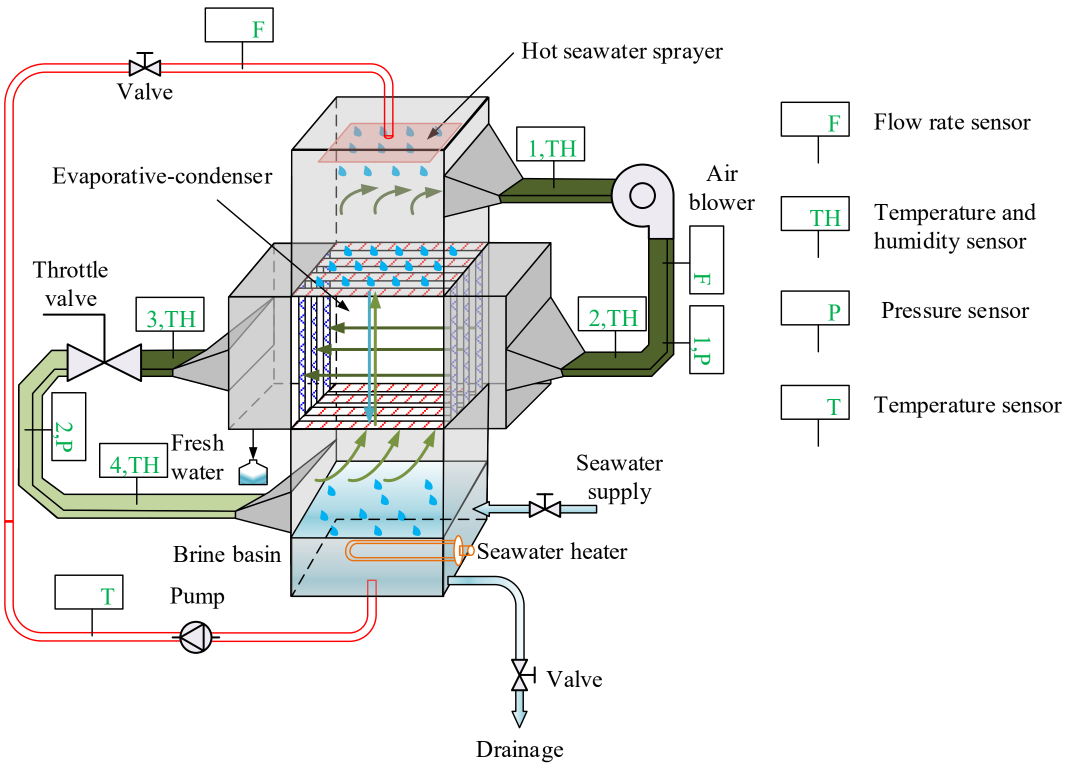

Fig. 2 Schematic of the novel E-HDH solar desalination system

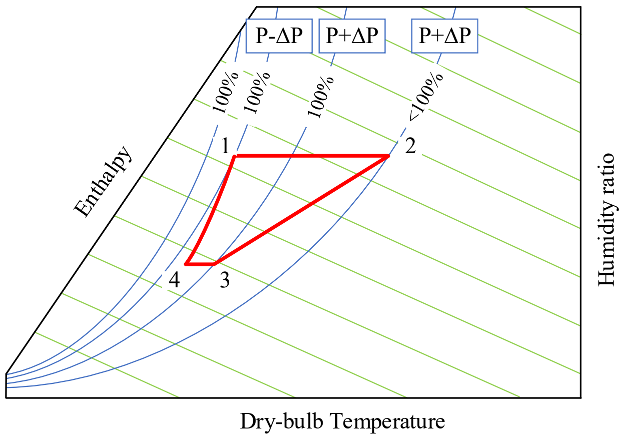

Fig. 3 State points of process air during E-HDH process in the psychrometric chart

The state points of process air during E-HDH process in the psychrometric chart is depicted in Fig. 3. In the operation of satisfactory performance, the moist air driven by air blower will experience the following procedures: 1→2→3→4. At this point, the throttle valve has been activated, and positive (P+ΔP) and negative (P-ΔP) pressure has formed in the air duct, which are beneficial for the evaporation of seawater and condensation of moist air. The saturated moist air after humidifying is compressed by the air blower to achieve the procedure of 1→2, a process of enthalpy-adding. Thus, the dry-bulb temperature of moist air is increased, and the absolute humidity ratio keeps constant. Then, the compressed unsaturated moist air passes through the condensation channel to experience the procedure of 2→3. The dry-bulb temperature is reduced to its dew point temperature firstly, and fresh water is then condensed on the surface of the heat transfer wall. Meanwhile, the state of moist air is returned to saturated from unsaturated. The saturated moist air becomes unsaturated when the moist air flows through the throttle valve from state point 3 to 4 (3→4) due to the pressure drop. At last, the unsaturated moist air after throttling passes through the evaporation channel to increase moisture content again from state point 4 to 1 (4→1).

3. Experimental setup

2.1. Experimental setup description

A photograph of the experimental setup is shown in Fig. 4. The major components include air blower, seawater heater, throttle valve, evaporative-condenser, spraying seawater tank, pump, flow rate sensors and temperature and humidity sensors.

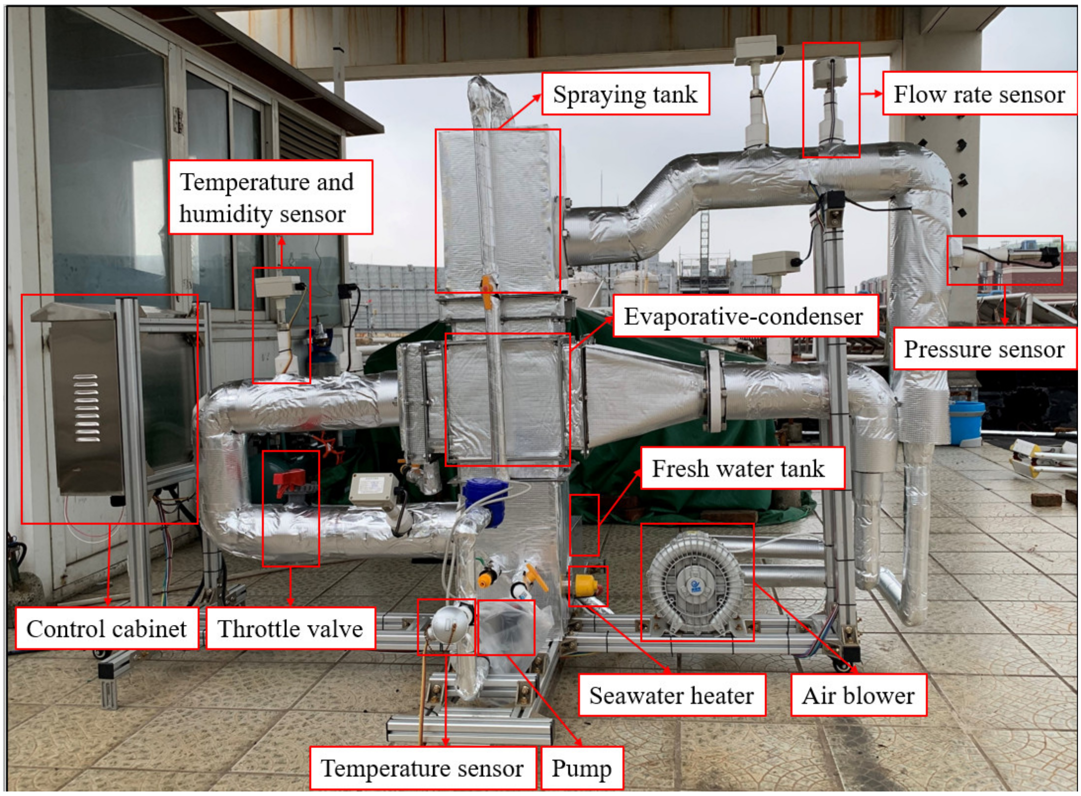

Fig. 4 Photograph of the experimental setup

An electrical heater instead of solar collectors is utilized to acquire the test results rapidly. Meanwhile, a turbofan instead of the traditional axial flow fan or centrifugal fan is used as the pressurizing device to provide a large pressure difference. Flange connection is adopted to ensure the airtightness of the whole system. Similarly, in order to the airtightness of the system, all the sensors are also installed with threaded connection. In order to avoid sucking spray seawater into the turbofan leading to burning out the motor, the air duct outlet of the spraying tank is therefore designed to rise at an Angle of 45 degrees. The pipe diameter of the air duct is 110mm. The control cabinet is designed to control the start and stop of air blower, pump and seawater heater, adjust the air blower frequency, and acquire experimental data.

Fig. 5 shows the configuration of the evaporative-condenser, it can be found that the size of evaporative-condenser is 300mm*300mm*300mm, and the pitch of fins is 5mm. The thickness of fins is set to 0.1mm to enhance the performance of heat transfer between moist air and seawater. Furthermore, the technique of concave and convex points is also applied to the fins for increasing the heat exchange area.

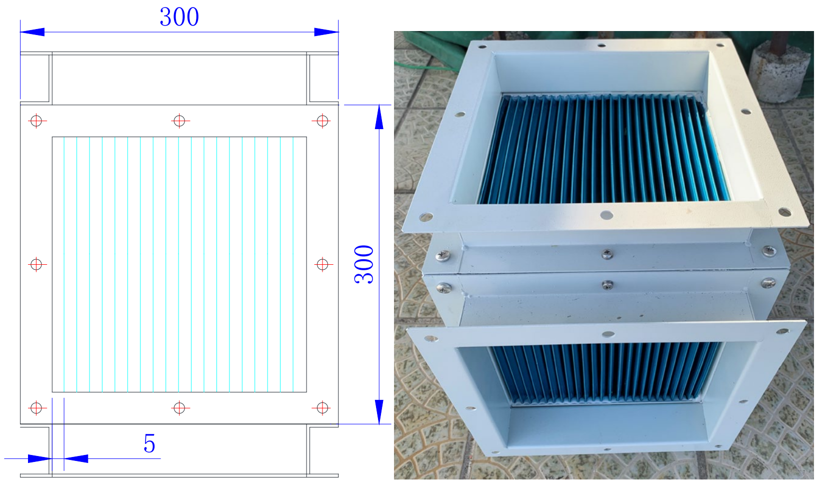

Fig. 5 Configuration of evaporative-condenser

The parameters of major component above mentioned in the experimental setup is illustrated in Table 1.

Tab. 1: Parameters of major components in the setup

Components	Parameters
Evaporative-condenser	Size: (Length × Width × Thickness) L ×W ×T=300 mm ×300 mm ×300 mm
Air blower	Max. volume flow: 80 m^3 h^{-1}, max. discharge pressure: 12 kPa, max. suction pressure: 11 kPa, power: 370W
Pump	Max. volume flow: 0.75 m^3 h^{-1}, power: 55W
Seawater heater	Power: 3 kW

2.2. Measurements and data acquisition

Some of parameters floating with the running time would be collected into computer by MODBUS-RTU communication protocol to keep real-time records. A throttle valve was adopted to control the pressure ratio and pressure difference. A frequency changer was used to adjust the speed of air blower so as to broaden the regulating capacity of the throttle valve. And the flow rate of seawater was adjusted by the seawater valve.

To evaluate the system performance, both process air temperature and relative humidity were measured by temperature and humidity sensors. Process air pressure was recorded by a pressure sensor. Fresh water production was collected by an electronic balance. A smart electrical meter was adopted to obtain the electrical consumption of the system. A hot-film anemometer was used to measure process air speed. Additionally, the temperature and humidity of process air, seawater temperature, electrical consumption and fresh water production were archived with a sampling interval of 1minute. The specific features of the testing apparatus are listed in Table 2.

Tab. 2 Specific features of the testing sensors

Parameters	Sensors	Testing ranges	Accuracy
Temperature	PT100	0-100 °C	±0.2 °C
Temperature and humidity	Temperature and humidity	0-100 °C 0-RH100%	±0.5 °C ±3%
Pressure	Pressure	-5.0 kPa-5.0kPa	±0.5%
Air speed	Hot-film anemometer	0-10 m s^{-1}	±3%
Water flow	Water meter	0-2.5 m^3 h^{-1}	±2%

Fresh water production	Electronic balance	0-20 kg	±1 g

4. Results and discussion

Fig. 6 shows the change trends of spraying seawater temperature and process air temperature with running time. The operating conditions are: the flow rate of spraying seawater is 0.75 m^3/h, and wind speed is 1.80 m/s. It should be point that, in this case, the spraying seawater temperature (T_s) is increased to 70 °C at the 10th minute by the seawater heater which is used to simulate the heating process of solar collector. It is clear that the moist air temperature is also increased with the increase of spraying seawater temperature, including the moist air outlet temperature of low-pressure (T_1), the moist air inlet temperature of high-pressure (T_2), the moist air outlet temperature of high-pressure (T_3) and the moist air inlet temperature of low-pressure (T_4). The seawater heater will be turned off when the spraying seawater temperature reaches the designed temperature. However, as the experiments went on, the moist air temperature is decreased without the heat source, thus leading to a reduction in the spraying seawater temperature. It is mainly because that the temperature difference between the moist air and the ambient air is objective existence, which is bound to cause some degree of heat loss even if we have already done the corresponding insulation measures. When the temperature of spraying seawater reaches 70 °C, the moist air inlet and outlet temperature of high-pressure are 65 °C and 63.6 °C, and the moist air inlet and outlet temperature of low-pressure are 62.4 °C and 66.2 °C, respectively.

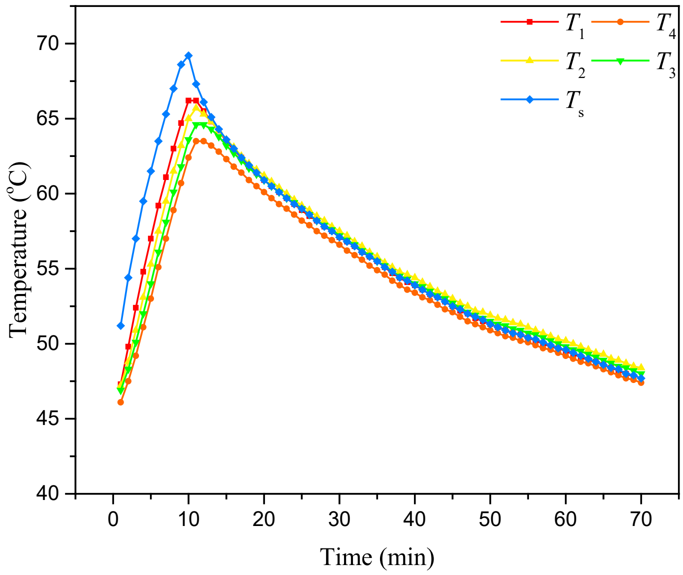

Fig. 6 The change trends of seawater and process air temperature with running time

Fig. 7 depicts the change trends of evaporation pressure and condensation pressure with running time. It can be found from the figure that the average evaporation pressure (P_1) is -1444 Pa, and the corresponding average condensation pressure (P_2) is 943 Pa. The value of pressure here is all relative to the atmospheric pressure, namely the gauge pressure. In general, when evaporation pressure is larger than the condensation pressure, it is advantageous for the system to increase the fresh water production. Therefore, the average pressure difference distribution which can be defined as the ratio of the pressure difference between atmosphere pressure and evaporation pressure to the value of pressure difference is 0.60. Need to point out that in order to effectively alleviate the heat resistance rise between moist air and heat transfer wall in the condensation channel due to the existence of the liquid film produced by the condensation of water vapor, air evacuation valve is opened regularly to blow away the liquid film in the condensation channel. Therefore, a sudden drop in both evaporation pressure and condensation pressure is presented in Fig. 7.

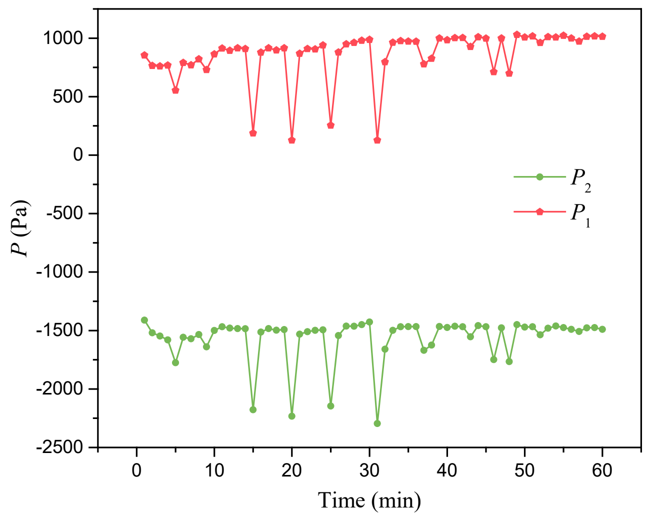

Fig. 7 The change trends of evaporation pressure and condensation pressure with running time

Fig. 8 shows the temperature difference between the moist air inlet temperature of high-pressure (T_2) and the moist air outlet temperature of high-pressure (T_3), and the temperature difference between the moist air outlet temperature of high-pressure and the spraying seawater temperature (T_s). According to the working principle of the system, the fresh water may be obtained only when the moist air inlet temperature of high-pressure is greater than the moist air outlet temperature of high-pressure. From fig. 8 we can see that the moist air inlet temperature of high-pressure is always greater than the moist air outlet temperature of high-pressure. Under the stable operating conditions, the average temperature difference is about 0.3 ºC. A measure of whether the system is self-sustaining which means that additional heat input is no longer needed, is the temperature difference between the moist air outlet temperature of high-pressure and the spraying seawater temperature. When the moist air outlet temperature of high-pressure is higher than the spraying seawater temperature, the condensation heat will be absorbed by the spraying seawater in the evaporation channel. It can be seen that the moist air outlet temperature of high-pressure is higher than the spraying seawater temperature from the 44th minute, and the average temperature difference is about 0.2 ºC.

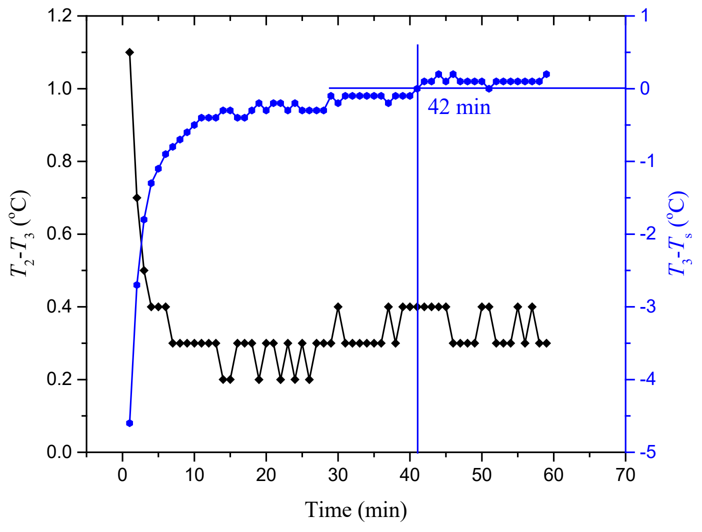

Fig. 8 The change trends of temperature difference with running time

Fig. 9 illustrates the change trends of fresh water production with running time. Within the 60-minute running, a total of 116 g fresh water was obtained accumulatively. Under the operating conditions, the power consumption of pump and air blower are 15 W and 100 W. Therefore, the system SEC which is used to evaluate the system performance in electrical energy consumption is 1.01 kg kW^{-1}h^{-1}. The reason why the system SEC is too low can be explained by the fig. 10.

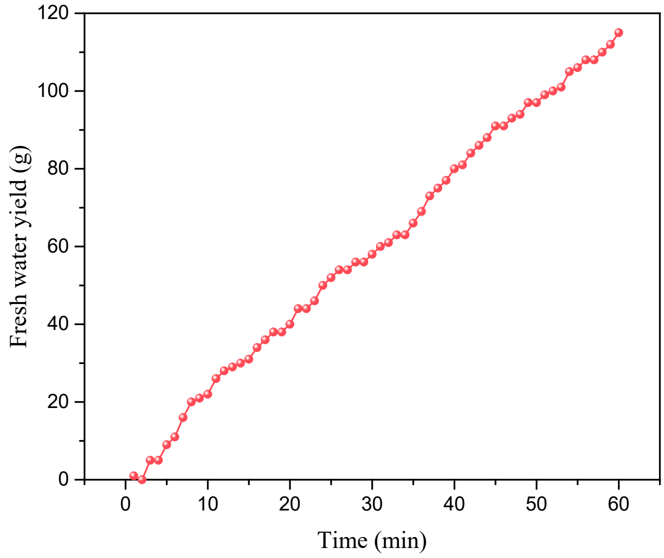

Fig. 9 The change trends of fresh water yield with running time

Fig. 10 shows the change trends of the temperature difference between the moist air inlet temperature of high-pressure (T_2) and the moist air outlet temperature of low-pressure (T_1), and the humidity ratio difference between the moist air inlet humidity ratio of high-pressure (ω_2) and the moist air outlet humidity ratio of high-pressure (ω_3). Limited by the evaporative-condenser manufacturing process, the maximum pressure capacity of the pressure difference between evaporation channel and condensation channel is only 2500 Pa. Otherwise, it will be broken down. Therefore, the temperature rise of the moist air after compressed is also small, which is 0.7 °C. Consequently, the temperature difference between the moist air inlet temperature of high-pressure (T_2) and the moist air outlet temperature of high-pressure (T_3) is small, leading to a small humidity ratio difference between the moist air inlet humidity ratio of high-pressure (ω_2) and the moist air outlet humidity ratio of high-pressure (ω_3). Under the stable operating conditions, the average humidity ratio difference is only 2.1 g kg^{-1}. Hence, the system fresh water production is low accordingly. The measure to overcome above problem is to improve the processing technology of evaporative-condenser to improve its pressure endurance.

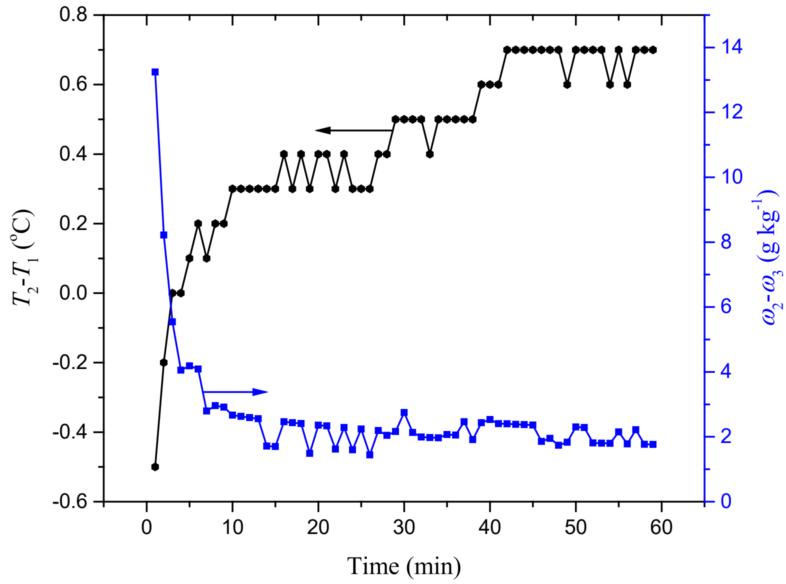

Fig. 10 The change trends of temperature difference and humidity ratio difference with running time

5. Conclusions

In this paper, an enhanced solar HDH (E-HDH) desalination system with weakly compressed air and internal heat recovery is proposed to lossless reuse the condensation latent heat and move away from a reliance on the cooling seawater and additional heat sources. An experimental setup has been designed and fabricated. Based on the experimental results, the system performance has also been investigated. The conclusions are listed as follows:

(1) The feasibility of enhanced-HDH desalination has been confirmed experimentally. Meanwhile, the theoretical analysis of the system self-sustaining is also verified.

(2) When the spraying seawater temperature is heated to 70 ºC, and both evaporation pressure and condensation pressure are -1444 Pa and 943 Pa, respectively, the fresh water production and SEC are 116 g h^{-1} and 1.01 kg $kW^{-1}h^{-1}$.

(3) Limited by the evaporative-condenser manufacturing process, the maximum pressure capacity of the pressure difference between evaporation channel and condensation channel is only 2500 Pa. Otherwise, it will be broken down. Therefore, the temperature rise of the moist air after compressed is also small. Therefore, the measure to overcome above problem is to improve the processing technology of evaporative-condenser to improve its pressure endurance. In the following work, we will improve the processing technology of the evaporative-condenser.

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