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Thermodynamic Analysis of A Novel Humidification Dehumidification Desalination System Driven by Heat Pump

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Abstract

In this paper, the humidification dehumidification (HDH) desalination system using a packed bed dehumidifier, with a mechanical vapor compression heat pump integrated, is proposed, and the condenser is coupled to raise the seawater temperature while the evaporator is applied to absorb the energy from the discharged brine as much as possible. For the proposed combined desalination system, the mass and energy balance equations are constituted, and then the corresponding thermodynamic performance at the design conditions, both for the desalination and heat pump unit, is obtained. The simulation results show that the peak values for the water production and gained-output-ratio (GOR) emerge as 76.40kgh^{-1} and 5.14 at the design conditions. Compared to the general HDH desalination system, the elevation magnitudes reach ΔGOR=1.69 and 2.86 at the same conditions.

Keywords: humidification dehumidification; packed bed dehumidifier; mechanical vapor compression heat pump; thermodynamic performance; gained-output-ratio;

1. Introduction

Desalination has been the effective measures to relieve the water resource shortage all over the world.

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Thereinto, the thermal desalination methods are the important constituent parts to produce freshwater [1-3], mainly containing the multi-stage flashing, multi-effect distillation and thermal or mechanical vapor compression. Nevertheless, owing to the complex structures, high energy and cost consumption, low thermal efficiency and inevitable pollution [4, 5], the corresponding desalination units based on the aforementioned methods were restricted. Accordingly, in order to reduce and eliminate the contradiction between the increase demand of desalination and the relevant contained defect, the clean, energy conservation and economic desalination methods are very potential, and the relevant research is urgent and significant [6].

The humidification dehumidification desalination method, which was proposed based on the water cycle in the atmospheric environment, belongs to such promising desalination type, with a high value of GOR [7, 8]. In light of the characteristics contained in the HDH desalination system, renewable energy, especially the solar version, is a good choice to be integrated [9]. A dual heated type of HDH system, in which the seawater and cycled humid air were both heated through solar radiation was proposed by Yildirm [10], and the relevant performance for different operation and design conditions was focused based on the mathematical models with the Runge-Kutta solution. Campos [11] built a mathematical models considering saturated air cycling within the HDH desalination system. Through the established experimental platform, four critical parameters were prescribed to minimize the sum of squared residuals for measured temperature values. After validating the mathematical models, a sensitive analysis from input and project parameters on the water production was executed. It was found that the absorbed heat from the solar heater, the height of the humidifier and the mass flow rate of the seawater were the most important factors to impact the distillate production, while the corresponding influences from the environment temperature is very limited.

Based on the previous literature survey, it is concluded that the driven source of the existing HDH desalination system was always appointed as the solar energy. Nevertheless, such heat source is limited by the weather condition or the industry distribution, and the aforementioned optimization measures were not effective absence of the heat sources. Hence, an unstinted heat source should be integrated into the HDH desalination system to overcome the illustrated defects. Heat pump is just the alternative scheme, which only consumes electricity to power the compressor. Actually, the heat pump usage in the field of desalination was not new [12]. A novel desalination system, which was constituted with a single effect desalination system and a solar assisted driven heat pump, was proposed by Hawlader [13]. Experiments were achieved through the desalination platform at different operation and meteorological conditions in Singapore. According to the analyzed effects from the feed and flashing temperature, the performance ratio as well as the coefficient of performance was evaluated.

It can be inferred that the combination of the heat pump and the HDH desalination unit will be useful to update the relevant desalination performance and remove the restriction from the solar energy or industrial waste heat. However, such innovative combined systems were not involved. In the present paper, the humidification dehumidification desalination system with packed bed dehumidifier is coupled with a mechanical vapor compression heat pump. The seawater is heated by the refrigerant in the condenser while the discharged brine provides the evaporation energy of the refrigerant in the evaporator. Based on the integration mechanisms between the desalination and heat pump unit, the thermodynamic performance at the design conditions is calculated and presented. The research results provide useful references for the design and further optimization of the HDH desalination system.

2. System description

The detailed schematic of the humidification dehumidification desalination system, coupled with a mechanical compression heat pump, is exhibited in Fig. 1. It is observed that the HDH desalination subsystem is constituted with the direct contact humidifier and dehumidifier, while the mechanical compression heat pump subsystem is made up of the compressor, condenser, valve and evaporator. Within the combined desalination system, the released heat of the refrigerant from the condenser is used to raise the seawater temperature while the carried energy in the hot brine is recovered through the evaporator. Obviously, compared to the general HDH desalination system, the discharged brine is further cooled, and the recovered energy is then applied to heat the seawater, with the advantages of the heat pump system exerted.

For the HDH desalination configuration, the feed seawater flows into the recuperator, transferring heat with the exhaust hot water of the dehumidifier. Consequently, after the temperature elevation in the recuperator, the preheated seawater is again heated under the releasing latent heat during the condensing of the refrigerant, which is selected as R22 in the nowaday investigation. Once the top temperature of the seawater is obtained, the hot seawater is then sprayed into the humidification, transferring heat and mass with the cycle air. As a result, both the humidity and temperature of the air is raised, while the hot seawater is cooled and concentrated. The humidified air enters the dehumidifier, and contact direct with the sprayed water. Hence, the hot and saturated humid air is dehumidified while the cold water is heated. Finally, the dehumidified air flows into the humidifier, closing the air thermal cycle, and the obtained hot water is cooled in the recuperator. With respect to the concentrated brine, it is cooled in the evaporator by the refrigerant.

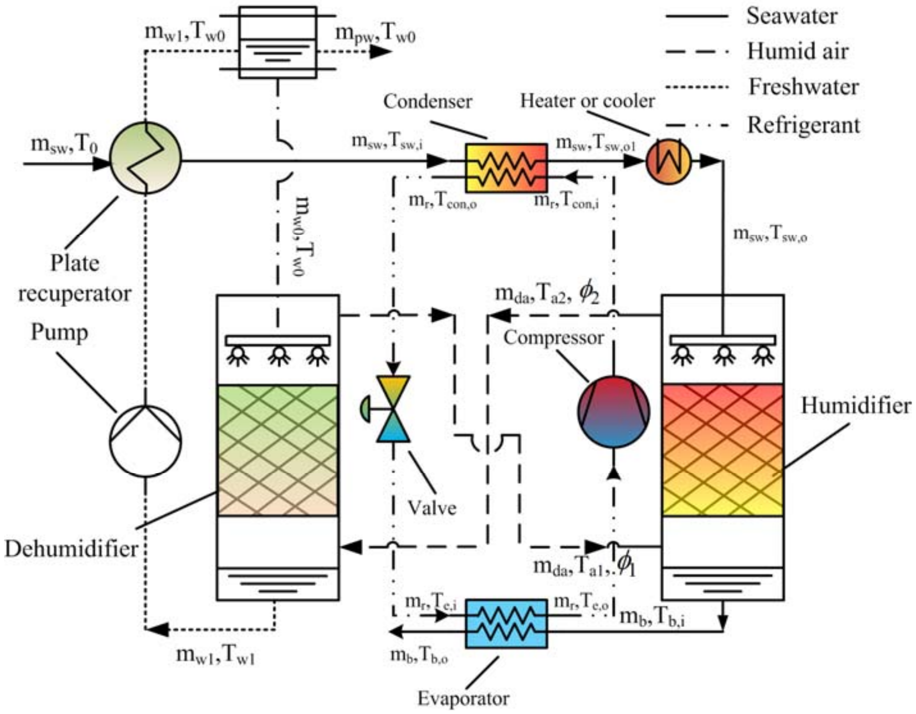

Fig. 1 Schemes of the novel humidification dehumidification desalination system driven by heat pump

Furthermore, at the aspect of the mechanical compression heat pump, the condensed refrigerant enters the throttle valve, and the refrigerant pressure is decreased to the evaporation pressure with an isenthalpic process. After that, the liquid refrigerant changes into vapor in the evaporator, absorbing thermal energy in the brine. Finally, the refrigerant is again pressurized in the compressor, and then it flows into the condenser.

3. Mathematical models of the HDH desalination system coupled with a mechanical vapor compression heat pump

The mathematical model of the HDH desalination subsystem can be found from the expressions by He [14]. In order to improve the thermodynamic performance of the general HDH desalination unit, the mechanical vapor compression heat pump, which is prominent in the energy conservation performance, is chosen and integrated. Thereinto, in the condenser, the preheated seawater is further heated when the refrigerant condenses simultaneously.

As a result, the relevant energy equations can be calculated in Eq. (6):

$$Q_{con} = m_r \cdot (h_{\mathrm{con,i}} - h_{\mathrm{con,o}}) = m_{sw} \cdot (h_{sw,o1} - h_{\mathrm{sw,i}}) \tag{6}$$

The saturated refrigerant liquid is obtained after the condensation. Consequently, in order to maintain the thermal cycle within the heat pump, the liquid refrigerant expands in the throttled valve with an isenthalpic process, and the refrigerant pressure is reduced to the evaporation value with the energy conservation equation exhibited in Eq. (7).

$$h_{e,i} = h_{con,o} \tag{7}$$

In any thermal system, the temperature of the discharged stream should be decreased as much as possible for a high efficiency. As a result, the concentrated hot brine is cooled during the evaporation of the refrigerant, presented in Fig. 2, and the relevant energy conservation equations are given out in Eq. (8).

$$Q_e = m_r \cdot (h_{\mathrm{e,o}} - h_{\mathrm{e,i}}) = m_{sw} \cdot (h_{b,i} - h_{\mathrm{b,o}}) \tag{8}$$

After the acquisition of the refrigerant vapor during the evaporation, the vapor must be pressurized to create the temperature difference between the refrigerant and seawater in the condenser. At the assumption of the adiabatic efficiency with η_c=0.8, the power consumption of the compressor can be obtained based on an isentropic compression.

$$W_c = m_r (h_{c,i,is} - h_{e,o}) / \eta_c \tag{9}$$

For the mechanical vapor compression heat pump, the coefficient of performance (COP) is always applied to measure the energy conversion situation. The specific expression of COP can be given as the beneficial heat load, Qc, divided by the consumed power during the compression, in Eq. (10).

$$COP = \frac{Q_{con}}{W_c} \tag{10}$$

Furthermore, gained-output-ratio is always used to characterize the thermal efficiency within the desalination unit, which is obtained as the ratio between the necessary latent heat for water production and the relevant energy consumption. Nevertheless, before the GOR calculation, the water production must be obtained first, and the performance of the simultaneous heat and mass transfer devices are critical. Consequently, the definition of effectiveness, ε, is extended both to the humidifier and dehumidifier [15, 16].

Accordingly, the thermal parameters within the HDH desalination system can be acquired based on the value of effectiveness. For the current HDH desalination system coupled with the mechanical vapor compression heat pump, the energy input, which is renewable energy or the industrial waste heat in the general HDH desalination system, is partially replaced by the power consumption of the compressor. Furthermore, the extra energy provided by the heater, which indicates the condensation heat is not satisfied to achieve the temperature ascent of the seawater, must be considered during the calculation of GOR. Accordingly, the specific expression can be given out.

$$GOR = \frac{m_{pw} h_{fg}}{W_c + Q_h} \tag{11}$$

Obviously, when the condensation heat is larger or equal to the heat demanded for the seawater temperature difference, the energy consumption come from the compression power completely.

$$GOR = \frac{m_{pw} h_{fg}}{W_c} \tag{12}$$

4. Results and discussion

For the heat pump coupled HDH desalination system, the relevant thermodynamic performance at the design conditions is first simulated, shown in Table 1. Actually, the effectiveness of the heat and mass transfer process, ε_h=0.85 and ε_d=0.85, are fixed, and the terminal temperature difference (TTD) within the evaporator is assumed at TTD_{e}=10K while the value of the pinch temperature difference (PTD) for the condenser is PTD_{c}=5K. With respect to the compression process, the designed pressure ratio is assumed at PR=4. Moreover, the impact laws from the compression pressure ratio, terminal temperature difference of the evaporator and pinch temperature difference of the condenser, on the performance of the HDH desalination system are also calculated and illustrated.

Table 1 Typical thermodynamic parameters of the HDH desalination system coupled with a mechanical vapor compression heat pump

S (g/kg)	ε_h	ε_d	Φ_1	Φ_2	T_0 (K)	PR	PTD_{con}(K)	TTD_e(K)
35	0.85	0.85	1	0.9	288.15	4	5	10

4.1. Thermodynamic analysis of the humidification dehumidification desalination system driven by heat pump

Compared to the combined desalination system, the integration of the heat pump changes the internal energy balance relations, and relevant performance is related both to the HDH desalination unit and mechanical vapor compression heat pump. Hence, the relevant new principles of mass and energy equilibrium within the HDH desalination system should be revealed and elaborated. Water production mainly resulted from the humidification and humidification is calculated and presented in Fig. 2.

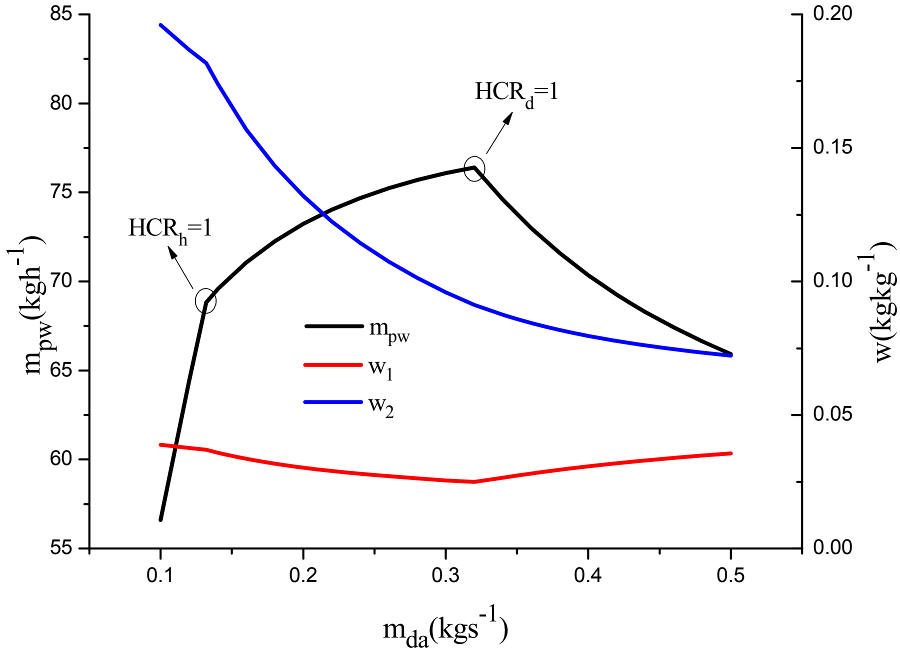

Fig. 2 Water production from the coupled HDH desalination system with the air mass flow rate

It is discovered that within the appointed range of the air mass flow rate, the difference of the humidity ratio drops with an amplitude of 76.71% from Δw=0.16kgkg^{-1} at m_{da}=0.1kgs^{-1} to Δw=0.04kgkg^{-1} at m_{da}=0.5kgs^{-1}. It is found that the above descent mainly results from the declination of the air humidity ratio after the humidification, w_2. Based on the theory of mass transfer [17], it can be inferred that the aforementioned declination of the humidity ratio is attributed to the deterioration of the mass transfer performance due to the elevation of air mass flow rate at the fixed mass flow rate of the sprayed seawater. Ultimately, under the completely reverse trend of the humidity ratio difference and mass flow rate of the humid air, a maximum water production as m_{pw}=76.40kgh^{-1} at the case of m_{da}=0.32kgs^{-1}, which is located in the point with the balance condition of the dehumidification, HCR_d=1.

Actually, the change laws of the water production can also trace back to the performance simulation process. As stated previously, in light of the fixed effectiveness, ε_h=ε_d=0.85, and the final maximum enthalpy difference for the involved working medium, ΔH_{max}, based on the given parameters, the actual total enthalpy difference, ΔH, can be obtained, which is the basic variable to determine all the parameters. It is found that the value of ΔH rises from ΔH=44.57kW to ΔH=54.42kW sharply until the balance condition of the humidifier, HCR_h=1, at m_{da}=0.13kgs^{-1}, and then increases to ΔH=63.44kW at the case of HCR_d=1. Finally, the value of ΔH descends to ΔH=55.17kW. After the acquisition of the total enthalpy difference, it is found that the trend of the water production is entirely consistent with that of the total enthalpy difference, ΔH.

After the water production of the HDH desalination system is gained, the heat pump performance is the next objective to be clarified, shown in Fig. 3. Once a value of 293.15K for the final discharge brine at the outlet of the evaporator, T_{bo}, is assumed, in combination with the terminal temperature difference, TTD_e=10K, it can be determined that the refrigerant will evaporate at the pressure of p_e=0.68MPa. Consequently, in light of the fixed compression pressure ratio, PR=4, the actual enthalpy elevation during the compression process can be calculated for each air condition, and the constant coefficient of performance for the mechanical vapor compression heat pump can be acquired as COP=3.83. Furthermore, it has been illustrated that the matching problem between the latent heat during the condensation and the heat demand for producing the hot sprayed seawater is important for the performance of the whole HDH desalination system. It is found in Fig. 4 that a negative heat load within the extra heat exchanger behind the condenser is always existing, which indicates the refrigerant out of the condenser should be further cooled for obtaining the sprayed seawater. Accordingly, the value of GOR can be calculated on the basis of the water production and the compression power, regardless of the heat load for the extra heat exchanger in Eq. (3). Evidently, owing to the peak value of water production, m_{pw}=76.40kgh^{-1}, and the bottom consumed power of W_c=9.99kW, a most prominent value of GOR as GOR=5.14 is acquired at the case of HCR_d=1.

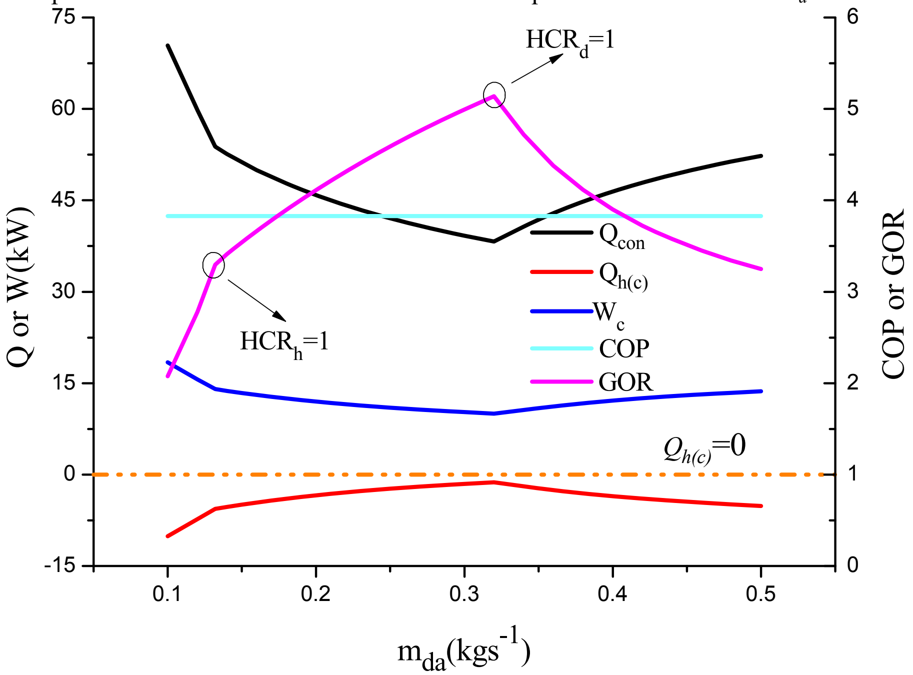

Fig. 3 Thermodynamic performance of the coupled HDH desalination system with the air mass flow rate

4.2. Comparison with the general HDH desalination system

In order to explain the great performance of energy conservation for the current configurations, the comparisons among different types of HDH desalination system are achieved, presented in Fig. 4, with the same effectiveness as ε_h=ε_d =0.9, the feed seawater temperature as T_0=303.15K, and the compression pressure ratio is fixed at PR=3. It is found that the respective peak values of GOR emerge as GOR=3.51, 5.20 and 2.34 for the general air-heated, combined and general water-heated type of HDH desalination system. It can be concluded that the heat pump coupled HDH desalination system has great potential at the aspect of energy conservation.

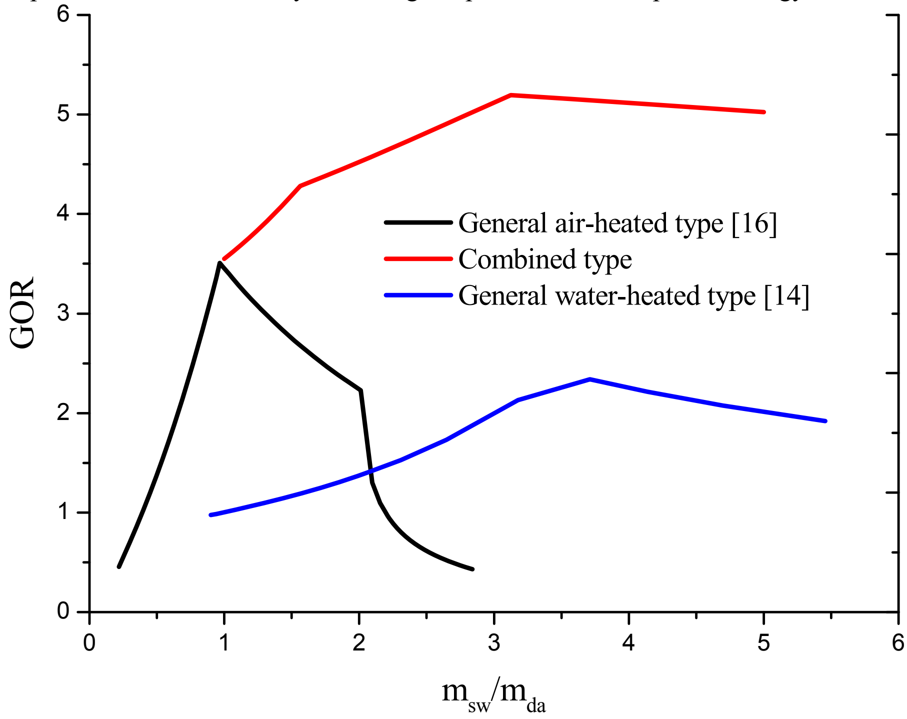

Fig. 4 GOR comparison between the combined and general HDH desalination systems

5. Conclusions

In the present paper, a combined desalination system, which is obtained based on the HDH desalination unit coupled with a mechanical vapor compression heat pump, is proposed and investigated. The corresponding thermodynamic performance is simulated and demonstrated at the design conditions. In light of the above obtained results, the specific conclusions are given as follows:

1. A maximum water production of m_{pw}=76.40kgh^{-1} at the balance conditions of the dehumidifier, HCR_d=1, is obtained due to a product between the humidity ratio difference before and after the humidification process and the air mass flow rate.
2. The value of GOR is raised to GOR=5.14 for the coupled HDH desalination system. Compared to the general HDH desalination system, including the air-heated and water heated type, the relevant elevation magnitude is ΔGOR=1.69 and 2.86 at the same conditions.

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