Investigation of optimal design of direct contact humidification-dehumidification desalination cycle

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Abstract

Ongoing water scarcity around the world caused researchers to investigate renewable based desalination systems for providing fresh water. Direct contact humidification-dehumidification desalination is one of the recently introduced technologies for decentralized small-scale water supply. Major energy source of the cycle is heat that can be provided from any available low-grade heat sources such as waste heat or solar thermal collectors. In this paper, a thermodynamic based analysis implemented to the cycle. And through a theoretically investigation, effect of various operational variables such as bottom and top temperature, flow rates and components effectiveness on the cycle performance parameters has been studied in order to evaluate the cycle performance and address optimal working condition. Understanding optimum working condition of the system can provide outstanding technical information for fabrication and sizing of experimental test rig as well as actual module in advance.

Keywords: humidification dehumidification, HDH, Desalination, Solar energy, GOR

1. Introduction

Providing fresh water is a vital need of human being for living all around the world. Desalination methods are the main technological solution for producing fresh water from saline or impure water. Therefore, various type of desalination methods were proposed and developed that can be categorized in membrane and thermal type.

The first and widely commercialized desalination processes based on evaporation technique are multi-stage flash distillation, multiple effect distillation and vapor compression. The second category of desalination processes uses membrane technologies including reverse osmosis, electro-dialysis and membrane distillation.[1] Humidification-dehumidification (HDH) desalination distillation is a one of recently well-studied technologies among the researchers.

The main features of the HDH cycle including remote area water supply, small scale application makes it a competitive alternative application among other desalination technologies. Availability of solar irradiation and any type of impure water such as seawater or brine can make it a sustainable solution for producing pure water. Therefore, all regions of the world like small islands close to the equator with availability of daily solar irradiance as well as saline water the HDH technology is a considerable option. Moreover, the components of the HDH technology is using available and affordable materials that can be find in market as well as maintenance of the system in not demanding in the terms of skilled labor [2, 3].

Basically, the direct contact (DC) HDH desalination system utilizes two main components for producing distilled water. These are humidifier and dehumidifier. As it shown in Fig. 1 in humidifier saline water after heating up through heat source sprayed over air in order to moisturizing it by increasing the air temperature. Then, the hot moisturized air transferred to dehumidifier in which cold fresh water spraying over it and condensed the vapor content of the humid air by decreasing the temperature of it. The distilled water with fresh cooling water will be collected together at the bottom of the dehumidifier and fresh water will be cooled down through a heat exchanger and sprayed again. It should be note that; humidifier and dehumidifier are heat and mass exchange (HME) devices in which there is simultaneous heat and mass transfer occurrence for these devices. Heated fresh water after dehumidifying process can be cooled down with saline water before it enters to heater. Therefore, the fresh water can be cool down as well as the saline water can be preheated in practical approaches for system designing. However, as matter of simplicity this possibility is not incorporated in this study.

There are different configurations of HDH cycles based on which stream is heated up as well as which stream is open or close. In close air/water cycles, air/water is circulated in a closed loop between humidifier and

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dehumidifier while water/air is open loop. The air in these systems can be circulated by either natural convection or fan forced by adding a blower. For instance, closed-air open-water (CAOW) water-heated cycle is demonstrated in Fig. 1. Therefore, different types of the HDH cycle can be considered such as close-air open-water(CAOW) air-heated cycle, close-water open-air (CWOA) water-heated and close-water open-air cycle (CWOA) air-heated. [2, 4, 5]

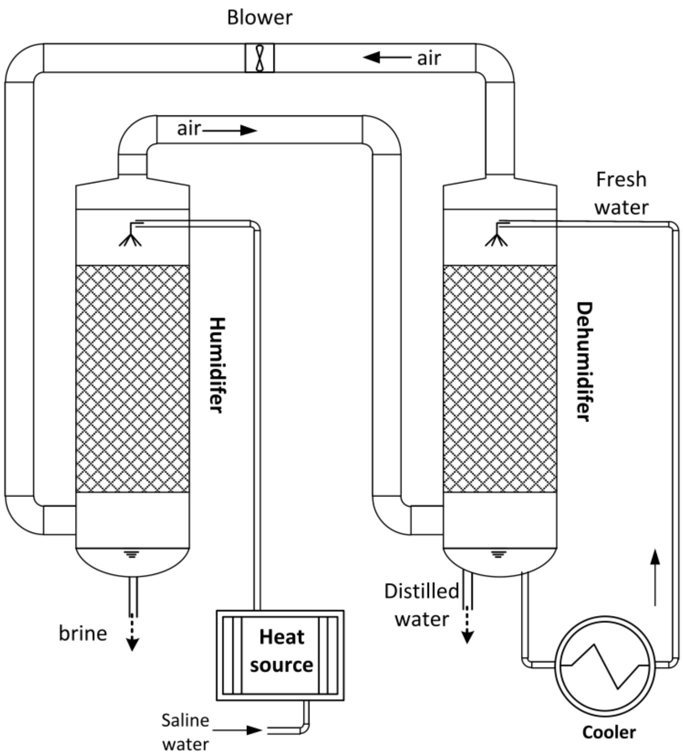

Fig. 1: Overview of the humidification-dehumidification with direct contact dehumidifier

There have been numerous works on non-direct condensers for dehumidification in the HDH cycle.[5-10] Moreover, in this area, several studies have been performed to replace the thermal energy sources of the process from fossil fuel with renewable ones and especially solar energy.[11-13] Also, in other studies, an alternative low-grade heat source such as industrial waste heat or geothermal [14-17]were considered for supplying thermal energy demand of the cycle. These sources of heat can be applied to the HDH cycle with direct contact as well. To make the dehumidifying process in non-direct HDH cycle cost effective, an efficient low cost dehumidification method can be alternated to condense water vapor out of the air stream. With a large fraction of the air/vapor mixture being non-condensable, direct contact condensation is considerably more effective than film condensation. In addition, direct contact condensation within a packed bed is more effective than droplet direct contact condensation.

Bharathan et al. [18] initially introduced a direct contact condenser approach to enhance the heat transfer rate in presence of non-condensable gas. James et al. [19] fabricated a laboratory scale direct contact condenser to study the variation of temperature, humidity, and condensation rate through the condenser system. They evaluated their result by considering a finite volume method for analyzing the packing condenser.

Eslamimanesh and Hatamipour[20, 21] conduct a theoretical analysis for the open-air open-water HDH cycle to study effect of working parameters on water production rate as well as an economic study of the system. Mehrgoo and Amidpour investigated the optimum water production rate utilizing constructal design theory for a fixed-size HDH system.[22] Ettouney [19] introduced different types of the dehumidifier including vapor compression, desiccant air drying, and membrane air drying.

A lack of study for the CAOW water-heated cycle can be observed from the literature review. Therefore, in this study a comprehensive theoretical analysis of a HDH cycle with direct contact dehumidifier performed in order to understand the optimum working condition of the system.

2. Mathematical modeling

In order to study cycle's behavior with variation of operational variables a theoretical study of the HDH cycle from thermodynamic standpoint a theoretical modeling is implemented. As it shown in Fig. 2 the cycle components are considered "black-box" that do not consider transport properties inside the components.

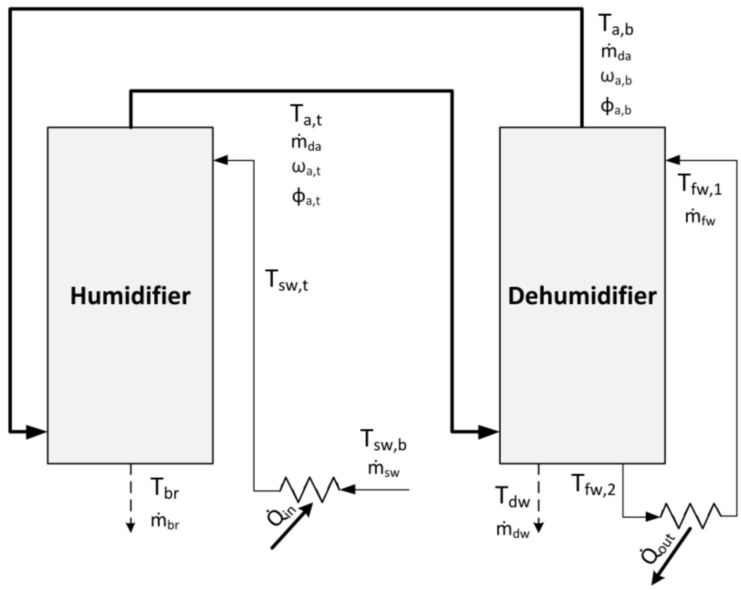

Fig. 2: Schematic diagram of the humidification-dehumidification with direct contact dehumidifier

Following assumptions are considered in theoretical modeling of the direct contact HDH cycle.

- Cycles operate at steady state and steady flow conditions.
- Humidifier and dehumidifier are adiabatic and there is no heat loss from any of the cycle pipelines to their surroundings.
- Pumping and fan powers are negligible in comparison to total thermal energy input of the system.[5, 23]
- Kinetic and potential energy terms are excluded in the energy balance.
- The water condensed in the dehumidifier is assumed to leave at a temperature which is the average of the humid air temperatures at inlet and outlet of the dehumidifier.[5]

The presented model in this study is based on the energy and mass balance implementation on each system components including humidifier, dehumidifier, heater and cooler, so they can be find in thermodynamic text books.[24] However, to solve the governing equation some cycle variables should be known. These parameters are saline water temperature of humidifier inlet, fresh water temperature of the dehumidifier inlet, effectiveness of humidifier and dehumidifier as well as relative humidity of air entering humidifier and dehumidifier. Therefore, for a range of variation of these variables the analysis is performed.

The governing equations can be summarized as following:

Humidifier energy and mass balance:

$$\dot{m}_{sw} + \dot{m}_{da}\omega_{a,b} = \dot{m}_{br} + \dot{m}_{da}\omega_{a,t} \tag{1}$$

$$\dot{m}_{sw}h_{sw,t} + \dot{m}_{da}h_{a,b} = \dot{m}_{br}h_{br} + \dot{m}_{da}h_{a,t} \tag{2}$$

Dehumidifier energy and mass balance:

$$\dot{m}_{da}\omega_{a,t} = \dot{m}_{dw} + \dot{m}_{da}\omega_{a,b} \tag{3}$$

$$\dot{m}_{fw}h_{fw,1} + \dot{m}_{da}h_{a,t} = \dot{m}_{fw}h_{fw,2} + \dot{m}_{da}h_{a,b} + \dot{m}_{dw}h_{dw} \tag{4}$$

It should be noted that, the dry mass flow of air is constant through the humidifier. Also, enthalpy of humid air is considered as a binary mixture of dry air and water vapour, in other words: $h_a = h_{da} + \omega h_v$

Heater and cooler:

$$\dot{Q}_{in} = \dot{m}_{sw}\left(h_{sw,t} - h_{sw,b}\right) \tag{5}$$

$$\dot{Q}_{out} = \dot{m}_{fw}\left(h_{fw,2} - h_{fw,1}\right) \tag{6}$$

In order to discover outlet streams conditions in the humidifier and dehumidifier, effectiveness equation needs to be defined and considered for mathematical solution. Principally, effectiveness compares the actual thermal energy versus ideal thermal energy transferred from each stream and is defined as actual enthalpy rate variation to the maximum possible enthalpy rate variation, in other words $\varepsilon = \Delta\dot{H}/\Delta\dot{H}_{\max}$ [25]. Therefore, effectiveness of humidifier and dehumidifier would be extracted as following:

$$\varepsilon_h = \max\left(\frac{\dot{H}_{a,t} - \dot{H}_{a,b}}{\dot{H}^{ideal}_{a,t} - \dot{H}_{a,b}}, \frac{\dot{H}_{sw,t} - \dot{H}_{br}}{\dot{H}_{sw,t} - \dot{H}^{ideal}_{br}}\right) \tag{7}$$

$$\varepsilon_d = \max\left(\frac{\dot{H}_{a,t} - \dot{H}_{a,b} + \dot{H}_{dw}}{\dot{H}_{a,t} - \dot{H}^{ideal}_{a,b} + \dot{H}_{dw}}, \frac{\dot{H}_{fw,2} - \dot{H}_{fw,1}}{\dot{H}^{ideal}_{fw,2} - \dot{H}_{fw,1}}\right) \tag{8}$$

In both humidifier and dehumidifier, the ideal outlet air enthalpy happens when the outlet air is fully saturated at the water inlet temperature, and the ideal outlet seawater enthalpy is when its temperature is equivalent to the inlet air temperature.

In addition to considering effectiveness equations, relative humidity of air at bottom and top air streams should be known. Nawayseh et al. [11] assumed that exit air from humidification column was saturated according to the experimental. Moreover, distilled water temperature is assumed as average of the humid air temperatures at inlet and outlet of the dehumidifier, in other words $T_{dw} = (\mathrm{T}_{a,t} + \mathrm{T}_{a,b})/2$ as is required that the number of equations and unknowns should be same. [5] Therefore, by solving the system of nonlinear equations, unknown variables can be obtained.

For incorporating accurate and reliable thermodynamic properties of the moist air and water and seawater, ASHRAE handbook[26] as well as Engineering Equation Solver (EES) software[27] are applied. EES software has completer library of thermophysical properties for a wide range of substances including moist air properties with the formulation presented by Hyland and Wexler[28] as well as water properties using the formulation of IAPWS (International Association for Properties of Water and Steam) [29].

The obtained system of nonlinear equations was solved utilizing the EES software, which calculates moist air and water properties using built-in functions. These functions have been previously defined in software and evaluate the thermophysical properties of various substances based on a set database in the software. EES is a numerical solver, using an iterative procedure for solving the system of equations. The EES automatically identifies and groups equations that are solved simultaneously. The convergence of the numerical solution is verified by using two methods: (i) ‘Relative equation residual’ which is the difference between left-hand and right-hand sides of an equation divided by the magnitude of the left-hand side of the equation; and (ii) ‘Change in variables changing in each iteration. The calculations are converged if the relative equation residuals are less than certain value for example 10^{-6} or if variable change is less than 10^{-9}. Both relative equation residuals and change in variables are adjustable for desirable precision. Besides, there are two stopping criteria consisting of (i) ‘number of iteration’ and (ii) ‘elapsed time’ that can be set for obtaining variables with higher accuracy. EES software is widely used by the scientific community for thermodynamic system evaluations for thermodynamic analysis.[30, 31]

3. Cycle metrics

In order to evaluate the HDH cycle performance in the terms of thermal energy recovery, energy efficiency and water production rate, performance parameters of the cycle are defined.[2] they are basically non-dimensional parameters.

Gain output ratio (GOR): GOR is the ratio of the latent heat of evaporation of the distillate water produced to the total heat input to the cycle from the heat source. It represents the amount of heat recovered in the cycle.

$$GOR = \frac{\dot{m}_{dw} h_{fg}}{\dot{Q}_{in}} \tag{9}$$

Recovery ratio (RR): recovery ratio is the amount of distilled water over inlet saline water to the cycle, which is a criterion for water production efficiency of the cycle. It should be noticed that for low recovery ratios, brine disposal treatment is not necessary.

$$RR = \frac{\dot{m}_{dw}}{\dot{m}_{sw}} \times 100 \tag{10}$$

4. Result and discussion

In order to study the effect of cycle's variables on the performance a range of variation is considered. The fresh water temperature in the cycle at the entrance to dehumidifier may range between 15–30 °C due to the seasonal temperature changes. The top brine temperature in the cycle at the humidifier inlet is assumed to be in the range of 65–80 °C which is basically a low-grade heat source temperature. The effectiveness of both the humidifier and dehumidifier are assumed to be within the range of 65–95%. As there is direct contact humidification and dehumidification process, it is expected that air is fully saturated. However, to study the effect of air humidity a relative humidity range of 70–100% is considered. Therefore, under the basic working condition including, fresh water temperature of 20 ºC and top brine temperature of 70 ºC as well as humidifier and dehumidifier effectiveness of 85% and top and bottom relative humidity of 90% simulation are performed.

In order to find effect of mass flow rate of saline water, fresh water and air on the system efficiency, mass flow rate ratio of humidifier is defined. ($mr_h = \dot{m}_{sw} / \dot{m}_{da}$). In Fig. 3 effect of these mass flow rates on GOR is illustrated. As it shown, by increasing both the flow rate of fresh water and flow rate of dry air at a constant saline water flow rate, the GOR of the cycle would be enhanced. In practical approaches the minimum values of mass flow rate ratio of saline water to fresh water based on available heat source and heat sink capacity can be determined. Therefore, here in a case of flow rate of saline water to fresh water equal to 0.5, higher value of GOR can be obtained.

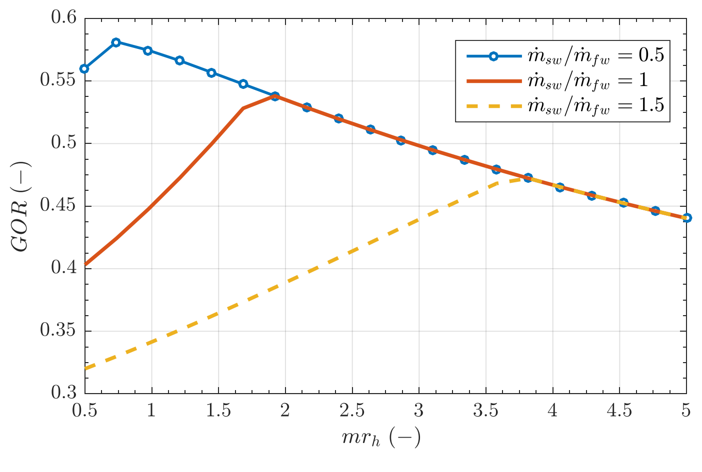

Fig. 3: Effect of mass flow rates saline water, fresh water and air on the cycle GOR

By determining the ratio of the saline water to fresh water, the effect of operational parameters on cycle performance can be studied in detail. Therefore, for flow rate ratio of saline water to fresh water equal to 0.5, impact of operational parameter on cycle performance are investigated in following section.

Fig. 4a demonstrates the GOR against mass rate ratio at fresh water inlet temperatures. As it shown by increasing the humidifier inlet fresh water temperature, both gain output ratio and recovery ratio are decreasing. The maximum value of GOR and RR happening at the same mass flow rate ratio of saline water to dry air. However, for lower temperature of fresh water the optimum happening at smaller mass flow rate ratios. A maximum of 0.62 for GOR can be obtained.

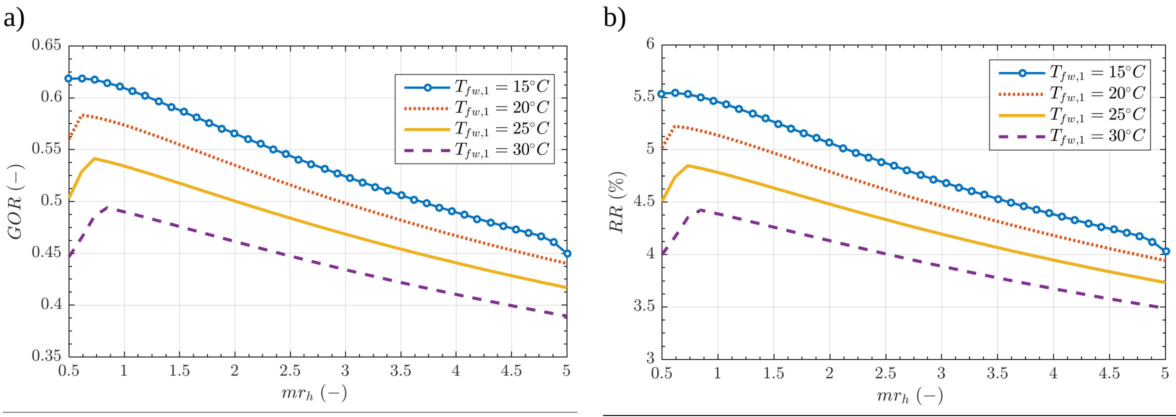

Fig. 4: Effect of inlet fresh water temperatures on gain-output-ratio (a), recovery ratio (b) at different mass flow rate ratios of humidifier

In Fig. 5 the effect of top brine temperature on GOR and RR in depicted. Generally, higher temperature of top brine temperature results is higher GOR and recovery ratio, however; the recovery ratio shows higher values of improvement for higher temperature in comparison to lower temperatures of top brine temperature. Moreover, the optimum value of mass flow rate ratio is slightly rising for higher temperatures of top brine temperature.

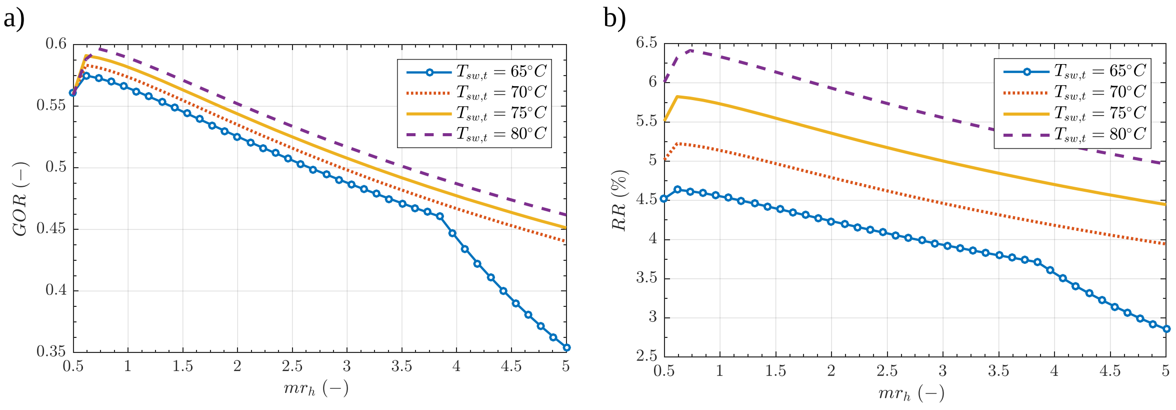

Fig. 5: Effect of top brine temperature on gain-output-ratio (a), recovery ratio (b) at different mass flow rate ratios of humidifier

Relation between the humidifier effectiveness with GOR as well as recovery ratio is shown in Fig. 6. As it displayed, improving the humidifier effectiveness affects the enhancement of both recovery ratio and GOR. Also, optimum point of mass flow rate ratio is almost same for all of the humidifier effectiveness values.

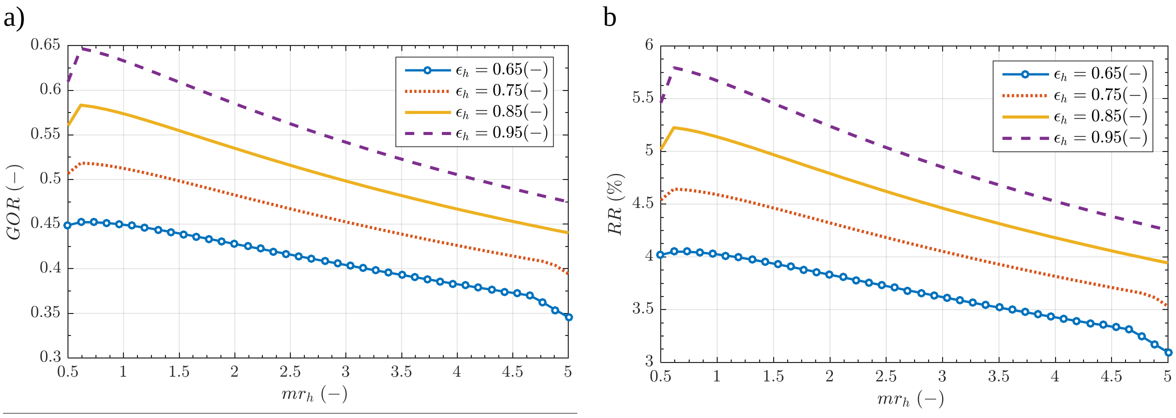

Fig. 6: Relation between the gain-output-ratio (a), recovery ratio (b) and humidifier effectiveness against mass flow rate ratios of saline water to dry air (mr_h)

In Fig. 7, relation of dehumidifier effectiveness with GOR and RR are presented. In general, by increasing the dehumidifier effectiveness both gain-output-ratio and recovery ratio would be enhanced. However, a high effectiveness dehumidifier value would keep wider spectrum of the mass flow rate ratios rather than a low effectiveness values.

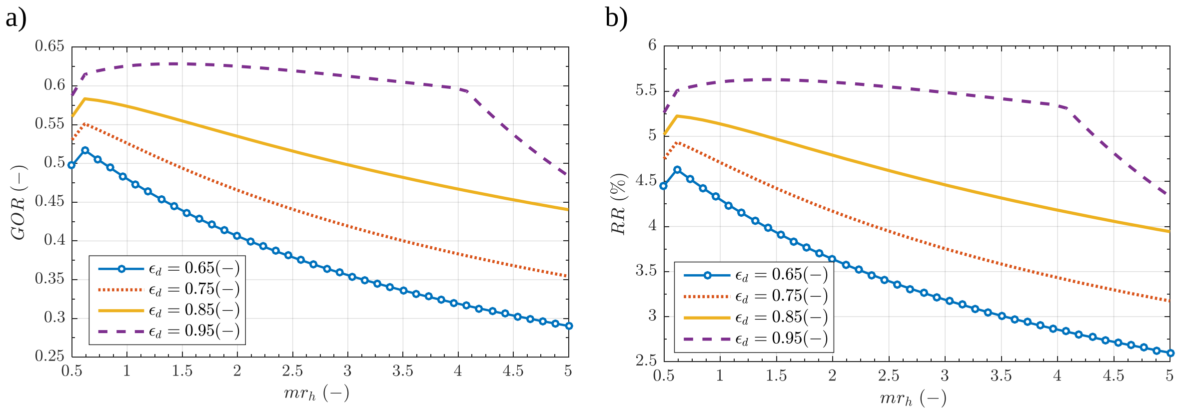

Fig. 7: Relation between the gain-output-ratio (a), recovery ratio (b) and dehumidifier effectiveness against mass flow rate ratios of saline water to dry air (mr_h)

Variation of the GOR and recovery ratio at different mass flow rate ratios is displayed in Fig. 8. As it shown, by intensifying the relative humidity of the air after humidifier outlet, both GOR and RR slightly would be grown. Further, the optimum mass flow rate ratio value shows a slight raise for higher mass flow rate ratios. It should ne mentions that in the experimental study [11] outlet air from humidification column indicated as fully saturated. However, to investigate the effect of air humidity on cycle performance this variable is studied as well.

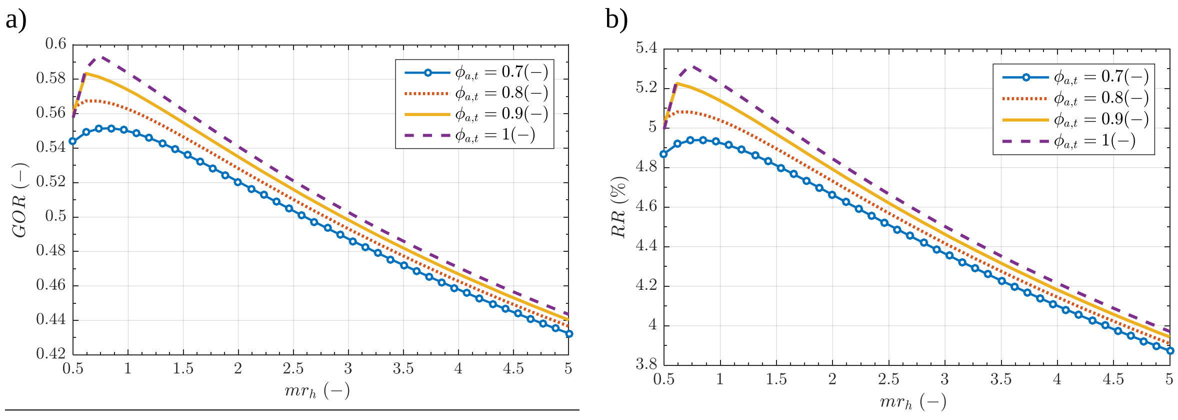

Fig. 8: variation of gain-output-ratio (a), recovery ratio (b) and air humidity after humidifying process against mass flow rate ratios of humidifier

In Fig. 9, variation of gain output ratio and recovery ratio for different values of air humidity are presented. Both GOR and recovery ratio are enhancing by growing the relative humidity. The effect relative humidity after dehumidifying process is pretty similar to the effect of relative humidity after humidification. The reason is that both processes are naturally similar to each other except the working temperature is different.

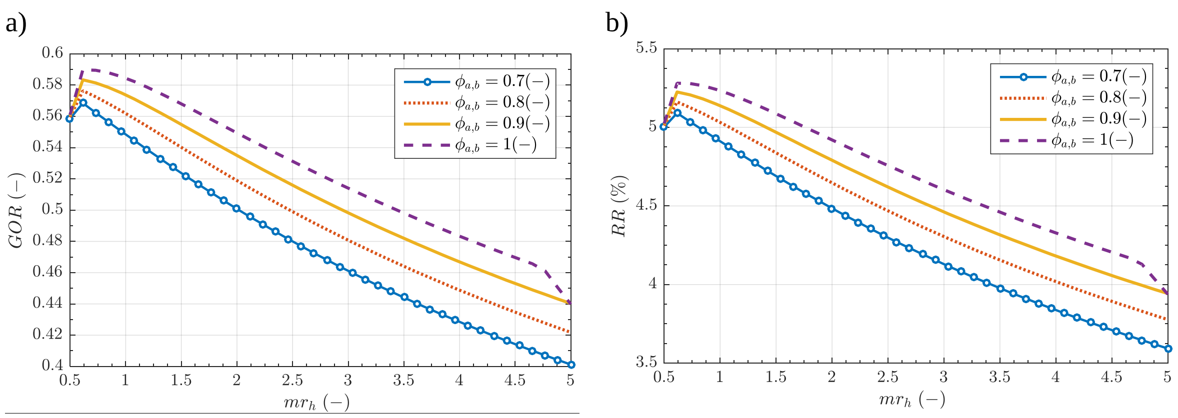

Fig. 9: variation of gain-output-ratio (a), recovery ratio (b), and air humidity after dehumidifying process against mass flow rate ratios of humidifier

5. Conclusion

A parametric study of close-air open-water water-heated HDH cycle with direct contact dehumidifier are performed in order to investigate the optimum working condition of system. To investigate the optimal working condition firstly the mass flow rate of fresh water to saline water is determined. It shown that, by increasing both the flow rate of fresh water and flow rate of dry air at a constant saline water flow rate, the GOR of the cycle would be enhanced. Notice that, in practical design approaches the values of mass flow rate ratio of saline water to fresh water can be determined based on available heating and cooling capacities. Then, at different working condition the optimum mass flow rate ratio of saline water to dry air are discovered. Air flow rate can be adjusted at constant saline water flow rate to reach desirable mass flow rate ratio.

Effect of the top brine temperature on recovery ratio in comparison to its effect on GOR is more considerable as it shows higher values of enhancement. Also, higher top brine temperature is the main variable than can help to improve the recovery ratio of the cycle.

Dehumidifier effectiveness is more important that humidifier effectiveness at high values of effectiveness. Also, the cycle can work on a wider spectrum of mass flow rate ratios with highly effective dehumidifier. As the effect of relative humidity on cycle metrics are small, therefore considering a fully saturated air can be assumed too.

Quantity	Symbol	Unit
Temperature	T	°C
flow rate	$\dot{m}$	kg s^{-1}
heat rate	$\dot{Q}$	kW
enthalpy rate (kW)	$\dot{H}$	kW
Water to air mass flow rate ratio (–)	mr	–
specific enthalpy (kJ/kg)	h	kJ kg^{-1}
Recovery ratio (%)	RR	%
gain output ratio (–)	GOR	–
specific heat capacity at constant pressure	cp	$Jkg^{-1}K^{-1}$
	Greek letters	
absolute humidity of dry air or humidity ratio	ω	$kg_{water}\, kg_{air}^{-1}$
relative humidity	φ	–
effectiveness	ε	–
difference or change	Δ	
	Subscripts	
air	*a*	
dry air	*da*	
distilled water	*dw*	
bottom	*b*	
top	*t*	
Seawater, saline water	*sw*	
maximum	*max*	
middle	*m*	
brine	*br*	
humidifier	*h*	
dehumidifier	*d*	
fw	*fresh water*	
1,2	*states of condition*	
	Acronyms	
closed-air open-water system	CAOW	
Direct contact	DC	
Cold-water open-air	CWOA	
Humidification-dehumidification	HDH	
heat and mass exchanger	HME	

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