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Study of influence of operational parameters on the mass condensate flux in the condenser of seawater greenhouse at Muscat, Oman

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ABSTRACT

The main objective of this theoretical work is to study the influence of the operating parameters such as relative humidity, dry-bulb temperature, seawater temperature, humid air flow rate, and seawater flow rate on fresh water produced by dehumidification of humid air in the condenser of an existing seawater greenhouse that is located in Muscat, Oman. The theoretical results show that the mass condensate rate of the process increases with the augmentation of the relative humidity, the dry-bulb temperature, and the humid air flow rate. However, the seawater temperature and the seawater flow rate have a negative effect. The maximum theoretical mass condensate rate obtained was about 155 kg/h. This result is obtained under the following conditions: for air dry-bulb temperature $T_{db} = 50°C$, relative humidity RH = 97%, inlet seawater temperature $T_{swin} = 28.33°C$, humid air flow rate $\dot{m}_{air} = 23.22$ kg/s, and seawater flow rate $\dot{m}_{sw} = 5.87$ kg/s.

Keywords: Condenser; Dry-bulb temperature; Relative humidity; Seawater temperature; Humid air flow rate; Seawater flow rate; Seawater greenhouse

1. Introduction

Greenhouses are multipurpose agricultural structures that can be used for various tasks. In cold climates, for instance, they are used to provide a warm environment for the cultivated plants inside them. However, in hot climates, they are used to provide a cool environment [1]. The humidification of air followed by dehumidification of water vapor to collect freshwater is not a new concept, but combining these processes in a greenhouse is a recent development. Water production from a solar still is less than the water requirement for a crop grown in an open irrigated field, but may be suitable to supply fresh water to protected cultivation. Solar desalination should be used in combination with water-efficient greenhouse concepts based on a controlled environment [2]. Paton and Davis [3] used the humidification–dehumidification method in a greenhouse type structure for desalination and for crop growth. Their seawater greenhouse produced fresh water and crop cultivation in one unit. It was suitable for arid regions that have seawater nearby. Many crops are very sensitive to the harsh external climatic

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conditions and require a controlled environment that suits their needs.

The climatic limitations that can be overcome by the use of greenhouses are high or low temperatures, intensive solar radiation, high precipitation, and high wind speed [1]. Unprotected outdoor cultivation can demand more than four times the amount of irrigation water, as compared with shaded or more protected cultivation [4]. Arid countries may suffer from lack fresh water but they generally benefit from great solar energy potential. Thus, solar desalination may provide a sustainable solution to supply dry regions with fresh water [5]. The use of greenhouses in arid regions decreases crop water requirements by reducing evapotranspiration. The plastic cover utilized on these structures changes locally the radiation balance by entrapping long-wave radiation and creates barrier to moisture losses. As a result, evapotranspiration is reduced by 60–80% compared to outside the greenhouse [6]. An example of seawater greenhouse system is a pilot plant at Al-Hail, Muscat, in the Sultanate of Oman. The fans alone will not be able to lower the ambient temperature; the air temperature will be slightly higher than the outside temperature in this case [7]. Therefore, the use of some cooling means is required. Evaporative cooling pads (humidifier) can be used to lower the temperature of the air entering the greenhouse through the pads [1]. Applying evaporative cooling reveals that there is a limitation on the relative humidity of the ambient air which depends on the ambient air dry-bulb temperature. Considering a 6 K temperature difference between the moist air and the evaporatively cooled surface seawater, for the heat transfer process in the condenser of an SWGH, the fresh water production starts at ambient air relative humidities below 54% for an ambient air dry-bulb temperature of 45°C [8]. For the evaporation process, the required heat is obtained from the air passing through the humidifier as well as from the water flowing down the humidifier [9]. The degree of achieved cooling depends on many factors including the inlet air temperature, humidity and air flow, the inlet seawater temperature and seawater flow, and the cooling effectiveness of the humidifier.

The high humidity levels might be an undesirable situation in certain instances. In greenhouses, high humidity levels result in the condensation of water vapor on the inner surface of the greenhouse cover [10]. Different techniques to dehumidify the air inside greenhouses have been practiced and studied [1]. The first method lowers the relative humidity by increasing the air temperature, the second method reduces it by removing moisture and increasing air temperature, and the third and fourth methods reduce it by removing moisture from the air.

The dehumidification process takes place in heat exchangers basically by condensing the water vapor carried in the flowing air. For this to happen, the surface temperature of the heat exchanger must be below the dew-point temperature of the moist air [11]. The temperature of the dehumidifier can be lowered to that extent by the use of a coolant fluid that has a temperature less than the dew-point temperature of the air. The dehumidification process is also affected by the conditions of the air. These include the air temperature, humidity, and air flow rate. The process depends on the temperatures of the air and coolant fluid. As the air relative humidity increases, the dew-point temperature approaches the air temperature [12]. This implies that at high humidity levels, a coolant with a temperature slightly lower than the air temperature will be sufficient for the dehumidification to take place. Air flow is a significant variable that influences the dehumidification. The coolant fluid temperature and flow rate are also factors affecting the dehumidification process. The freshwater produced by this seawater greenhouse is less than the irrigation demand of the greenhouse. Therefore, enhancing freshwater production in the seawater greenhouse in Oman is feasible and therefore possible improvements should be studied.

Mathematical simulation of heat and mass processes of the condenser operating of the seawater greenhouse was done by Tahri et al. [13] in the first steps to be undertaken in order to optimize the performance of the seawater greenhouse.

Given the complexity of interaction of operating parameters in the condenser of seawater greenhouse, the calculation method of degrees of freedom allows us to identify all the variables that affect the performance of the installation. This analysis is based on the principles of calculation of the heat and mass balances. We can note that our analysis is very rigorous in the selection of externally controllable variables. Basing on simplifying assumptions, and taking into account the equation of heat balance coupled with mass balance as well as the thermodynamic properties of the mixture (air-water, seawater), the analysis of the sensitivity of the model to all operating variables were highlighted.

Our work is focused on the simulation of installation functioning on the one hand, and the determination of the operating conditions for producing a maximum flow rate of fresh water on the other hand. Furthermore, our model includes all key variables on the system performance. Previous works by other researchers were based on experimental studies without attaching importance to the localization of an optimal operating range. We precise that water needs are highly variable depending on the season and with relation to the humidity conditions.

In this work, we calculate the degrees of freedom of the condenser in order to specify the operating variables which affect its performance. The theoretical study of the condenser using the model developed according to Tahri et al. [13] was used to simulate the influence of the operating parameters such as the dry-bulb temperature, the relative humidity, the seawater temperature, the humid air flow rate, and the seawater flow rate on fresh water produced by desalination in the condenser.

2. Condenser process description

The condenser of the seawater greenhouse (Fig. 1) is a heat exchanger where the seawater is the coolant and the humid air is the hot fluid. The condenser consists of a set of 302 rows of parallel tubes arranged vertically and with an angle of 30° with the direction of flow of humid air (Fig. 2). Each row has 16 identical vertical tubes with a diameter of 33 mm and a height of 1.8 m where the total number of tubes is 4,832. The arrangement of the tubes was organized to ensure the passage of coolant from one tube to another. All tubes passes in a single row have the form of a coil. Seawater enters with a constant flux ($\dot{m}_{sw}$) and a known temperature (T_{swin}) in the first row of each tube and it leaves the last tube in the same row with a temperature T_{swout}. The humid air from the second evaporator runs perpendicular to the condenser. It enters through the tubes with a velocity (v_{air}), temperature (T_{dbin}), and a relative humidity (RH_{in}). This humid air will leave from the condenser keeping the same speed; v_{air}, with a temperature of T_{dbout}, and a relative humidity of RH_{out}. The contact of humid air with the outer cold surfaces of the tubes of the condenser will result in the condensation of the water vapor. The formed condensate, as a liquid film of low thickness, falls along the tubes to be collected in the reservoir of fresh water [13].

Fig. 1. Photo of the condenser in the seawater greenhouse at Al-Hail, Muscat, Oman.

3. Simulation of operational parameters of condenser

The simulation of the condenser leads necessarily to determine:

(1) the temperature profile of the two fluids;
(2) the heat exchange coefficient;
(3) the mass condensate rate.

Depending on various design variables of the condenser and given the complexity of interactions between variables on the one hand and to ease the model on the other hand, we felt more appropriate to use the approach of degrees of freedom of the condenser.

To determine all the design variables which can be controlled externally, we have proceeded by analysis of degrees of freedom of every element present in the condenser based mainly on the Gibbs phase rule on one hand and the heat and mass balances principles on the other hand. Applying these rules, we note that the number of degrees of freedom calculated of the condenser is equal to 5. We can enumerate it as follows:

(1) the relative humidity of inlet air stream;
(2) the temperature of inlet air stream;
(3) the temperature of inlet seawater stream;
(4) flow rate of humid air stream;
(5) flow rate of refrigerant stream (seawater).

However, we can declare that the model developed by Tahri et al. (Fig. 3) reflects very faithfully the dynamic behavior of the condenser of the greenhouse. It is quite obvious that the optimal performance cannot be achieved if the analysis of the influence of the operating parameters is studied.

This system is quite complex to such an extent that it is difficult to clearly distinguish between interaction parameters. For this, the design variables used in the simulation are externally controllable variables in order to analyze their impact on the plant performance.

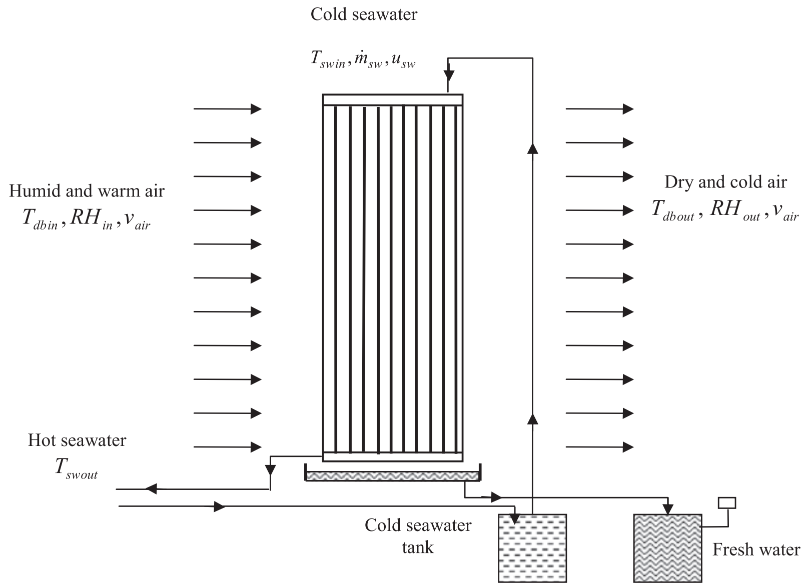

Fig. 2. Schematic diagram of the condenser used in the seawater greenhouse [13].

So, in the next part, we simulate the influence of the real values of operational parameters of the condenser using Fortran 90 software. For the condenser of seawater greenhouse at Muscat, the maximum and the minimum values of different parameters measured *in situ* are given below:

(1) The relative humidity of inlet air stream ($\mathrm{RH_{in}}$) varies from 80 to 98%.
(2) The temperature of inlet air stream (T_{dbin}) varies from 30 to 50℃.
(3) The temperature of inlet seawater stream (T_{swin}) varies from 20 to 30℃.
(4) The mass flow rate of inlet humid air stream ($\dot{m}_{\mathrm{air}}$) varies from 11.74 to 23.48 kg/s.
(5) The mass flow rate of inlet seawater stream ($\dot{m}_{\mathrm{sw}}$) varies from 3.81 to 14.52 kg/s.

The values of inlet relative humidity and inlet dry-bulb temperature of humid air were used in the model for the calculation of the adiabatic saturation temperature (T_{sat}) for the first and the last tubes in the row of the condenser [13] according to Eq. (1):

$$\frac{H_{\mathrm{sat}} - H}{T_{\mathrm{sat}} - T_{\mathrm{db}}} = -\frac{C_{\mathrm{s}}}{h_{\mathrm{fg}}} \tag{1}$$

where H_{sat} is the humidity at the adiabatic saturation temperature (kg water vapor/kg dry air), T_{db} is the dry-bulb temperature (℃), T_{sat} is the adiabatic saturation temperature of air (℃), h_{fg} is the latent heat of vaporization at adiabatic saturation temperature (kJ/kg), H is the humidity of air at the dry-bulb temperature (kg water vapor/kg dry air), and C_{s} is the specific humid heat of the air (kJ/kg ℃).

Firstly, the inlet's dry-bulb temperature (T_{dbin}) was used in the calculation of the pressure of the water vapor $(P_{\mathrm{sat}})_{T_{\mathrm{db}}}$ for each tube of the condenser [13] according to Eq. (2):

$$\log_{10}(P_{\mathrm{sat}})_{T_{\mathrm{db}}} = 8 - \frac{1689.52}{230 + T_{\mathrm{db}}} \tag{2}$$

where $(P_{\mathrm{sat}})_{T_{\mathrm{db}}}$ is the vapor pressure of pure water at the dry-bulb temperature (mmHg), the relative humidity ($\mathrm{RH_{in}}$) was used in the calculation of the partial pressure of water vapor (P_{vap}) in the air–water mixture for each tube [13] according to Eq. (3):

$$P_{\mathrm{vap}} = (P_{\mathrm{sat}})_{T_{\mathrm{db}}}\ \mathrm{RH} \tag{3}$$

where P_{vap} is the partial pressure of water vapor in the air–water mixture (mmHg) and RH is the relative humidity (%).

The value of the temperature of the seawater at the entrance (T_{swin}) was used in the calculation of the exchanged heat flux (Q) through each tube [13] according to Eq. (4):

$$Q = U\ A_{\mathrm{s}}\ \frac{(T_{\mathrm{sat}} - T_{\mathrm{swin}}) - (T_{\mathrm{sat}} - T_{\mathrm{swout}})}{\ln\left(\frac{(T_{\mathrm{sat}} - T_{\mathrm{swin}})}{(T_{\mathrm{sat}} - T_{\mathrm{swout}})}\right)} \tag{4}$$

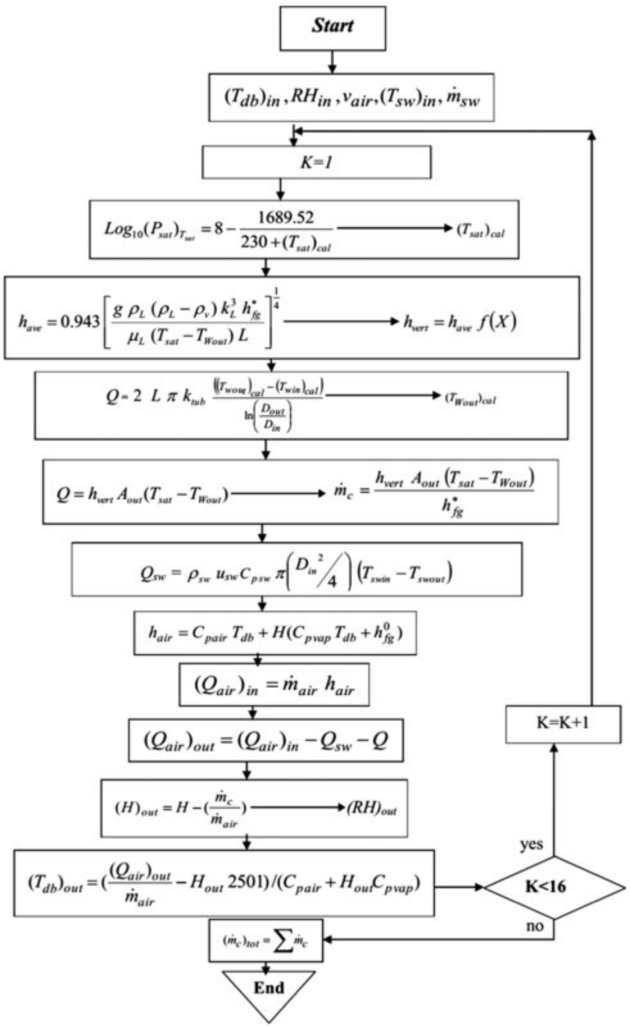

Fig. 3. Flowchart of Tahri et al. model used in the simulation of the condenser.

where U is the overall heat transfer coefficient for each tube (W/m^2 ℃), A_s is the external surface area of the tube (m^2), T_{swin} and T_{swout} are the inlet and the outlet seawater temperatures in each tube of condenser, respectively (℃).

The value of the inlet mass flow rate of seawater ($\dot{m}_{sw}$) was used in the calculation of the heat flux transferred from the humid air to the seawater (Q_{sw}) in the first tube according to Eq. (5):

$$Q_{sw} = \rho_{sw}\, u_{sw}\, C_{P_{sw}}\, \pi\, \frac{(D_{in})^2}{4}(T_{swout} - T_{swin}) \tag{5}$$

where Q_{sw} is the heat flux transferred from the humid air to the seawater in tube (W), C_{Psw} is the specific heat of seawater (J/kg ℃) which is assumed as a constant, ρ_{sw} is the density of seawater (kg/m^3), u_{sw} is the mean velocity of seawater in the tube which was assumed as a constant through each tube (m/s), and D_{in} is the inner diameter of tube (m).

We note that, the mass flow rate of seawater is calculated according to Eq. (6):

$$\dot{m}_{sw} = \rho_{sw}\, u_{sw}\, \pi\, \frac{(D_{in})^2}{4} \tag{6}$$

The inlet mass flow rate of the humid air ($\dot{m}_{air}$) was used in the calculation of the heat flux of air (Q_{air}) in the entrance of the first tube according to Eq. (7):

$$(Q_{air})_{in} = (h_{air})_{in}\, \dot{m}_{air} \tag{7}$$

where Q_{air} is the heat flux of air (W), $\dot{m}_{air}$ is the mass flow rate of air (kg/s), and h_{air} is the enthalpy of air (J/kg dry air).

At $T = 0$℃, the water liquid is saturated. For this reference state, the enthalpy of humid air was calculated according to Eq. (8):

$$h_{air} = C_{pair}\, T + H\, [h_{fg}^0 + C_{pvap}\, T] \tag{8}$$

4. Results and discussions

4.1. Influence of inlet relative humidity of air

The influence of the inlet relative humidity on the fresh water produced by dehumidification in the condenser of seawater greenhouse at Muscat, Oman, is shown in Fig. 4. It can be noted that the simulation results give an increasing trend of the curve $\dot{m}_c = f(\mathrm{RH})$. Indeed, increasing the inlet relative humidity of air leads to a large heat exchange in the condenser. This heat exchange of latent heat is counterbalanced by the activation of condensation phenomenon which leads to higher condensate flow rate. The theoretical analysis of the relationship between $\dot{m}_c$ and RH in the flowchart corroborates these results. It can be noted that the rate of increasing of condensate flow rate is $\frac{\Delta\, \dot{m}_c}{\Delta\, \mathrm{RH}} = 1.285$ kg/h %.

4.2. Influence of inlet dry-bulb temperature of air

Fig. 5 shows the influence of the inlet dry-bulb temperature of air on the fresh water produced in the condenser. For a variation of air temperature from 34 to 50℃, the results of simulation of condenser functioning at atmospheric pressure show that the

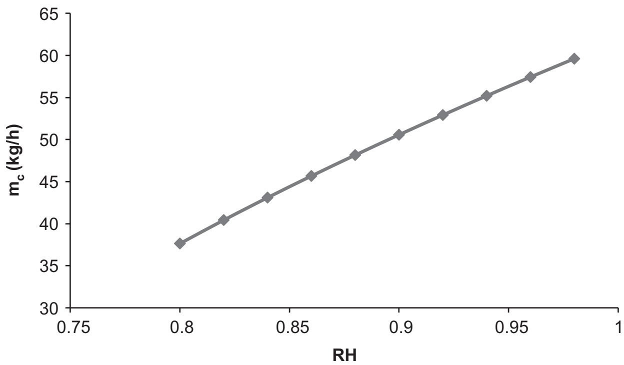

Fig. 4. Influence of relative humidity on fresh water produced by the condenser for T_{db} = 34.6℃, T_{swin} = 28.33℃, $\dot{m}_{air}$ = 23.22 kg/s, and $\dot{m}_{sw}$ = 5.87 kg/s.

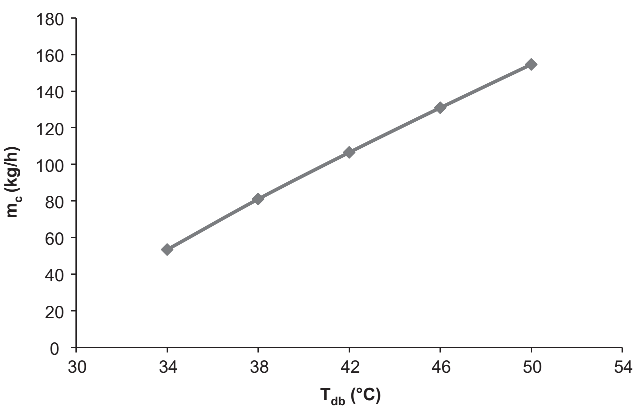

Fig. 5. Influence of dry-bulb temperature on fresh water produced by the condenser for RH = 0.97, T_{swin} = 28.33℃, $\dot{m}_{textair}$ = 23.22 kg/s, and $\dot{m}_{sw}$ = 5.87 kg/s.

flow rate of condensate is strongly linked to ambient air conditions. It is evident from the figure that when the inlet dry-bulb temperature of air increases, the productivity of fresh water is increased. For air temperatures rising and its constant humidity, we note that the saturation conditions remain the major obstacle to the condensation. Therefore, it is possible to improve the operating condensation by recycling the air in the greenhouse to reach saturation conditions. It can be noted that the rate of increasing of condensate flow rate is $\frac{\Delta \dot{m}_c}{\Delta T_{db}} = 6.371\text{kg/h ℃}$.

4.3. Influence of inlet seawater temperature

The influence of the inlet seawater temperature on the fresh water produced by dehumidification in the condenser is shown in Fig. 6. For condensation conditions dictated by the vapor pressure of the gaseous phase, any increase in T_{swin} results in the displacement of gap temperature to the unfavorable areas. This decreasing linear tendency is observed when the inlet temperature of the cold stream T_{swin} increases. The outlet temperature T_{swout} consequently increases following a linear behavior with a negative impact on the mass flow rate of condensate. An average temperature of 25℃ seems to give flow rate trampling the optimal performance of the plant. We note that the regression rate of condensate flow rate is $\frac{\Delta \dot{m}_c}{\Delta T_{swin}} = -0.35$ kg/h ℃.

4.4. Influence of inlet humid air mass flow rate

Fig. 7 shows the influence of the inlet humid air mass flow rate on the fresh water produced in the condenser. Any increase in mass flow rate of humid air leads to an augmentation of its enthalpy. Indeed, any increase in velocity of the humid air is reverberate by an intensification of heat exchange taking account from the impact of turbulence on the one hand, and reduces the film thickness of condensate to promote the mass transfer of vapor to the tube wall on the other hand. We note that the rate of increasing of condensate flow rate is $\frac{\Delta \dot{m}_c}{\Delta \dot{m}_{air}} = 0.6$.

4.5. Influence of inlet seawater mass flow rate

The influence of the inlet seawater mass flow rate on the fresh water produced by dehumidification in the condenser is shown in Fig. 8. It can be noted that the simulation results give a decreasing trend of the curve $\dot{m}_c = f(\dot{m}_{sw})$. It is clear that this correlative dependence is similar to a falling shape and the weak cooling flow rates lead to considerable cooling, while increasing of the condensate flow rate consequently.

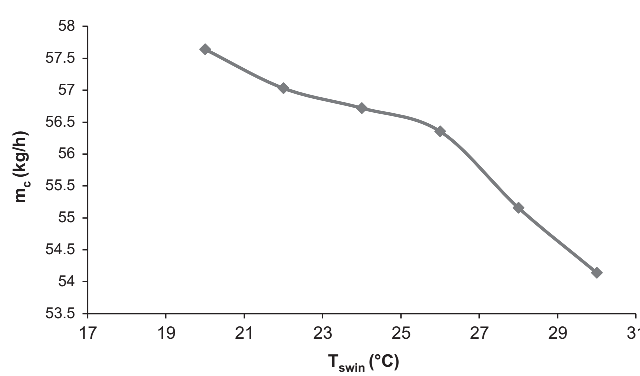

Fig. 6. Influence of seawater temperature on fresh water produced by the condenser for RH = 0.97, T_{db} = 34.6℃, $\dot{m}_{air}$ = 23.22 kg/s, and $\dot{m}_{sw}$ = 5.87 kg/s.

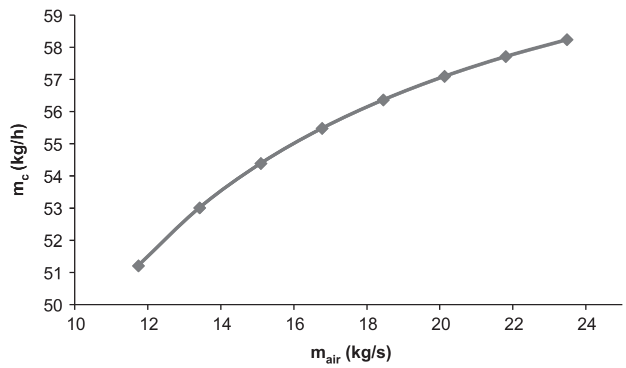

Fig. 7. Influence of air flow rate on fresh water produced by the condenser for RH = 0.97, T_{db} = 34.6˚C, T_{swin} = 28.33˚C, and $\dot{m}_{sw}$ = 5.87 kg/s.

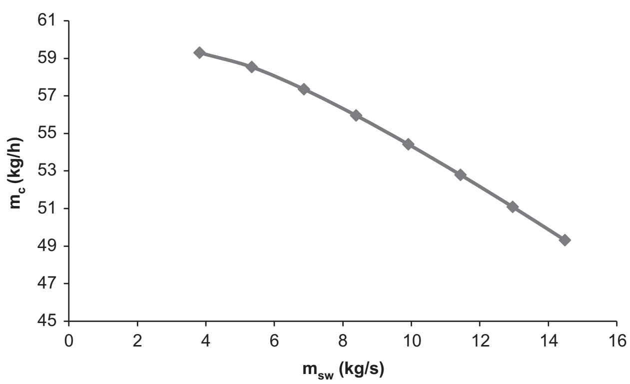

Fig. 8. Influence of seawater flow rate on fresh water produced by the condenser for RH = 0.97, T_{db} = 34.6˚C, T_{swin} = 28.33˚C, and $\dot{m}_{air}$ = 23.22 kg/s.

This situation offers better conditions for condensation. It can be noted that the regression rate of condensate flow rate is $\frac{\Delta \dot{m}_c}{\Delta \dot{m}_{sw}} = -0.935$.

5. Conclusion

The results presented in this paper are a calculation of the operational variables using the degrees of freedom principle for the condenser which is the main unit in the plant. We discussed the influence of the operating parameters such as the dry-bulb temperature of humid air, the relative humidity of air, the seawater temperature, the humid air flow rate, and the seawater flow rate on fresh water produced by desalination in the condenser of seawater greenhouse by simulation. The results showed the sensitivity of flow rate of condensate to these parameters. Specially; it can be noted that the condensate flow rate increases with relative humidity of air, temperature of air entering, and humid air flow rate. But the temperature and the flow rate of seawater affect the flow rate of condensate negatively.

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Nomenclature

A_s	—	heat transfer area
C_p	—	specific heat
C_s	—	humid specific heat
D	—	diameter of tube
H	—	absolute humidity
h	—	enthalpy
h_{fg}	—	latent heat of vaporization
$\dot{m}$	—	mass flux
P	—	pressure
Q	—	heat flux
RH	—	relative humidity
T	—	temperature
U	—	overall heat transfer coefficient
u	—	velocity
Greek symbols		
ρ	—	density
Subscripts		
air	—	air
c	—	condensate
db	—	dry-bulb
in	—	inner
sat	—	saturation
sw	—	seawater
swin	—	seawater inlet
swout	—	seawater outlet
vap	—	vapor

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