Desalination and Water Treatment
www.deswater.com

19 (2010) 294–300
July

doi: 10.5004/dwt.2010.1487

Numerical analysis of solar desalination using humidification–dehumidification cycle

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Received 6 August 2009; Accepted in revised form 29 December 2009

ABSTRACT

This paper has studied a desalination unit with humidification–dehumidification cycle which uses solar energy as its source of heating (SDHD). For thermodynamic analysis, mass and energy balance equations have been written for the humidifier, condenser and other cycle components. The resulted nonlinear equations have been solved simultaneously to study and analyze the effects of cycle parameters on the amount of desalinated water produced by the plant. Therefore the effects of brackish inlet water flow rate, its temperature, solar collector area, condenser characteristic, humidifier characteristic and inlet air conditions on the rate of fresh water production have been investigated and discussed.

Keywords: Solar desalination; Humidification–dehumidification cycle; Thermodynamic analysis; Simulation

1. Introduction

Fresh water is one of the main concerns in the new century. Population grows fast and potable water resources decrease. In the other hand energy crises would also be another issue that must be well addressed by the politicians and also scientists. Developing desalination plant with using renewable energy (particularly solar energy) is one of the important options to overcome the mentioned concerns. Particularly needs to plants that could respond to this request for the remote regions.

Solar desalination unit functioning by humidification and dehumidification (HD) can be utilized as an effective and promising technology for producing desalinated water in remote areas. This process is, mainly, based on the ability of air to be saturated with water. In addition it would be very efficient at the regions for which the relative humidity of air is significant such as the coast of the Persian Gulf and the Oman Sea in the south of Iran.

Many studies have been carried out about various types of HD cycle desalinations [1–6]. These studies have investigated different ways of increasing the production of desalinated water and performance of plant. Goosen et al. [7] with the aid of the HD process, examined some economic and thermodynamic aspects of solar desalination. Their report was based on the fact that commercial production of solar desalination is economically and efficiently advantageous. Parekh et al. [2] carried out a comprehensive investigation on the background of solar desalination using humidification–dehumidification (SDHD) systems. They looked at the development of solar stills historically, and concluded that the main factor of the development of these systems was frequent use of the latent heat of condensation; summarizing the results from prominent technologies of past SDHD units, they concluded that most of the researchers had indicated that

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the effect of inlet air flow rate on the cycle was insignificant. However, the effect of feed water flow rate on the efficiency of a SDHD unit was described as significant.

Al Hallaj et al. [6] undertook an experimental study on an SDHD unit. In their unit the air circulates by natural or forced convection and is humidified by the constant rate of water obtained either from a collector (indoor type) or from an electrical heater (outdoor type). Their results in indoor and outdoor conditions showed factors of performance and daily production of desalinated water. In outdoor conditions, their results showed higher fresh water production compared to that of solar stills, whereas the effect of air velocity was formerly regarded only in lower performance temperatures.

Nafey et al. [8] carried out an experimental work on the SDHD process. Their plant consisted of humidification and dehumidification towers, which were located next to flat-plate solar collectors (for air heating) and water concentrator (for heating water). They found that the effect of air velocity was insignificant, while great influence of inlet water and air temperatures on the production of desalinated water was observed. They predicted the fresh water production numerically. Their experimental and theoretical findings are in good agreement [9]. Aybar and Assefi [10] reviewed and compared direct and indirect solar distillation system. Ziqian et al. [11] have studied a solar heating system on an absorption solar desalination unit. They showed that the unit possessed more advantages when it operated at a higher temperature and lower pressure.

Multi-effect humidification–dehumidification (MEHD) is another interesting plant that was studied by Chafik [3] and Ben Mahmud et al. [12]. This technique includes humidification and air heating in several stages which leads to an increase in the moisture density in air flow. Hou et al. [13] mentioned that in most of the previous studies regarding the SDHD technology, obtaining optimal conditions of design, was a difficult and complicated procedure. Using Pinch method, they proposed a design for optimizing the performance of the SDHD process. Their results show that as the temperatures of the sprayed (humidifier tower) and cooling water (in condenser) are known, there is an optimal rate of flow for the ratio of water to dry air. Recently Zhani et al. [14] studied water desalination working with the HD method using solar energy. They recommended using a cylindrical evaporation tower and a condensation tower than parallelepiped geometry. They also advised to integrate a storage tank to ensure a continuous use of the unit.

The present work intends to investigate the effects of different parameters on the fresh water production by a humidification–dehumidification plant to find the suitable conditions of both flow characteristics and also plant characteristics (humidifier, dehumidifier and solar collector). Therefore the effects of brackish inlet water flow rate, its temperature, solar collector area, condenser characteristic, humidifier characteristic and inlet air conditions on the rate of fresh water production have been presented and discussed.

2. Plant description

The considered plant consists of many parts including air and water solar collectors, condenser and humidifier. Fig. 1 illustrates a schema of this cycle. Feed water (brackish water, turbid water, flowing water with high heaviness and seawater) flows into the condenser and then goes through a water solar collector. Hot water then sprays through the humidifier for saturating the air coming in from the other side and finally brain water goes out of the cycle. In the other side, air after becoming hot at the solar heating section, flows through the humidifier to absorb fresh water and become saturated. It then passes through the condenser to be cold and fresh water to be distilled.

3. Mathematical formulation

Energy and mass balance equations have been considered for all parts of the cycle. A few assumptions that were believed not have a significant effect on the analysis have been made for simplicity of calculation. These are as follows:

1. The process is assumed to be in steady state condition.
2. Heat loss is neglected.

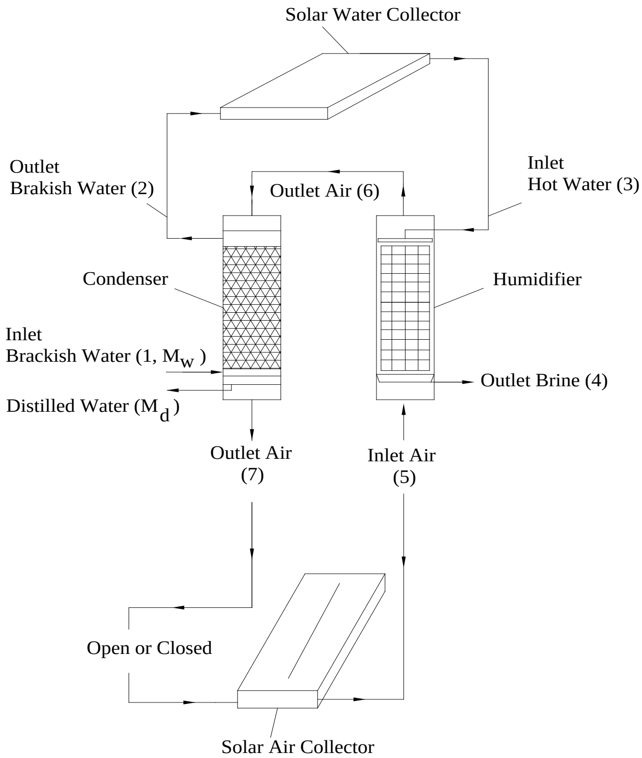

Fig. 1. Schematic of the plant.

3. Since the operating pressure is close to the atmospheric pressure, air and water vapor are assumed to behave as ideal gas.
4. Saturated air is at the exit of the humidifier and also at the exit of the condenser.
5. Kinetic and potential energy changes are neglected.

Accordingly, mass and energy balance equations in the humidifier (Fig. 2) are defined as follows:

$$\dot{m}_a h_{a5} + \dot{m}_{v5} h_{v5} + \dot{m}_{w3} h_{f3} = \dot{m}_a h_{a6} + \dot{m}_{v6} h_{v6} + \dot{m}_{b4} h_{f4} \quad (1)$$

$$\dot{m}_{v6} + \dot{m}_{b4} = \dot{m}_{v5} + \dot{m}_{w3} \quad (2)$$

$$M_5(h_6 - h_5) = Ka\,V\left[\frac{(h_3 - h_6) - (h_4 - h_5)}{\ln\dfrac{h_3 - h_6}{h_4 - h_5}}\right] \quad (3)$$

In the above equation KaV, the humidifier characteristic, could be determined by the following imperial equation [15]:

$$\frac{KaV}{M_w} = 0.07 + A.N\left(\frac{M_w}{M_5}\right)^{-n} \quad (4)$$

where A and n are constants and are given experimentally for different fill matrix. In this simulation A and n are considered to be 0.07 and 0.68 respectively. N is the number of decks and is considered 24 [16]. Humidity ratio is characterized as a function of atmospheric pressure, steam partial pressure and dry bulb temperature [17].

$$w_n = \frac{m_{vn}}{m_a} = 0.622\frac{P_{vn}}{P - P_{vn}} \quad (5)$$

Relative humidity is also defined as follows:

$$\Phi_n = \frac{P_{vn}}{P_{gn}} \quad (6)$$

The energy and mass balance equations for the condenser which is shown in Fig. 3 are defined as:

$$\dot{m}_a h_{a6} + \dot{m}_{v6} h_{v6} + \dot{m}_{w1} h_{f1} = \dot{m}_a h_{a7} + \dot{m}_{v7} h_{v7} + \dot{m}_d h_{f7} + \dot{m}_{w2} h_{f2} \quad (7)$$

$$M_d = M_a(W_6 - W_7) \quad (8)$$

$$M_w cp_w (T_2 - T_1) = A_{\mathrm{cond}} U_{\mathrm{cond}} \mathrm{LMTD} \quad (9)$$

LMTD is condenser's logarithmic average temperature difference which is described by:

$$\mathrm{LMTD} = \frac{(T_6 - T_2) - (T_7 - T_1)}{\ln\dfrac{(T_6 - T_2)}{(T_7 - T_1)}} \quad (10)$$

Enthalpy and humidity ratio for saturation condition can be obtained from the following relationship [14].

$$H = 0.00585T^3 - 0.497T^2 + 19.87T - 207.61 \quad (11)$$

$$W = 2.19T^3(10^{-6}) - 1.85T^2(10^{-4}) + 7.06T(10^{-3}) - 0.077 \quad (12)$$

Heating input energy at the solar collector is calculated by:

$$Q_u = F_R\,A\left[I\ \tau_\alpha - U_L(T_i - T_a)\right] \quad (13)$$

Details of solar collector formulation would be found in [18]. The above nonlinear equations have been solved simultaneously to find thermodynamic condition at

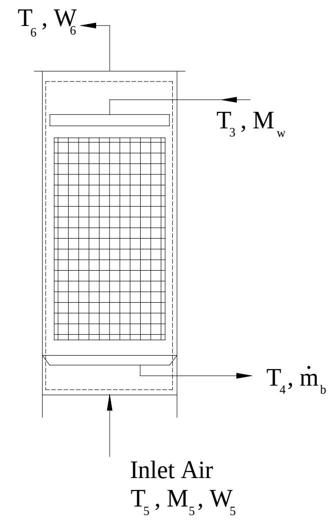

Fig. 2. Humidifier.

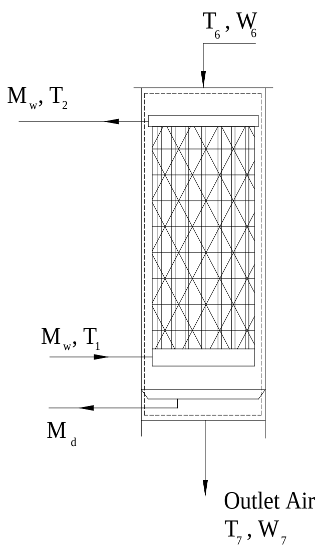

Fig. 3. Condenser (dehumidifier).

each part of the cycle and to find the rate of fresh water production.

4. Validation and results

In order to validate the present mathematical model comparison with the results of Orfi et al. [19] has been done. However, all necessary details of their work for comparison have not been found. As seen in Fig. 4, the trend of the results is similar and the difference is believed to arise from the difference on some of the data in these two works which are missing in the work of Orfi et al. [19] and they were not exactly the same as [19] in this work. A comparison is also made with the results of Nawayseh et al. [17] which is shown in Table 1. As it is shown, the concordance between the results is good. Thus the mathematical model is reliable and could be used to study the plant.

In this study not only the amount of water production is increased but also the cost of installation is important to know and to decrease as much as possible. Therefore in addition to the study of the effect of some flow parameters on the potable water production, the effects of plant components on the fresh water production are also investigated. Thus the effects of flow rates of air and water in the cycles, inlet water and air temperatures, packing height of the humidifier, the heat and mass transfer area in the condenser and humidifier and collector's surface have been analyzed in relation to the amount of desalinated water production.

Fig. 5 shows the effect of inlet brackish water flow rate at different temperatures on the fresh water production. In general, for the given conditions, increasing the inlet brackish water augments the fresh water production up to the situation for which the exit air from the humidifier becomes saturated (ϕ_6 = 100%). Then by increasing Mw, the operating condition of solar water heater does not change and the temperature of the brackish water at the inlet of the humidifier decreases. The latter has a negative effect on the amount of water needed for saturating air at the other side and consequently fresh water production decreases. In addition, it is also seen that decreasing the brackish inlet temperature, significantly augments the fresh water production as result of increasing the performance of the condenser.

Table 1
Comparison of the present simulation with the results of [17]

	Nawayseh et al. [17]	Present work
Q,(kW	1.4	1.4
T_1, °C	25	25
T_5, °C	35	35
Production, kg/h	1.31	1.27

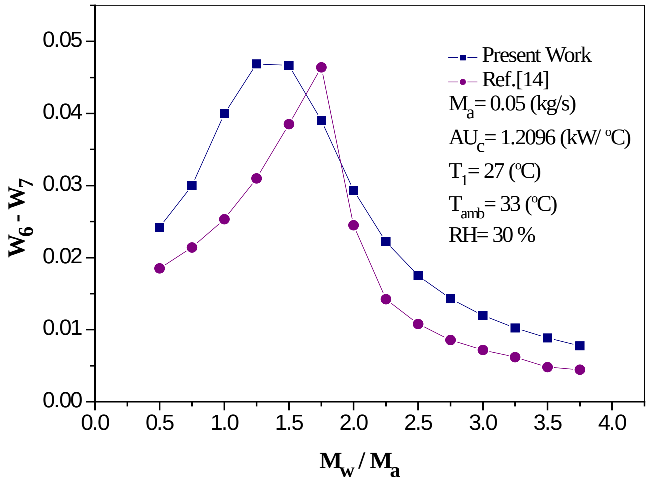

Fig. 4. Qualitative comparison of the present result with the results of [19].

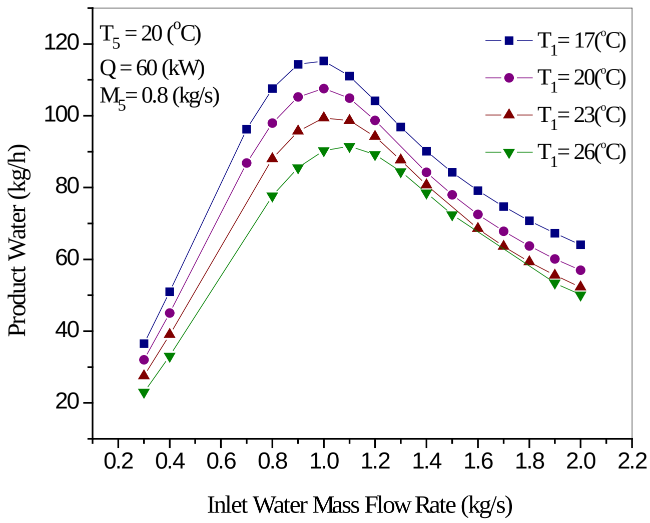

Fig. 5. Effect of brackish water flow rate on the fresh water production at different inlet water temperatures.

The effect of inlet air mass flow rate and its temperature on the fresh water production is presented in Fig. 6. An optimum air flow rate for given conditions is seen. This figure shows that the air mass flow rate has a significant effect on the fresh water production. Increasing the inlet mass flow rate higher that the optimum point causes to air at the exit of humidifier could not be completely saturated and consequently fresh water production decreases. As it is expected, increasing the air inlet temperature augments distilled water production significantly because more water could be mixed with the air to become saturate.

For the given inlet air and brackish water mass flow rate, the effects of condenser characteristic ($A_{cond}U_{cond}$) on

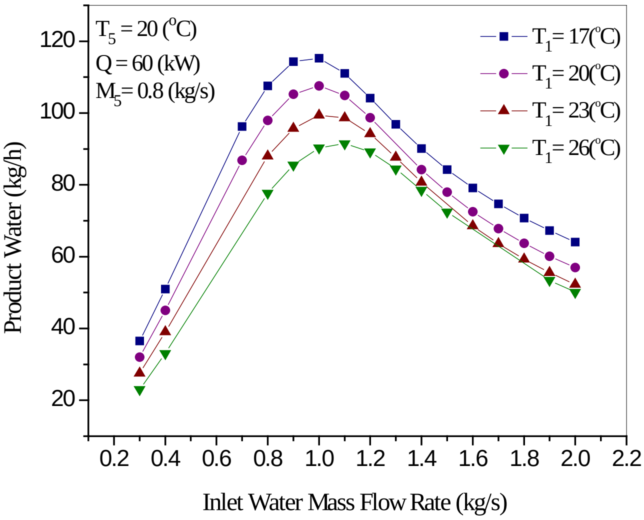

Fig. 6. Effect of inlet air flow rate on the fresh water production at different inlet air temperatures.

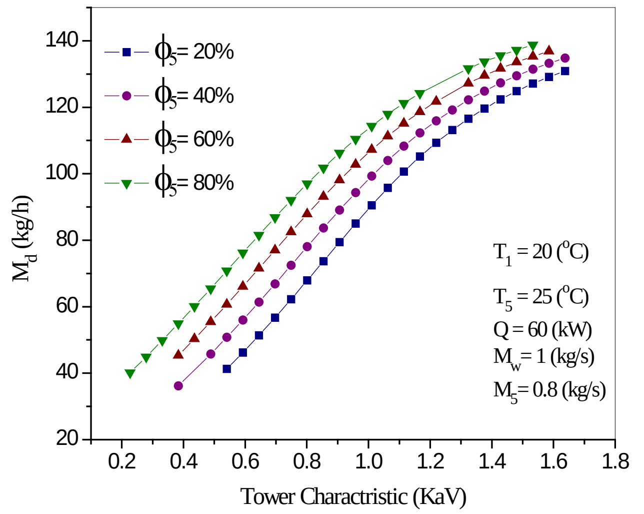

Fig. 8. Effect of humidifier characteristics (KaV) on the fresh water production.

the fresh water production are shown in Fig. 7 for different brackish water inlet temperature. As seen, fresh water production does not significantly change from a certain value of $A_{cond}U_{cond}$ (≈30 at these conditions). This means that an optimum condenser characteristic exists which should be considered by designers. Again it is seen that increasing the brackish water temperature augments the rate of fresh water production. Humidifier characteristics are presented by the KaV parameter for a given water flow rate.

The effect of KaV parameter for different air inlet temperatures on the fresh water production is shown in Fig. 8. As seen, the amount of distilled water is not significantly increased by selecting KaV larger than 1.5. It shows that the fresh water production increases steeply

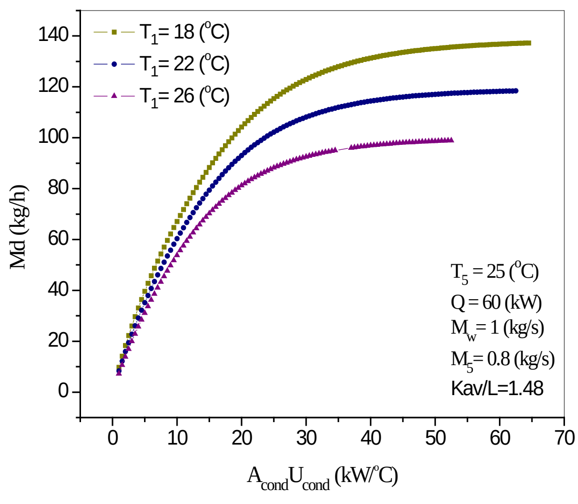

Fig. 7. Effect of condenser characteristics (AU) on the fresh water production.

with augmenting KaV because the relative humidity of air at the humidifier exit increases rapidly. After that air relative humidity approaches to 100% and the rate of mass transfer decreases. Thus a minimum value for the KaV exists to be also economically efficient. It is also shown that for given KaV inlet water with higher temperature could produce more fresh water.

The effect of humidifier height for a constant humidifier cross section on the fresh water production is shown in Fig. 9. Using a humidifier with a higher height could increase the air relative humidity at the humidifier exit. The latter could increase the fresh water production. But the rate of increasing changes at a certain level (≈ 120 kg/h), since the relative humidity of air at the humidifier exit does not significantly vary for that flow and geometrical conditions. It is also shown that if the relative humidity of the inlet air augments smaller humidifier is needed for a particular fresh water production.

Fig. 10 shows the necessary solar collector area for different inlet air temperatures to produce a certain amount of fresh water. It is seen that the rate of increasing the necessary solar collector area increases more rapidly than the fresh water production when distilled water production is wished to be higher than about 120 kg/h which means that economically it would not be efficient.

For designing such a plant for particular inlet conditions for the air and brackish water side, temperature at different points of plant must be known. Fig. 11 shows the variations of temperature at different parts of the plant for different heat inputs to the plant. In general, as expected, increasing the input heating energy (at the water solar heating) augments temperature at different sections of the plant. However, the temperature of humidified air and brain water at the exits of the humidifier approach each other at 120 kW heating input for the considered

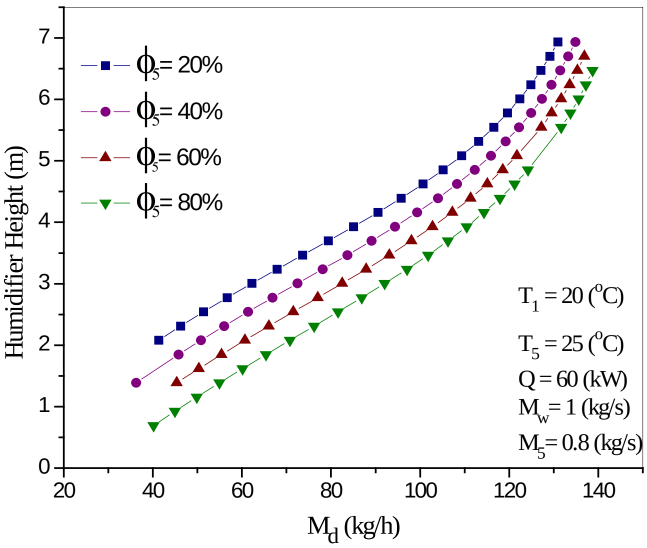

Fig. 9. Effect of humidifier height on the fresh water production.

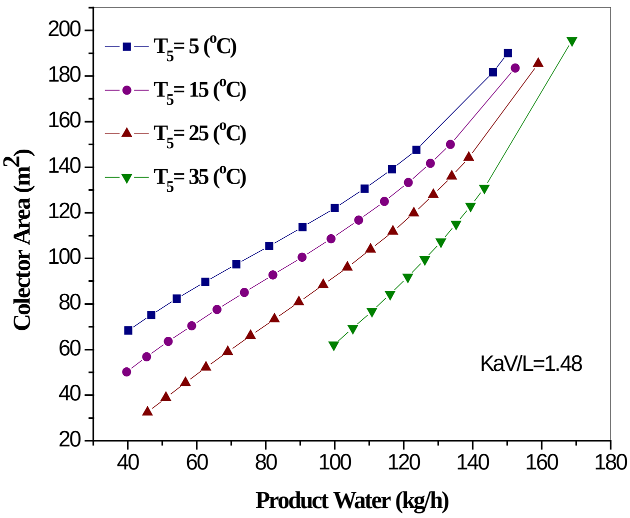

Fig. 10. Effect of solar collector area on the fresh water production.

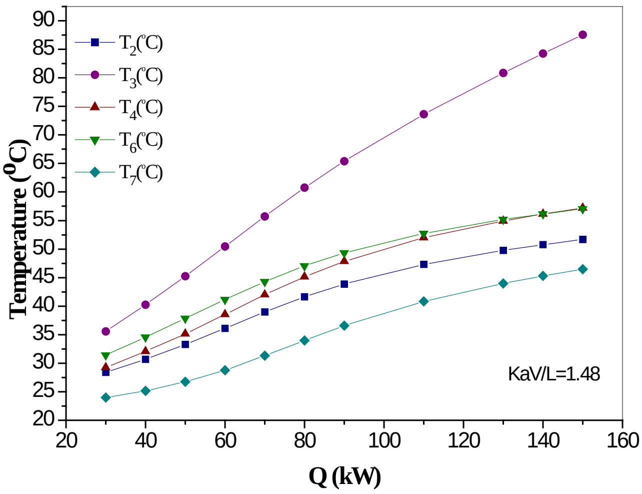

Fig. 11. Effect of heating input on the variation of temperature at the different components of the cycle.

conditions. Using more heating energy does not have a positive effect on the humidifier.

5. Conclusions

Solar water desalination using humidification and dehumidification of air is investigated numerically. Mass and energy balance equations at different parts of the plant have been solved simultaneously to study the effects of different parameters on the rate of fresh water production. It is shown that for given conditions an optimum value of the inlet brackish water flow rate and also an optimum value of inlet air flow rate exist. In addition, increasing the temperature of water at the inlet of the humidifier could increase the fresh water production. It is also shown that increasing the condenser performance (from a certain value based on other conditions) could not have a significant effect on the fresh water production. The same argument is fairly true for the humidifier characteristic value (*KaV*), its performance on augmenting the amount of fresh water decreases from certain values. These show that based on the conditions and desired fresh water production the condenser characteristics and humidifier characteristics have an optimum configuration (performance economy).

6. Symbols

a — Heat and mass transfer area per volume of the humidifier, m^2/m^3
A_{cond} — Condenser heat transfer surface, m^2
A_e — Solar collector area, m^2
C_{pa} — Specific heat of air, kJ/kg °C
C_{pw} — Specific heat of water, kJ/kg °C
F_R — Solar collector heat removal factor
h — Enthalpy, kJ/kg
I — Solar irradiance
K — Mass transfer coefficient, kg/m^2s
LMTD — Logarithmic temperature difference, °C
$\dot{m}$ — Mass flow rate, kg/s
M_5 — Air inlet water, kg/s
M_d — Fresh water production rate, kg/h
M_w — Brackish inlet water, kg/s
N — Number of decks
Q — Input heating energy, kW
Q_u — Effective heating energy at the solar collector, kW
T — Temperature, °C

U_{cond} — Overall heat transfer coefficient at condenser, $kW/m^2\ °C$
U_L — Overall loss coefficient at the collector, $kW/m^2\ °C$
V — Volume of humidifier, m^3
W — Air relative humidity, kg H_2O/kg dry air
α — Solar collector absorption coefficient
ϕ — Relative humidity, %
τ — Solar collector absorption coefficient

Acknowledgment

The authors would like to thank Sistan and Baluchestan Rural Water and Wastewater Company for supporting this research.

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