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Experimental validation of an optimized solar humidification-dehumidification desalination unit

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ABSTRACT

This paper presents experimental validation of a solar humidification-dehumidification (HD) desalination unit which has been constructed based on optimization results reported in previous paper. The unit designed and constructed for 10 L/h fresh water production and was tested during 2 months period. The tests reveal that optimization before construction will result in better performance in practice. Productivity is the most important parameter which improved from 2% to 3.5% in previous experimental works to more than 4%. Moreover, experimental results are in agreement with theoretical ones.

Keywords: Desalination; Humidification; Dehumidification; Optimization; Mathematical programming

1. Introduction

With diminishing water resources, the problem of drinking water seems to be most important issue in the near future. It is predicted that more than half of the world's population will suffer from water shortage by 2025. One of the best solutions of this problem is desalination. In 2006, around 13,000 desalination plants were producing about 38 million m^3/day fresh water with an upward tendency.

Conventional methods are suitable for large and medium capacity of fresh water production. But most remote arid areas need low capacity desalination systems. Air humidification-dehumidification (HD) desalination is a suitable choice for production of fresh water when the demand is decentralized. Advantages of this method are simplicity of process and possibility of using a wide range of thermal energies such as solar, geothermal, exhaust waste and fossil fuel. Also because of low temperature demand of this method it is very compatible with solar energy and total required thermal energy can be obtained from solar radiation.

Although first attempts regarding HD method traces back to the texts of 16th century [1], the first applied research on this method was done in 1968 by Garg et al. [2]. Furthermore, most of researches have been done after 1990 and several units have been built. These units are similar in base but different in type of equipments.

Nawayseh et al. [3] constructed a pilot unit in Malaysia (1999). This unit consisted two vertical PVC pipes as humidifier and dehumidifier chambers. They

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used wooden packing for evaporator. The condenser had been constructed from a finned cylinder with a pipe welded around it. Air circulation in this unit was natural and its production and distillated percent reached 2.2 kg/h and 3.2%, respectively.

Another unit was built by Ben Bacha et al. [4] in Tunisia (1999). Thorn trees were used for evaporator and condenser was constructed from polypropylene plates. The thermal insulation of chambers was achieved by the plates of polypropylene which covered the whole internal surface of chambers and protected the external insulating styrofoam layer against corrosion. Presented curves show that production of the unit at humidifier inlet water temperature of 70°C, is 7.2 kg/h and achieved distillated percent is 2.5%. Solar flat plate collectors of 7.2 m^2 were used to heat salt water. Then by using storage tank required surface area was decreased to 6 m^2.

Orfi et al. [5] constructed another unit in Tunisia (2004). In this unit also air heating solar collectors was used. A horizontal wooden evaporator which had a rectangular cross section was used. In order to improve the heat and mass exchange, five parallel plates made with wood and covered with textile (cotton) were fixed in the evaporator. The condenser chamber was also horizontal and had a rectangular cross section. It concluded two rows of long finned cylinders made of copper. Maximum reported value for production of this unit was 1.3 L/h. Also presented theoretical curves show maximum distillated percent of 3%. But in practice a much lower value was obtained.

Mentioned works and most of other reported works have a similar procedure. First a unit has been constructed based on primary design. Then theoretical and experimental studies and simulation have been done on the unit. Finally by variation of effective parameters the operation performance of the unit has been improved.

In two previous papers [6,7] air HD process and effects of different parameters on its performance were analyzed and a mathematical programming model was presented to optimize the process with different objective functions. Then the model was developed by adding the solar part and finally a low cost design for solar HD desalination was obtained. In the next step a unit with capacity of 10 L/h based on optimization results was constructed which has been located at Iranian Research and Development center for Chemical Industries (IRDCI), Karaj, Iran. In this paper, results of testing on this unit are introduced, validated and analyzed.

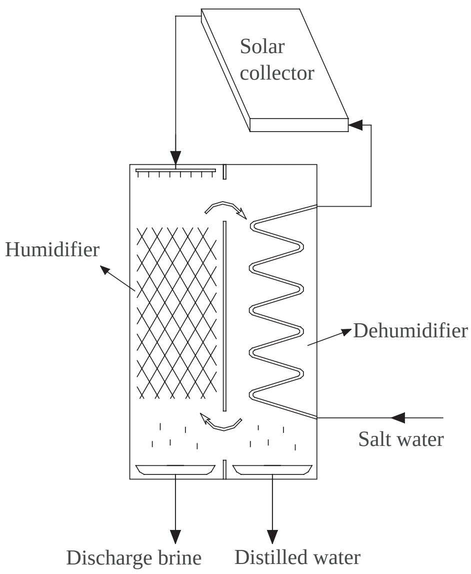

Fig. 1. Sketch of solar HD unit with closed air cycle and water heating.

2. Governing equations

Fig. 1 shows a solar HD system with closed air cycle. Governing equations of HD process based on following assumptions were completely explained in previous papers [6,7].

- The HD processes take place in adiabatic conditions.
- Air and water distribution in the towers is uniform horizontally and hence there is only vertical gradient of temperature and humidity.

These equations are summarized as:

$$d\dot{m}_{we} = d\dot{m}_{ve} = \dot{m}_{ae} d\omega_e \tag{1}$$

$$\frac{dT_{we}}{S_e dz} = \frac{h_{we} a_{He} (T_{we} - T_{ie})}{\dot{m}_{we} C_{we}} \tag{2}$$

$$\frac{d\omega_e}{S_e dz} = \frac{k_{ae} a_{Me} (\omega_{ie} - \omega_e)}{\dot{m}_{ae}} \tag{3}$$

$$\frac{dT_{ae}}{S_e dz} = \frac{h_{ae} a_{He} (T_{ie} - T_{ae})}{\dot{m}_{ae} (C_{ae} + \omega_e C_{ve})} \tag{4}$$

$$h_{we} a_{He} (T_{we} - T_{ie}) dz = h_{ae} a_{He} (T_{ie} - T_{ae}) dz + L_{ve} k_{ae} a_{Me} (\omega_{ie} - \omega_e) dz \tag{5}$$

$$\omega_{ie} = f_{exp}(T_{ie}) = 2.19 \times 10^{-6} T_{ie}^3 - 1.85 \times 10^{-4} T_{ie}^2 + 7.06 \times 10^{-3} T_{ie} - 0.077 \quad (6)$$

$$d\dot{m}_d = d\dot{m}_{vc} = \dot{m}_{ac} d\omega_c \quad (7)$$

$$\frac{dT_{wc}}{S_c dz} = \frac{h_{wc} a_{Hc} (T_{ic} - T_{wc})}{\dot{m}_{wc} C_{wc}} \quad (8)$$

$$\frac{d\omega_c}{S_c dz} = \frac{k_{ac} a_{Mc} (\omega_c - \omega_{ic})}{\dot{m}_{ac}} \quad (9)$$

$$\frac{dT_{ac}}{S_c dz} = \frac{h_{ac} a_{Hc} (T_{ac} - T_{ic})}{\dot{m}_{ac} (C_{ac} + \omega_c C_{vc})} \quad (10)$$

$$h_{wc} a_{Hc} (T_{wc} - T_{ic}) = h_{ac} a_{Hc} (T_{ic} - T_{ac}) + L_{vc} k_{ac} a_{Mc} (\omega_{ic} - \omega_c) \quad (11)$$

$$\omega_{ic} = f_{exp}(T_{ic}) \quad (12)$$

$$\dot{Q} = \dot{m}_{wc} C_p (T_{we}(n) - T_{wc}(m)) \quad (13)$$

$$\dot{Q}_{sol} = \frac{\dot{Q}}{\eta_{col}} \quad (14)$$

$$\eta_{col} = 0.8 - 6.8 \left(\frac{T_{col,in} - T_{amb}}{I_T} \right) \quad (15)$$

$$A_{col} = \frac{\dot{Q}_{sol}}{I_T} \quad (16)$$

f_{exp} (in Eqs. (6) and (12)) is an experimental function and has been introduced by Stocker and Jones [8]. Also Eq. (15) is an experimental relation for estimation of solar flat plate collector efficiency [9].

3. Mathematical programming model

Main constraints of mathematical programming model of solar HD system were presented in previous section. Algebraic equations are used in to the model in original form. But for applying the differential equations, firstly they should be formed as algebraic equations using the finite difference method. So, humidifier and dehumidifier towers are divided into ‘n’ and ‘m’ elements respectively. By adding some operational constraints such as recycling relations, pinch temperature difference, relations of packing characteristics, air speed limitations etc, which were discussed in detail and reported in previous papers [6,7], the model constraints are expressed as:

$$T_{we}(i+1) = T_{we}(i) \left(1 - \frac{z_e h_{we} a_e S_e}{\dot{m}_{we} C_{we}} \right) + T_{ie}(i) \frac{z_e h_{we} a_e S_e}{\dot{m}_{we} C_{we}}, \quad 1 \le i < n \quad (17)$$

$$T_{ae}(i+1) = T_{ae}(i) \left(1 + \frac{z_e h_{ae} a_e S_e}{\dot{m}_{ae} C_e} \right) - T_i(i) \frac{z_e h_{ae} a_e S_e}{\dot{m}_{ae} C_e}, \quad 1 \le i < n \quad (18)$$

$$\dot{m}_{we}(i+1) = \dot{m}_{we}(i) + \dot{m}_{ae}(\omega_{ae}(i+1) - \omega_{ae}(i)), \quad 1 \le i < n \quad (19)$$

$$L_{ve} k_{ae} (\omega_{ie}(i) - \omega_{ae}(i)) = h_{we}(T_{we}(i) - T_{ie}(i)) + h_{ae}(T_{ae}(i) - T_{ie}(i)), 1 \le i \le n \quad (20)$$

$$\omega_e(i) = f_{exp}(T_{ae}(i)), 1 \le i \le n \quad (21)$$

$$\omega_{ie}(i) = f_{exp}(T_{ie}(i)), 1 \le i \le n \quad (22)$$

$$T_{wc}(j+1) = T_{wc}(j) \left(1 + \frac{z_c h_{wc} a_c S_c}{\dot{m}_{wc} C_{wc}} \right) - T_{ic}(j) \frac{z_c h_{wc} a_c S_c}{\dot{m}_{wc} C_{wc}}, \quad 1 \le j < m \quad (23)$$

$$T_{ac}(j+1) = T_{ac}(j) \left(1 - \frac{z_c h_{ac} a_c S_c}{\dot{m}_{ac} C_c} \right) + T_{ic}(j) \frac{z_c h_{ac} a_c S_c}{\dot{m}_{ac} C_c}, \quad 1 \le j < m \quad (24)$$

$$L_{vc} k_{ac} (\omega_{ic}(j) - \omega_c(j)) = h_{wc}(T_{wc}(j) - T_{ic}(j)) + h_{ac}(T_{ac}(j) - T_{ic}(j)), \ 1 \le j \le m \quad (25)$$

$$\omega_c(j) = f_{exp}(T_{ac}(j)), \ 1 \le j \le m \quad (26)$$

$$\omega_{ic}(j) = f_{exp}(T_{ic}(j)), \quad 1 \le j \le m \quad (27)$$

$$\dot{m}_{we}(1) = \dot{m}_{wc} \quad (28)$$

$$\dot{m}_d = \dot{m}_{we}(1) - \dot{m}_{we}(n) \quad (29)$$

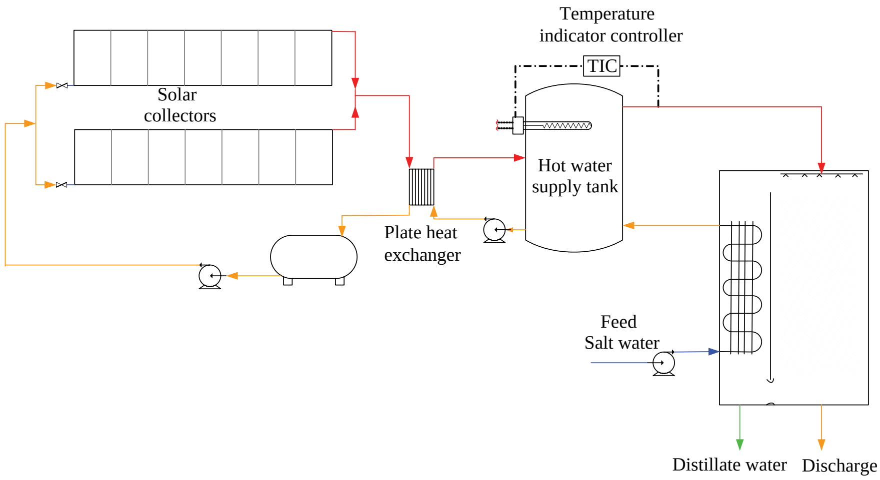

Fig. 2. PFD of constructed solar HD desalination unit.

$$\dot{Q} = \dot{m}_{\mathrm{wc}} C_{\mathrm{wc}} (T_{\mathrm{we}}(1) - T_{\mathrm{wc}}(1)) \tag{30}$$

$$A_{\mathrm{col}} = \frac{\dot{Q}}{I_{\mathrm{T}}\left[0.8 - 6.8\left(\frac{T_{\mathrm{col,in}} - T_{\mathrm{amb}}}{I_{\mathrm{T}}}\right)\right]} \tag{31}$$

$$\dot{m}_{\mathrm{R}} T_{\mathrm{we}}(n) + \dot{m}_{\mathrm{feed}} T_{\mathrm{feed}} = \dot{m}_{\mathrm{wc}}(m) T_{\mathrm{wc}}(m) \tag{32}$$

$$\dot{m}_{\mathrm{R}} + \dot{m}_{\mathrm{feed}} = \dot{m}_{\mathrm{wc}}(m) \tag{33}$$

$$\dot{m}_{\mathrm{ae}} = \dot{m}_{\mathrm{ac}} \tag{34}$$

$$T_{\mathrm{ae}}(1) = T_{\mathrm{ac}}(1) \tag{35}$$

$$T_{\mathrm{ae}}(n) = T_{\mathrm{ac}}(m) \tag{36}$$

$$T_{\mathrm{ac}}(1) - T_{\mathrm{wc}}(1) \geq T_{\mathrm{pc}} \tag{37}$$

$$T_{\mathrm{ac}}(m) - T_{\mathrm{wc}}(m) \geq T_{\mathrm{pc}} \tag{38}$$

$$T_{\mathrm{we}}(i) - T_{\mathrm{ae}}(i) \geq T_{\mathrm{pe}}, \quad 1 \leq i \leq n \tag{39}$$

$$\frac{kaV}{L} = (1.2222H + 0.3667) \times \left(\frac{L}{G}\right)^{-0.62} \tag{40}$$

4. Constructed system description

In previous papers [6,7] the presented model was solved with different objective functions and most effective parameters on performance of system was discussed. At the next step a pilot unit was constructed based on the solution obtained by cost objective function. Fig. 2 shows PFD of this unit and its supplements.

In the humidifier, structure packing with 152 m^2/m^3 specific area having dimensions 30 × 40 × 240 cm^3 was used. For dehumidification a fin-tube heat exchanger was used composed of 150 m total length of copper pipe with inner diameter of 11 mm and 30 aluminum fins. Dimensions of this condenser are 28 × 40 × 196 cm^3 and it has one inlet and three outlets to test different areas.

The chamber containing humidifier and dehumidifier was constructed from poly methyl meta acrylate (PMMA) plates in dimensions of 67 × 44 × 300 cm^3. PMMA material is a thermal insulator and transparent material which allows in situ monitoring of distillation process (Fig. 3).

Flat plat collectors were used for heating of outlet salt water of dehumidifier. First the water is heated in a closed cycle in collectors and then the heat is transferred to the salt water through a plate heat exchanger and the heated salt water is pumped to a storage tank. Total surface of collectors used were 28 m^2 which were more than required surface by about 30%. Fig. 4 shows solar collectors and equipments of hot water cycle.

5. Experimental validation and results

The unit was tested during 2 months period. The obtained productivity in practice was quite satisfactory. Although the system had been optimized based on production cost [3–5], productivity was also improved in comparison with previous experimental works. Most of reported productivity values are between 2% and 3.5%, while this value reached more than 4% in this work. Also production per unit volume

Fig. 3. Humidifier and dehumidifier towers of HD desalination unit.

was high. Table 1 presents a comparison between this unit and some other ones.

Fig. 5 shows theoretical and experimental productivity as a function of humidifier inlet water temperature. A considerable fact in this figure is that the experimental curve has been situated above the theoretical one. Also a similar figure has been presented by Ben Bacha et al. [4]. The main reason is that some of injected hot water into the humidifier flashes before reaching to the top surface of the packing. Moreover, in the HD process always a little amount of salt water suspended in air at the top of the tower enters to dehumidifier as tiny drops which causes that distillated water flow rate is more than predicted value as well as increasing of impurity in fresh water. For example in the tests total hardness of produced water was 86 ppm when total hardness of feed water was about 4500 ppm.

Fig. 4. Collectors and hot water storage tanks of HD desalination unit.

Fig. 6 shows effect of humidifier inlet water temperature on specific thermal energy consumption of HD process. As was predictable, direction of energy variation is in contrary to productivity variation. In Fig. 7 values of dehumidifier outlet water temperature measured at different humidifier inlet water temperatures is compared with model's predictions.

Fig. 8 shows effect of feed salt water on productivity and in Fig. 9 dehumidifier outlet water temperature at different feed water temperatures is compared with the predicted values.

As concluded by Soufari et al. [6], heat transfer areas of both humidifier and dehumidifier have optimum values. To validate this result various areas of packing and condenser was tested. In Fig. 10 measured productivities at two different humidifier inlet water temperatures have been shown. Firstly notable difference between each two parallel columns confirms importance of humidifier inlet water temperature. Secondly it is observed that with decreasing of packing area at high temperature of humidifier inlet water, productivity does not decrease very much.

Fig. 11 shows measured productivities at two different dehumidifier inlet water flow rates. As shown at flow rate of 250 kg/h maximum productivity has been obtained when all of condenser and packing areas were used. But at flow rate of 150 kg/h maximum productivity has been obtained by using whole condenser area and 3/4 of packing area. This proves importance of proportion between packing and condenser area and shows that at each flow rate of the dehumidifier inlet water, there is an optimum value for packing to condenser area ratio. In Fig. 11 maximum productivity has been obtained at less flow rate of feed water (150 kg/h). But maximum absolute production and hence production per unit volume are less than those obtained at flow rate of 250 kg/h. Therefore production cost at flow rate of 250 kg/h is less than 150 kg/h, which confirms optimization results.

Table 1
A comparison between characteristics of present unit and some other constructed units

	Nawayseh et al. [3]	Ben Bacha et al. [4]	Orfi et al. [5]	Present work
Humidifier characteristics	*Chamber*: PVC pipe with 29.5 cm inside diameter and 3 m height *Evaporator*: wooden packing with 11.9 m^2 area	*Chamber*: made of aluminum, internal surface covered by polypropylene as thermal insulation, external surface insulated against corrosion using Styrofoam layer *Evaporator*: thorn trees	*Chamber*: horizontal with rectangular cross section *Evaporator*: made of wood, five wooden parallel plates covered with textile (cotton) were fixed in the evaporator	*Chamber*: constructed from poly methyl meta acrylate (PMMA) *Evaporator*: structure packing made of polypropylene with specific area of 152 m^2/m^3 and 44 m^2 total area
Dehumidifier characteristics	*Chamber*: PVC pipe with 29.5 cm inside diameter and 3 m height *Condenser*: finned cylinder with 17 cm diameter and 3 m height made of galvanized steel plates. 19.2 m copper pipe welded around it	*Chamber*: made of aluminum, internal surface covered by polypropylene as thermal insulation, external surface insulated against corrosion using Styrofoam layer *Condenser*: polypropylene plates	*Chamber*: horizontal with rectangular cross section *Condenser*: two rows of long finned cylinders made of copper with 1.5 m^2 area and total length of 28 m	*Chamber*: constructed from PMMA plates *Condenser*: fin-tube heat exchanger with 150 m total length of copper pipe and and 30 aluminum sheets, 33 m^2 total area
Humidifier dimensions (m)	Φ 0.316 × 3 (circular cross section)	1.2 × 0.5 × 2.5	2 × 0.9 × 0.2	0.32 × 0.42 × 3
Dehumidifier dimensions (m)	Φ 0.316 × 3 (circular cross section)	1.2 × 0.36 × 3	–	0.3 × 0.42 × 3
Feed water flow rate (kg/h)	Max. 68	288		250
Fresh water flow rate (kg/h)	Max. 2.2	7.2	1.3	10
Distillate percent	3.3% (at low flow rates max. 4.4%)	2.5%	Less than 2%	4% (at low flow rates >5%)

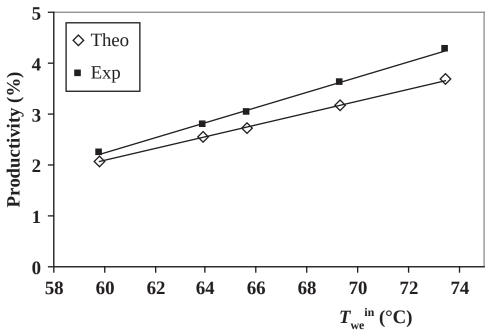

Fig. 5. Effect of humidifier inlet water temperature on productivity

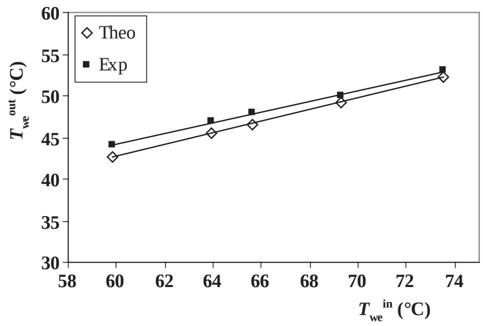

Fig. 8. Productivity vs. dehumidifier inlet water flow rate.

6. Conclusion

In this paper, characteristics of a Solar HD desalination unit constructed based on optimization result was presented and experimental results were validated. In comparison with previous works distillation percent and production per volume of unit have been improved and are quite satisfactory. Therefore it can be concluded that optimization before construction results in better performance. Also by variation of humidifier inlet water temperature and dehumidifier inlet water flow rate, measured productivity was compared with model prediction and was shown that the difference is desirable. Finally decreasing packing and condenser area shows that the ratio of humidifier to dehumidifier heat transfer area is an important parameter and has an optimum value.

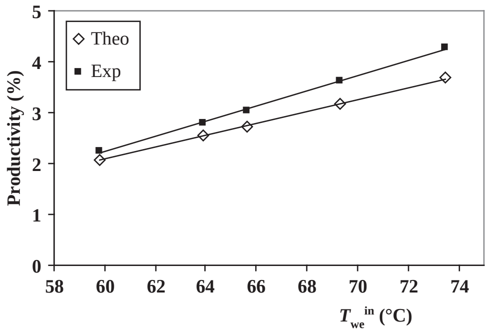

Fig. 6. Effect of humidifier inlet water temperature on specific thermal energy consumption.

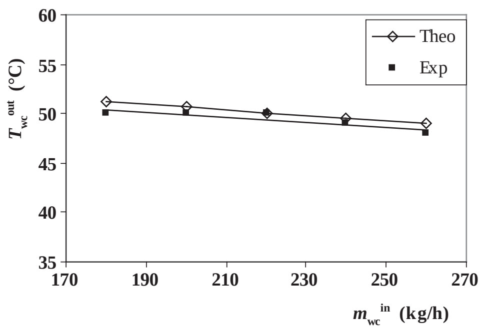

Fig. 9. Measured and predicted temperatures of dehumidifier outlet water for different dehumidifier inlet water flow rates.

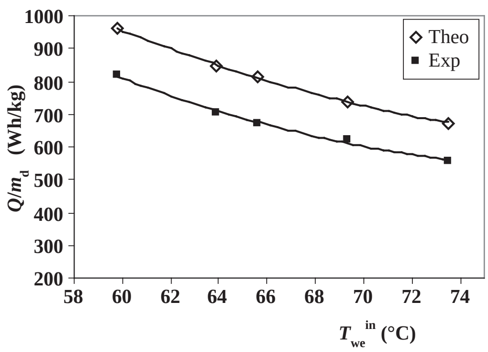

Fig. 7. Measured and predicted temperatures of dehumidifier outlet water for different humidifier inlet water temperatures.

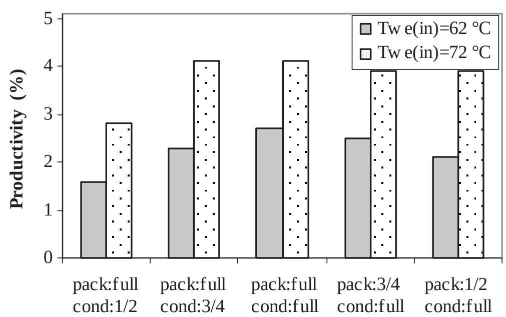

Fig. 10. Effect of heat transfer area on productivity at two different humidifier inlet water temperatures $\dot{m}_{wc}^{in} = 250$ kg/h.

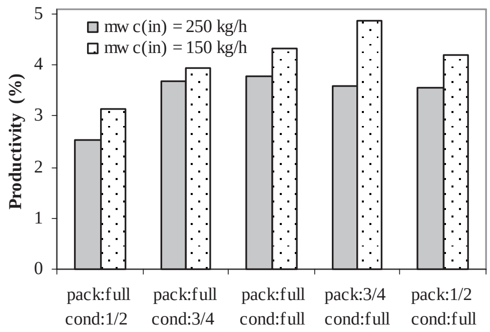

Fig. 11. Effect of heat transfer area on productivity at two different dehumidifier inlet water flow rates $T_{we}^{in} = 70°C$.

Symbols

A	surface area, m^2
a	specific area, m^2/m^3
C	specific heat, J/kg K
G	gas mass flow rate, kg/s
H	packing height, m
h	heat transfer coefficient, W/m^2 K
I_T	tilted irradiance, W/m^2
k	mass transfer coefficient, kg/m^2 S
L	liquid mass flow rate, kg/s
L_v	evaporation latent heat, J/kg
$\dot{m}$	mass flow rate, kg/s
P	power, kW
$\dot{Q}$	heat transfer rate, W
S	cross section, m^2
T	temperature, K and °C
V	packing volume, m^3
z	height, m

Greek

ω	specific humidity, kg vapor/kg dry air
η	efficiency

Subscripts

a	air
amb	ambient
c	condenser (dehumidifier)
col	collector
d	distillated water
e	evaporator (humidifier)
H	heat
i	interface
M	mass
P	pinch
v	vapor
w	water

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