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THERMODYNAMIC ANALYSIS AND OPTIMIZATION OF SOLAR DESALINATION UNIT, USING HUMIDIFICATION-DEHUMIDIFICATION CYCLE USABLE IN REMOTE WATERLESS AREAS BY APPLYING DOE METHOD

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ABSTRACT

This paper has studied and numerically simulated a desalination unit with humidification-dehumidification cycle which uses solar energy as its source of heat (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 analyzed numerically using Newton method, and analysis of cycle parameters have been done to determine the amount of desalinated water produced by the system. In addition, sensitivity of cycle's different parameters has been examined using analysis and design of experiment (DOE) method in order to obtain optimum conditions.

KEYWORDS: Solar desalination humidification-dehumidification cycle, DOE method.

INTRODUCTION

The increasing development of renewable energies including the sun shows that using this energy in the process of desalination has been suitable and useful, while in recent years, swift rise of the cost of fossil fuels and availability of sun's energy, even in the remote regions of the world, has attracted many to the usages of sun's energy.

Solar desalination unit functioning by humidification and dehumidification 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 mixed with a large amount of water vapour. Using hygrographs, thermodynamic principles show this fundamental process clearly.

Many studies have been carried out about various types of HD cycle desalinations [4-6] . These studies have investigated different ways of increasing the production of desalinated water and performance efficiency of various machines. Goosen, et al. with the aid of HD process, examined some economic and thermodynamic aspects of solar desalination. Their report was based on this fact that commercial production of solar desalination is economically and efficiently advantageous.

NOMENCLATURE

$A_c\ (m^2)$	Solar collector area
$A_{cond}\ (m^2)$	Condenser heat transfer surface
$a\ (m^2/m^3)$	Heat and mass transfer area per volume of the humidifier
$C_{pa}\ (kJ/kg\ ^\circ C)$	Specific heat of air
$C_{pw}\ (kJ/kg\ ^\circ C)$	Specific heat of water
F_R	Solar collector heat removal factor
$M_w\ (kg/s)$	Brackish inlet water
$M_5\ (kg/s)$	Air inlet water
$M_d\ (kg/h)$	Fresh water production rate
$\dot{m}\ (kg/s)$	Mass flow rate
$Q\ (kW)$	Input heating energy
$Q_u\ (kW)$	Effective heating energy at the solar collector
$T\ (^\circ C)$	Temperature
$U_L\ (kW/m^2\ ^\circ C)$	Overall loss coefficient at the collector
$U_{cond}\ (kW/m^2\ ^\circ C)$	Overall heat transfer coefficient at condenser

Special characters

$\varphi\ (\%)$	Relative humidity
τ	Solar collector abso
α	Solar collector abso

Parekh et al. carried out a comprehensive investigation on the background of SDHD systems. They looked at the development of solar stills historically, and concluded that the main factor of the development of these systems is 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 have indicated the effect of inlet air flow rate in the cycle is insignificant. However, the effect of feed water flow rate on the efficiency of a SDHD unit has been described as significant.

Al Hallaj, et al. undertook an experimental study on a SDHD unit. In their unit the air circulates by natural or forced convection and is humidified by the constant 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, the results showed higher production of desalinated water compared to that of solar stills, whereas the effect of air velocity was formerly regarded only in lower performance temperatures. Their investigation indicates higher advantage of applying this kind of water desalination which uses natural and forced air circulation in high and low temperatures, respectively.

Nafey, et al. carried out an experimental work on SDHD process and presented in Suez, Egypt. This plant consists of humidification and dehumidification towers, which are located next to flat-plate solar collectors (for air heating) and water concentrator (for heating water). In the results obtained, air velocity was considered insignificant, and the results showed great influence of inlet water and air temperatures on the production of desalinated water. They introduced an overall equation for estimation of system's production of desalinated water, and proposed a general computational design for SDHD unit [9]. Laboratory and theoretical findings corresponded to each other agreeably.

We can also refer to MEH solar water desalination technology presented by Chefik; Ben Mahmud, et al. 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. stated that in most of the previous studies regarding SDHD technology, obtaining optimal conditions of design, has been a difficult and complicated procedure. using Pinch method, they proposed a design for optimizing the performance of SDHD process. 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.

DESCRIPTION OF SYSTEM FUNCTIONING

The analyzed system consists of main sections including air and water solar collectors, condenser and humidifier tower. Figure 1 illustrates a schema of this system. Feed water (brackish water, turbid water, flowing water with high heaviness and seawater) flow into the condenser through point number one.

Normally, the model of condenser for air-water flows is a type of extended surface heat exchanger. In this system, extended surface Heat exchanger is used because heat transfer coefficient at the side of air is far smaller than that of liquid side, and we need more heat transfer surface at the side of air.

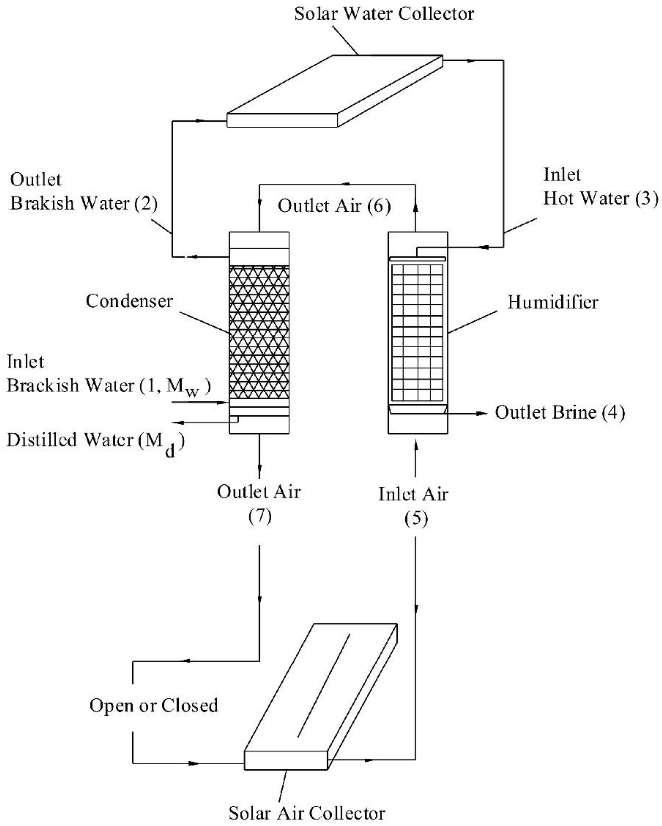

Figure 1 A Schema of Solar Water Desalination Cycle

Therefore, the humid air starts flowing at the side of the vanes which are fixed on the tubes. Feed water runs through the tubes and is preheated by the recovery of condensation latent heat.

Then receive heat in solar collector and after that flows into humidifying tower. Humidifying tower consists of a packing. Packing material is selected from wooden chips which are put above each other separately by some distribution porous plates.

Hot water stream flows into the humidifying tower through the top and is sprayed on the packing. Thus, air and water flow contact level is increased and between the two fluids heat exchange occurs.

Producing and obtaining relatively high humidity in outgoing air from the humidifying tower brings good efficiency for the cycle.

Hot and humid water passes in and out of the side of condenser's vane tubes through the top of humidifier tower. When humid air touches condenser's plates, heat and condensation exchange occurs and the distilled water exits the condenser as desalinated water.

The procedures of the designed sys
following characteristics:

- Air and water flow cycle can be either closed or open.
- Heating energy can be produced by solar collectors, steam flow procedures, diesel motors and other forms of renewable energies.
- The humidifying tower output humid air can be returned to the solar collector and re-humidified; therefore, the ratio of humidifying tower air output from humidifier increases satisfactorily.

EQUATIONS AND COMPUTATIONAL METHOD

The overall details of computational model in every section of desalination unit are based on thermodynamic laws and the main hypotheses used for obtaining a computational method are as follows:

1. The process is assumed to be in constant and stable conditions.
2. Heat and equipment mass loss in the environment is ignored.
3. Air and water vapor are assumed as ideal gas.
4. Humidifying tower and condenser output air is assumed in satiation state.
5. Kinetic and potential energy changes are relatively minor.
6. In energy balance, pumps' consumption power is ignored.

Accordingly, mass and energy balance equations in humidifier are defined as thus:

$$M_a\left(h_6 - h_5\right) = M_w\ h_{f3} - M_b\ h_{f4} \tag{1}$$

$$M_{v6} + M_{b4} = M_{v5} + M_w \tag{2}$$

In which m_a is inlet dry air's mass flow rate (kg/s), h_5 and h_6 humidifier's inlet and outlet air enthalpies (*kJ/kg dry air*), h_{f3} and h_{f4} humidifier's inlet and outlet water enthalpies, and M_b and M_w are respectively outlet salt water and humidifier inlet water mass flow rate.

In the above equation, M_{v5} and M_{v6} are mass flow rate of humidifier's inlet and outlet vapor (kg/s) and the enthalpy of the air passing through the tower (*kJ/kg dry air*) is defined thus:

$$h = H / M_a = cp_a T + W\,(2500.9 + 1.82\,T) \tag{3}$$

Humidity ratio is characterized as a function of atmospheric pressure, steam minor pressure and dry bulb temperature. [13]

$$w_n = \frac{m_{vn}}{m_a} = 0.622\frac{P_{vn}}{P - P_{vn}} \tag{4}$$

In which n is given spots in the cycle and is water's minor pressure in dry bulb temperature *(kPa)* and atmospheric pressure. Relative humidity is also defined as follows,

$$\Phi_n = \frac{P_{vn}}{P_{gn}} \tag{5}$$

Where Φ is relative humidity and P_g is saturate air pressure at the temperature of T_a *(kPa)*.

Condenser energy, mass and heat transfer balance equations are defined thus:

$$M_a\left(h_6 - h_7\right) = M_w\ \left(h_{f2} - h_{f1}\right) - M_d h_{f7} \tag{6}$$

Where M_d is desalinated water's mass debye.

$$M_d = M_a(W_6 - W_7) \tag{7}$$

$$M_w cp_w\,(T_2 - T_1) = A_{cond} U_{cond}\,LMTD \tag{8}$$

Where M_w is the mass flow rate of feed water *(kg/s)*, U_{cond} is overall heat transfer coefficient for condenser (kW/m^2 °C), A_{cond} is the heat transfer surface area of condenser (m^2) and *LMTD* is logarithmic mean temperature difference which is described thus:

$$LMTD = \frac{(T_6 - T_2) - (T_7 - T_1)}{\ln\dfrac{(T_6 - T_2)}{(T_7 - T_1)}} \tag{9}$$

Enthalpy and humidity ratio for saturate air can be obtained from the following relationship. [15]

$$H = 0.00585\,T^3 - 0.497\,T^2 + 19.87\,T - 207.61 \tag{10}$$

$$W = 2.19\,T^3(10^{-6}) - 1.85\,T^2(10^{-4}) + 7.06\,T\,(10^{-3}) - 0.077 \tag{11}$$

The equations shown above are nonlinear and Newton method has been used for solving them.

Figure 2 shows an illustration of a flat-plate solar collector.

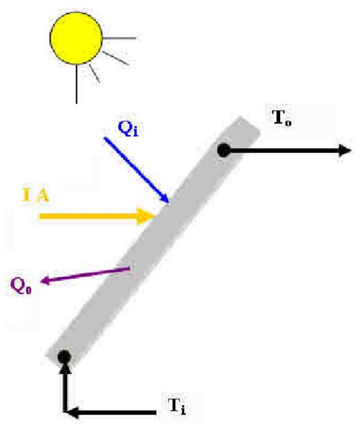

Figure 2 An illustration of a flat-plate solar collector

There I is the solar intensity (W/m^2) plate collector area (m^2), then the solar ir collector is shown by the following,

$$Q_i = I \times A \tag{12}$$

Since heat is lost from collector's surface and walls through convection, transfer and emission, an absorption coefficient must be defined, therefore,

$$Q_i = I(\tau_\alpha) \times A \tag{13}$$

When the collector absorbs heat, its temperature exceeds environment's, and tends to give its temperature to the environment. The amount of this heat lost from the collector (Q_O) depends on the universal heat lost coefficient (U_L) and also on the collector's temperature.

$$Q_o = U_L\ A\,(T_c - T_a) \tag{14}$$

So, the rate of useful energy received by the collector (Q_U) in stable conditions, is described as follows,

$$Q_u = Q_i - Q_o = I\ \tau_\alpha \times A - U_L\ A\,(T_c - T_a) \tag{15}$$

In addition, the heat received by the collector can be described according to the withdrawn amount of fluid heat which is flowing in the collector.

$$Q_u = Mc_P\,(T_o - T_i) \tag{16}$$

Eq. 15 includes collector's parameter of average temperature (T_c), which appears to be difficult to obtain. This makes the quantity of Eq. 15 complicated. Hence, we can define a factor which is able to express the real effective energy coefficient as the overall effective coefficient.
This factor is introduced and stated as a factor of collector heat withdrawal (F_R) by the following expression,

$$F_R = \frac{Mc_P\,(T_o - T_i)}{A\left[I\,\tau\alpha - U_L\,(T_i - T_a)\right]} \tag{17}$$

Maximum effective energy of collector is fulfilled when the whole surface of the collector is in the temperature of the inlet water.
Real, withdrawn effective energy is obtained by multiplying the amounts of collector heat withdrawal factor by maximum effective energy. This makes us rewrite Eq.15 as follows,

$$Q_u = F_R\,A\left[I\ \ \tau_\alpha - U_L\,(T_i - T_a)\right] \tag{18}$$

Eq. 18 is used extensively for obtaining the amount of useful energy withdrawn from the collector. A solar collector's level of effectiveness is described by collector efficiency (η).
This efficiency shows the ratio of effective energy to the amount of radiation received by the collector's surface at a given time

$$\eta = \frac{Q_u}{A\,I} \tag{19}$$

$$\eta = F_R\ \tau_\alpha - F_R\,U_L\left(\frac{T_i - T_a}{I}\right) \tag{20}$$

If $F_R\ \tau_\alpha$ and U_L parameters are assumed constant, then efficiency is defined as a linear function with three parameters which are performance conditions; intensity of solar radiation (I), surrounding air temperature (T_a), temperature of collector's inlet fluid (T_i).
Therefore, this flat-plate solar collector can be evaluated approximately by measuring these three parameters which are obtained from laboratory scales.

DISCUSSION AND CONCLUSION

Firstly, a comparison between the present study and Orfi, et al. has been presented to make sure of the verity of numerical results obtained in Figure 3. The results achieved from the cycle, despite the unavailability of some input parameters [14], show a good behavior.
Additionally, a comparison between the present study and that of Nawayseh, et al. has been done in Table 1 which contains a difference of approximately %3.
A sample system has been designed and analyzed for higher capacities 50(kg/h) of water desalination system.
Study of those parameters which influence the cycle is done based on the average data of Chabahr city climate, including sun radiation, environment temperature, relative humidity, wind velocity and seawater temperature which are shown in Table 2, and Table 3 shows some of the designed parameters which influence the analysis of the cycle.
In order to analyze the cycle respecting the importance of initial investment for construction, one of the fundamental priorities for the designer has been selecting a type of cheap

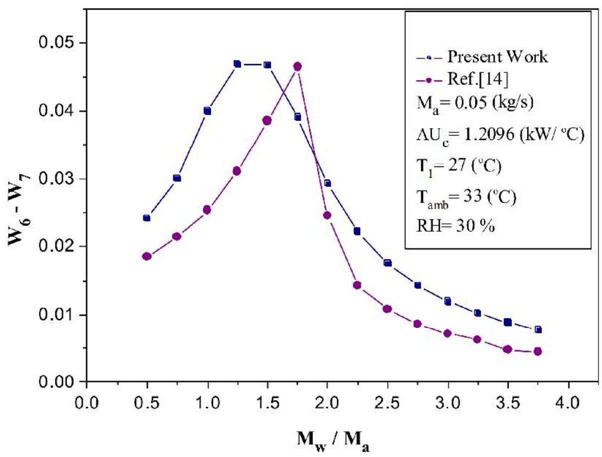

Figure 3 A comparison of the Cycle's Results with other works

Table 1 Comparison of the cycle's results with other works, inlet water mass flow rate, M_w, 0.011 (kg/s). air mass flow rate, M_5, 0.01(kg/s).

	Q (kW)	*T_1 (°C)*	*T_5 (°C)*	*Product (kg/h)*
Ref [15]	1.4	25	35	1.31
Present work	1.4	25	35	1.27

For this reason such parameters as mass flow rates of air and water in the cycles, inlet water and air temperatures, packing height, humidifying chamber, heat and mass transfer surface in condenser and the humidifying tower and collector's absorber-plate surface have been analyzed and investigated in relation to the amount of distilled water produced.

Table 2 Average weather data of Chabahr city Climate

Weather data	**Value**
Solar intensity, I (W/m^2)	665
Ambient temp, T_{amb}	28
Relative humidity, φ (%)	60
Speed of wind (m/s)	3.1
Seawater temp, T_1 (°C)	25

condenser with smaller surface area and dimensions regarding the amount of desalinated water produced, and with lower inlet flux (smaller surface area for the collector) and finally simpler design, and having smaller humidifying tower.

Table 2 Parameter Values Effective on Cycle

Parameters	**Value**
U_L(W/m^{2}°C)	4.605
U_{cond}(W/m^{2}°C)	780
τ_α	0.881
F_R	0.8501

Our aim of analyzing the cycle using DOE technique has been obtain the amount of desalinated water based on specific input parameters including the amount of inlet air and water flow rates, system's inlet water and air temperatures, inlet water's humidity and the level of heat flux provided for the cycle.

Tha advantage of this method for analyzing the parameters of design is based on cost, and also on determining the minimum and maximum amount of desalinated water in the range of changes in given parameters. That is because temperature, the level of humidity ratio and enthalpy in different spots of the cycle detemine the collector, condenser and humidification tower's interface, and this allows the designer to design for the fixed values between the analyzed values of the parameters in the range of attained minimum and maximum values, respecting the initial input parameters. (Offers the designer a large range of values for specifying the collector and condenser's interface.) In order to study the effect of these parameters on the minimum and maximum amount of produced desalinated water, DOE technique is utilised for preparing the calculation chart. In the method of experiments analysis design, using analysis and variance method, preliminary examination of the aforesaid values was done in table 4, Table 5 shows some constant parameters which affect the cycle's process. Then, for enhancing the accuracy of variance analysis, 3^k factorial was used and for the affective parameters achieved from 2^k factorial, three values were included.

Table 4 The Range of Analyzed Values

Parameter	**(Low level)**	**(High level)**
T_1 (°C)	5	27
T_5 (°C)	33	43
Q (kW)	5	12
M_w (kg/s)	0.04	0.1

Table 5 Constant Parameter Values

Parameters	Value
Condenser Area (M^2)	40
Overall Heat Transfer Coefficient (w/m2°C)	0.035
Tower Characteristic (KaV)	$M_w\,(0.07 + 1.62\,((M_w / M_5)^{-0.62})$
Relative humidity, φ (%)	30
Inlet Air Mass FlowRate.M_5 (kg/s)	0.05

Therefore, 81 tests were executed in order to obtain the input variables of DOE analysis. Meters obtained from DOE analysis based on 3^k factorial can be seen in figure 4-6. The changes of desalinated water production rate, inlet water temperature, inlet water flow rate and temperature and heat flux given to the cycle are shown.

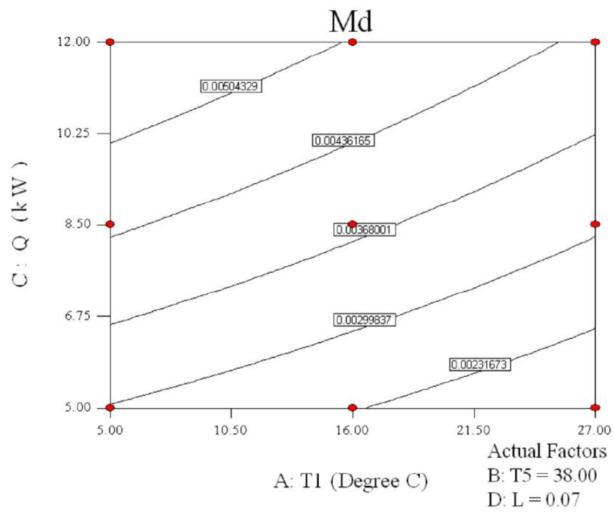

Table 4 The meter for the level of distilled water product in relation to inlet water temperature to the cycle and the amount of heat flux given to the cycle. (3k Factorial)

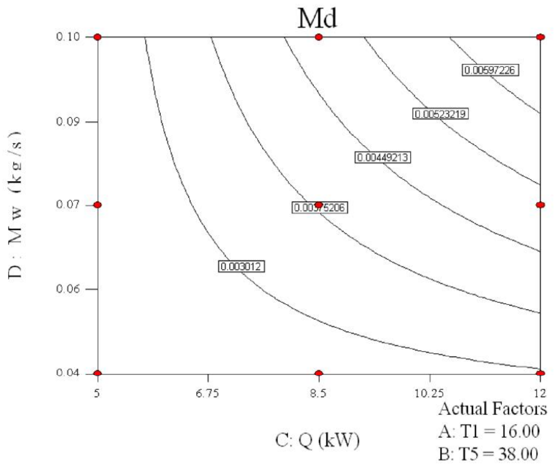

Figure 5 The meter for the level of desalinated water production in relation to the provided heat flux and cycle's inlet water flow rate

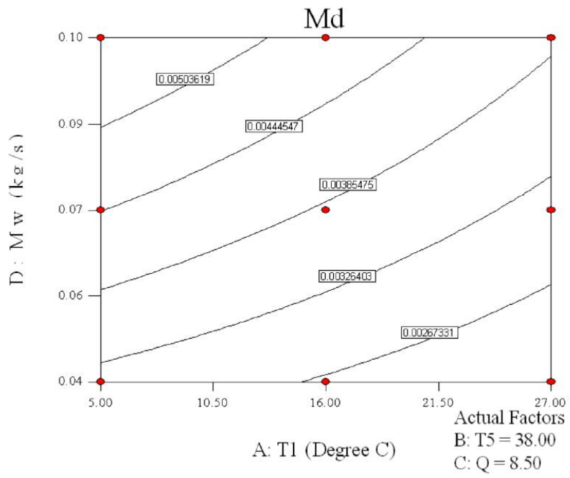

Figure 6 The meter for desalinated water production level in relation to water temperature and flow rate (3k Factorial).

CONCLUSION

The analysis of solar desalination parameters was done and it was known that by increasing such parameters as inlet air temperature, inlet air and water rates and heat and mass transfer surfaces in a given range, level of desalinated water production can be increased, and also by reducing the system's inlet water temperature and increasing the relative humidity of humidification tower's inlet air, heat transfer surface can be reduced for the humidifier. In this study, sensitivity analysis was carried out to identify the important parameters, and the results showed that the level of temperature and the flow rates for inlet water and air have significant effect in optimization of the cycle.

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