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Theoretical Study for a Solar Powered Desalination Unit using Humidification –Dehumidification Technique

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Abstract: A solar powered desalination unit which is working on a humidification –dehumidification technique (HDH) is one of the most important techniques used in seawater desalination in remote and rural areas. It is easy to design, operate and maintain. In this paper, a theoretical study based on a design methodology for a solar assisted desalination unit working on a HDH principle under the prevailing conditions of Tajoura-Libya is carried out. The main target is to study the effect of different design and operating parameters that influence the performance of the unit and its productivity under different design scenarios; (spring, summer, autumn and winter). Results show that the productivity of the unit is increased with a corresponding increase in the inlet air mass flow rate to the solar air heater, inlet water mass flow rate to the humidifier and cooling water mass flow rate to the dehumidifier. A significant increase in the productivity of the unit is achieved when the initial water temperature and the initial mass of water inside the storage tank were increased. Moreover, Gained Output Ratio, GOR, values vary between (0.27 and 0.79) for winter and spring designs and (1.94 and 2.75) for autumn and summer designs respectively. In general, the productivity of the unit is estimated to be within a range from a minimum of (2 ~ 4) kg/m^2.day, in winter to a maximum of (10 ~ 12) kg/m^2.day, in summer, which makes it very convenient for using in rural and remote areas.

دراسة نظرية لوحدة تحلية تعمل بالطاقة الشمسية باستخدام تقنية الترطيب وإزالة الرطوبة

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ملخص: تعتبر وحدة التحلية التي تعمل بمبدأ الترطيب والتجفيف من أهم التقنيات المستخدمة في تحلية المياه في المناطق البعيدة والنائية. فهي سهلة التصميم والتشغيل والصيانة. في هذه الدراسة تم اجراء دراسة نظرية اعتمدت على منهجية تصميم وحدة تحلية تعمل بمبدأ الترطيب والتجفيف بمساعدة الطاقة الشمسية تحت الظروف الجوية لمدينة تاجوراء- ليبيا. هدفت هذه الدراسة الى معرفة

تأثير العوامل التصميمية والتشغيلية المختلفة على أداء الوحدة وانتاجيتها تحت سيناريوهات تصميمية مختلفة (الربيع، الصيف، الخريف، الشتاء). أظهرت النتائج أن إنتاجية الوحدة تزيد زيادة مناظرة في معدل سريان الهواء الداخل الى مسخن الهواء الشمسي، ومعدل سريان المياه الداخلة إلى المرطب ومعدل سريان مياه التبريد الداخلة إلى المجفف. تم الحصول على زيادة مهمة في الإنتاجية عندما تم زيادة درجة حرارة المياه الابتدائية وكتلة المياه في الخزان. بالإضافة الى ذلك، تراوحت قيم نسبة الانتاج المكتسبة، (GOR)، بين (0.27 و 0.79) لتصميمي الشتاء والربيع و (1.94 و 2.75) لتصميمي الخريف والصيف على التوالي. بشكل عام، تم تقدير إنتاجية الوحدة في مدى من أقل قيمة لها وتتراوح بين (2~4) كيلوجرام/ م^2. يوم، في فصل الشتاء إلى أقصى قيمة لها تتراوح بين (10~12) كيلوجرام/ م^2. يوم، في فصل الصيف، هذا ما يجعل هذه الوحدة ملائمة جدا للاستخدام في المناطق النائية والبعيدة.

Keywords: **desalination, humidification, dehumidification, GOR, solar air heater, fresh water, HDH.**

1. INTRODUCTION

In many countries around the world, lack of potable and fresh water is a dilemma. Libya is facing a serious water supply shortage due to an imbalance between limited water resources and its demands. The country's population has tripled since 1950s. As a result of the population growth and the improvement of living standard, the country is confronted with a severe lack of water resources. Water deficits of about 1154 to 4339 Mm^3 have been estimated for the years 1998 and 2025, respectively. There is an urgent need for addressing this problem properly to avoid serious impact on the sustainability of the country development [1].

Libya lies in North Africa region where precipitation rates are very low and fresh water sources are too limited. However, this region is one of the highest solar radiation areas in the world in addition to the other favorable climatic conditions of air temperature and relative humidity. An estimated daily average of solar radiation falling on the ground level is about 8 kilowatt - hour per square meter during the month of June. The duration of sun brightness during the year reaches approximately 3100 hours on the coastal strip and 3900 hours on the southern regions. In other words, 95% purity of the skies or clear days each year is available. Therefore, seawater desalination is the most practical resort to overcome the issue of fresh water shortage [2]. There is no doubt that these data call for the need to pay attention to this energy source and to exploit the use of appropriate technologies to the largest extent possible. Major desalination processes consume a large amount of energy derived from oil and natural gas as heat and electricity, while emitting harmful CO_2 gas. Solar desalination has emerged as a promising renewable energy-powered technology for producing fresh water. Combining the principle of humidification-dehumidification with solar desalination results in an increase in the overall efficiency of the desalination plant, and therefore appears to be the best method of water desalination with solar energy [3]. An extensive research work has been carried out in literature. These studies were mainly focusing on the influence of prevailing weather conditions, different design parameters and operating conditions on the performance and productivity of the HDH system. In addition, different system configurations were also investigated. They were evaluated both theoretically and experimentally, [3-10]. In the present study, a theoretical investigation of a solar powered desalination unit working on HDH principle under the prevailing conditions of Tajoura-Libya is presented. The main objective is to study the effect of different design scenarios (spring, summer, autumn and winter) and operating conditions that affect the performance of the unit and its yield under the prevailing conditions of Tajoura-Libya.

1.1 HDH working principle

The solar powered humidification dehumidification-desalination process is a simple, economical and efficient method for small capacity of fresh water production. A schematic view of a solar desalination unit using humidification-dehumidification principle proposed for this study is shown in Figure 1. The basic elements of the system are solar air heater, a humidifier with storage tank and a dehumidifier.

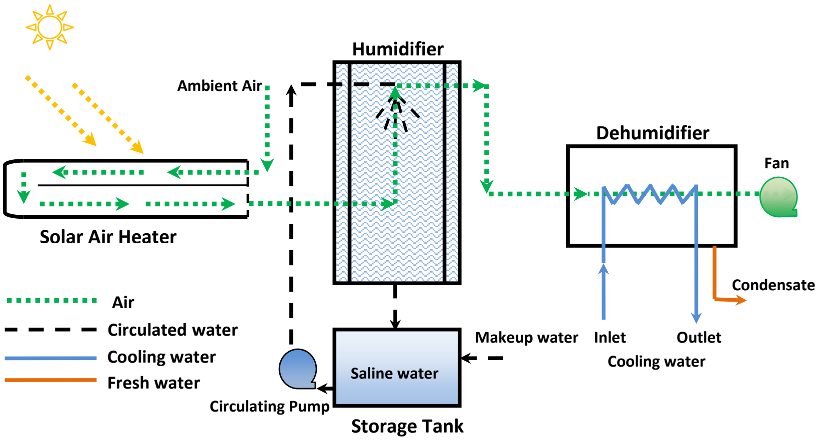

Figure (1). A schematic view of a solar desalination system using humidification- dehumidification principle.

The operation principle of the system is based on the fact that air is heated by using solar air heater and saline water by auxiliary heater and then, these heated air and saline water are forced to enter the humidifier where air is humidified by saline water. Thereafter, the air loaded with water vapor enters the dehumidifier where water vapor condenses on the cooling coil surface and turns into fresh water. HDH systems are ideal for application in small scale systems. They have no parts which require extensive maintenance work like membranes or high temperature steam lines. There is also no bottle-neck in applying HDH for tough and varied water qualities [2]. In view of above merits, HDH systems are convenient for application in arid and remote areas and limited applications. Moreover, there is no need for skilled or qualified personnel to operate and maintain such systems.

2.MATHEMATICAL MODELING

The system shown in Figure 1 consists of a double-pass flat plate solar air heater with two glass covers, humidifier, water storage tank and dehumidifier. This system is working based on the idea of the closed water/open air cycles, (CWOA) and forced air circulation. Mathematical modeling is started by writing the energy balance equations for the system components; solar air heater, storage tank, humidifier and dehumidifier in addition to other related basic and empirical relations that are used for computing heat transfer rates, heat transfer coefficients, physical and thermal properties. The mathematical model and its assumptions for this study are based on the model described in references [4&7]. Appendix (A) shows the main energy balance equations. The size and dimensions of the double pass solar air heater and other basic design and operating parameters are listed in Table (1) below.

2.1. Solution procedure

The ordinary differential equations presented in Appendix (A) are numerically manipulated and solved simultaneously using fourth order Runge-Kutta method (RK-4) by using MATLAB code. The time interval is chosen to be one second and the initial values of T_{g2}, T_{g1}, T_{a1}, T_{a2}, T_{w1} (second glass cover temperature, first glass cover temperature, first air pass temperature, second air pass temperature and storage tank water temperature) respectively, are assumed to be nearly equal to the ambient temperature and T_b, T_p (basin plate temperature and absorber plate temperature) are assumed to be 5 and 10 °C above the ambient temperature respectively.

Table (1). Basic design and operating parameters [7].

Size and dimensions		
W= 1.0, L=1.0	Width and length of solar air heater.	(m)
D = 0.05	Channel thickness of the solar air heater	(m)
x = 0.025	Distance between the two glass covers	(m)
A_c = W*L	Area of the solar air heater	(m^2)
A_s =1.0	Base area of the water storage tank	(m^2)
Th_g =0.003, Th_b=0.002, Th_p=0.001	Thickness of the glass, basin plate and absorber plate	(m)
Physical properties		
m_g= 4.05, m_b =7.85, m_p = 4.5	Mass of the glass, basin plate and absorber plate	(kg)
C_{p-g} = 840, C_{p-b} = 460, C_{p-p} = 840	Specific heat for glass, basin plate and absorber plate	(J/kg.K)
C_{p-a} = 1007, C_{p-w} = 4178	Specific heat of air and water	(J/kg.K)
ρ_g = 2700, ρ_b=7850, ρ_p= 8950	Density of the glass, basin plate and absorber plate	(kg/m^3)
α_g = 0.05, α_p= 0.95	Absorptivity of the glass and the absorber plate	
ε_g = 0.9, ε_b=0.95, ε_p= 0.95	Emissivity of the glass, basin plate and absorber plate	
τ_g = 0.95	Transmissivity of the glass	
$\sigma = 5.67*10^{-8}$	Stefan-Boltzman constant.	(W/m^2K^4)
U_{loss} = 3	Overall heat transfer coefficient	(W/m^2K)
Operating parameters		
m_{w1}=500	Mass of water inside the storage tank	(kg)
M_a =0.027, M_{w1}=0.028, M_{w3}=0.05	Mass flow rate of air, feed water and cooling water	(kg/s)
T_{mw} = 20, T_{w3} = 15	Inlet temperature of make-up water and cooling water	(^oC)

3. RESULTS AND DISCUSSION

In this section, the effect of different design and operating parameters on the HDH unit performance is illustrated and clarified under the prevailing conditions of Tajoura-Libya. The measured weather data for Tajoura Typical Meteorological Year (TTMY) are used in this study [11]. For the different design scenarios; (spring, summer, autumn and winter), the typical weather data of the days (21 March, 21 June, 21 Sep. and 21 Dec.), were chosen respectively as input data for the simulation program between eight o'clock in the morning until ten o'clock in the evening. Figure 2 shows the measured meteorological data for the specified days in Tajoura, namely ambient air temperature, wind speed, humidity and solar radiation on tilted surface.

3.1 The effect of inlet air mass flow rate

Figure 3 illustrates the effect of inlet air mass flow rate on the productivity of the unit under different design scenarios, (spring, summer, autumn and winter). In general, it can be seen that the productivity of the unit is increased by increasing inlet air mass flow rates. When comparing different design scenarios, it can be shown that summer and autumn designs have higher productivity rates than those of winter and spring designs. Changing prevailing weather conditions; solar radiation, air temperature, humidity and air velocity, among seasons and other related parameters are highly affecting the situation. For summer design, the productivity of the unit ranges from about (3 kg/day) to (11 kg/day) corresponding to inlet air mass flow rates of (Ma=0.005 kg/s) and (Ma=0.05 kg/s) respectively. Similarly, for autumn design, the productivity ranges from about (2.9 kg/day) to (10.5 kg/day) at the same flow rates mentioned above. These values are the maximum among the other design scenarios (winter and spring). On the other hand, the trend of the productivity values is completely different for winter and spring designs. In both cases, as inlet air mass flow rate increases, the productivity of the unit increases to its maximum value at a certain point and then it decreases as inlet air mass flow rates become higher. The maximum values are (3 kg/day) for winter design and (4.3 kg/day) for spring design respectively.

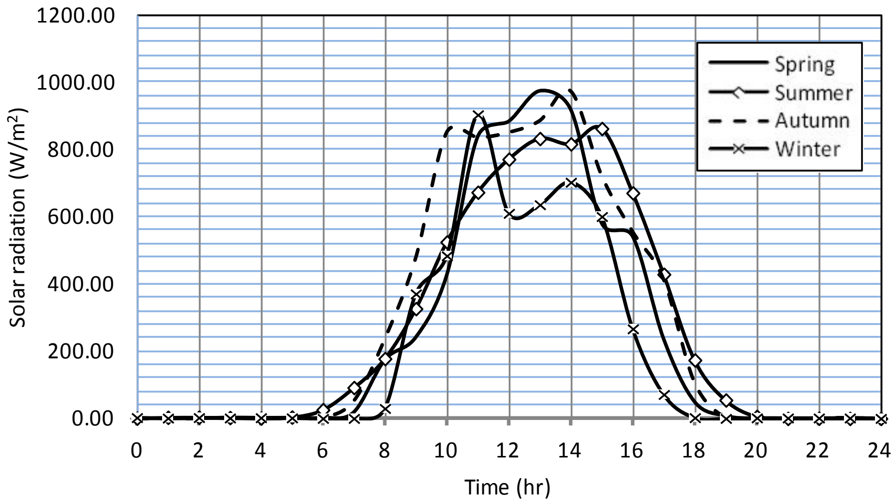

Figure (2-a). Measured meteorological data for Tajoura-Libya; Solar radiation on tilted surface [11].

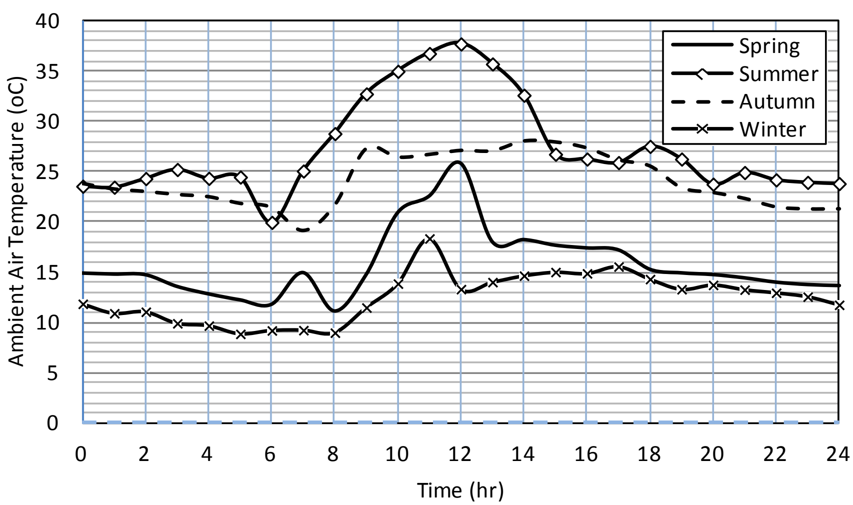

Figure (2-b). Measured meteorological data for Tajoura-Libya; Ambient Air Temperature [11].

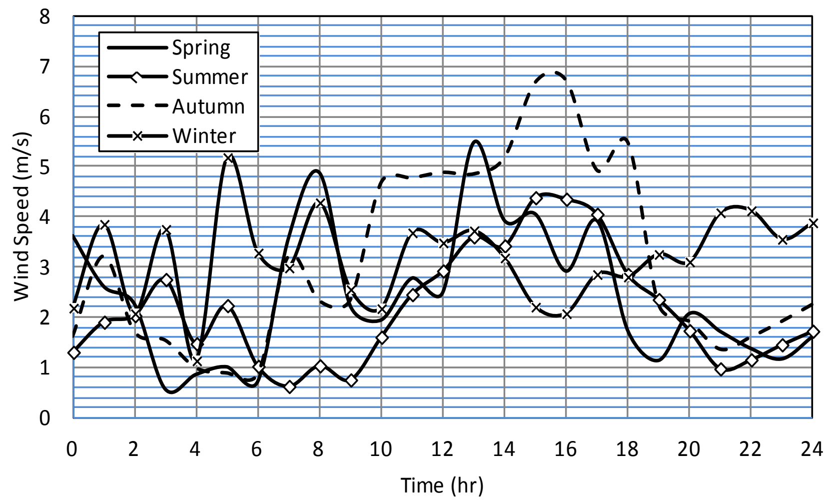

Figure (2-c). Measured meteorological data for Tajoura-Libya; Wind Speed [11].

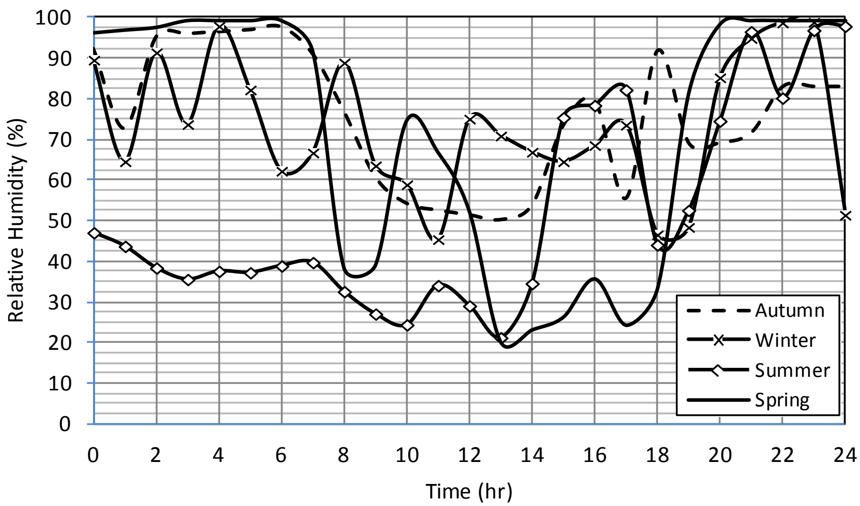

Figure (2-d). Measured meteorological data for Tajoura-Libya; Relative Humidity [11].

This is attributed to the change in prevailing weather conditions among seasons as depicted in Figure 2 and to the fact that, wet-bulb temperature of the air at the outlet of the solar air heater decreases by increasing air mass flow rate. In addition, at a constant water mass flow rate, dry-bulb temperature of the air leaving the humidifier decreases and gets closer to wet-bulb temperature of the air at the inlet of the humidifier and moisture content decreases as well. As a result, water temperature in the storage tank and the rate of vaporization in the humidifier are decreased too. Moreover, at a constant cooling water mass flow rate and temperature, increased air mass flow rate increases the absolute humidity of the air leaving the dehumidifier. For these reasons, the productivity of the unit is decreased; however, the productivity of the unit increases with the increasing value of air mass flow rate up to that optimum value since the air leaving the humidifier carries more water vapor to the dehumidifier.

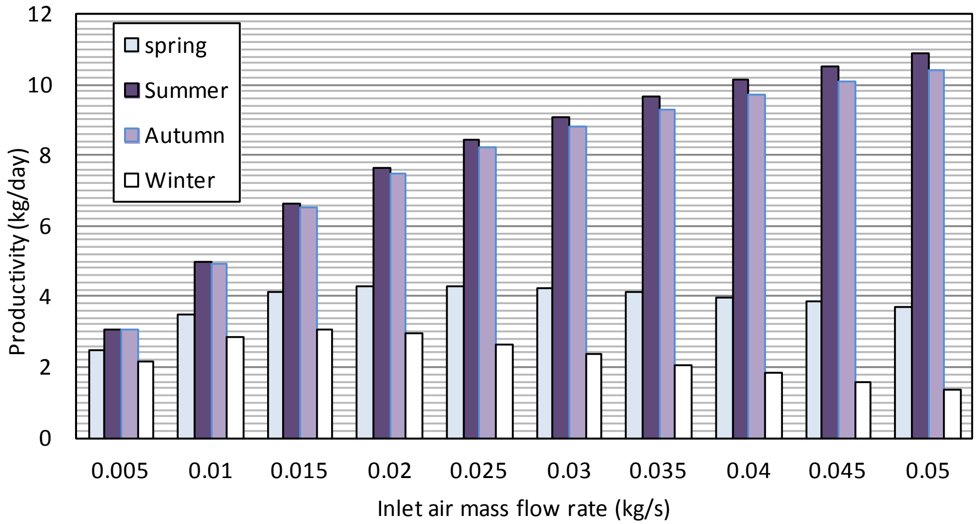

Figure (3). The effect of inlet air mass flow rate on the productivity of the unit.

3.2 The effect of the humidifier inlet water mass flow rate

The effect of the humidifier inlet water mass flow rate on the unit productivity is shown in Figure 4. In general, the productivity of the unit increases with the increasing values of humidifier inlet water mass flow rates. When the inlet mass flow rates vary between (0.005 to 0.05 kg/s), the productivity of the unit increases from (7.6 to 9 kg/day) for summer design and from (7.2 to 8.8 kg/day) for autumn design. Both designs give higher yield of fresh water when compared to spring and winter designs. In addition to the change of prevailing conditions among seasons outside the unit, the temperature of water entering the humidifier is higher than the wet-bulb temperature of air at the inlet of the humidifier. Moreover, as the air is brought into contact with water in the humidifier, the water temperature drops and wet-bulb temperature of the air leaving the humidifier at saturation state increases. For this reason, at a constant air mass flow rate, when the water mass flow rate is increased, the wet-bulb temperature of the air leaving the humidifier increases and approaches the water temperature at the inlet of the humidifier. As a result, the moisture content of the air leaving the humidifier increases and a significant improvement in the unit productivity is noticed.

3.3 The effect of the dehumidifier cooling water mass flow rate

The effect of the dehumidifier cooling water mass flow rate on the system productivity is shown in Figure 5. By increasing the cooling water mass flow rate at a constant cooling water temperature, a significant drop in the surface temperature of the cooling coil can be achieved which results in an increase of the condensation rate of the water vapor on the cooling coil surface and, thus, the unit gives higher yield. Again, both summer and autumn designs give higher values of productivity when compared to spring and winter designs. The maximum values reach up to (8.7 and 8.5 kg/day) respectively at cooling water mass flow rate of

(0.05 kg/s), whereas these values are reduced to (2.5 and 4.5 kg/day) for winter and spring designs at the same cooling water mass flow rate. Change of prevailing weather conditions among seasons outside the unit is another cause for that.

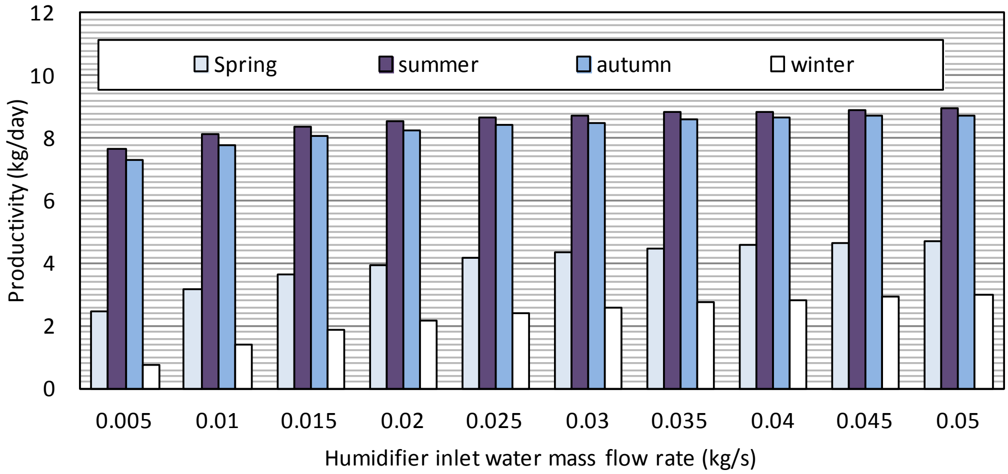

Figure (4). The effect of humidifier inlet water mass flow rate on the productivity of the unit.

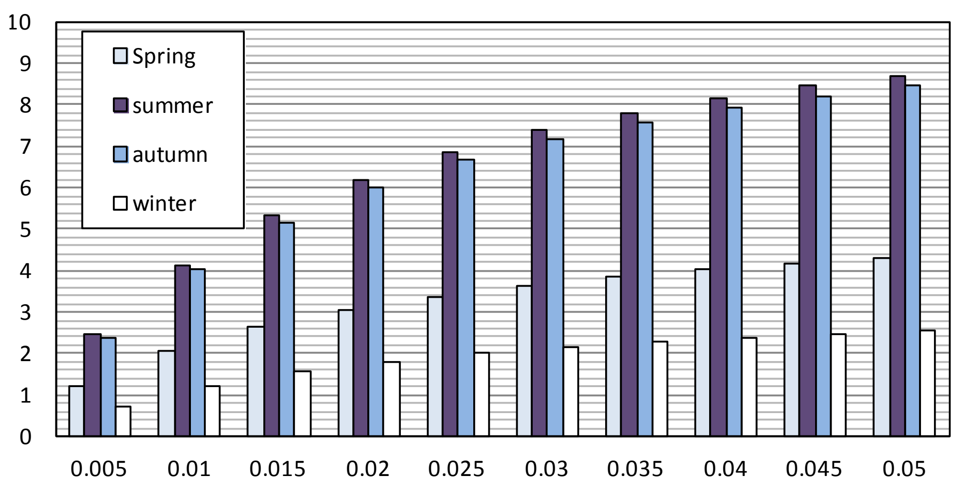

Figure (5). The effect of dehumidifier cooling water mass flow rate on the productivity of the unit.

3.4 The effect of dehumidifier cooling water temperature

The effect of dehumidifier cooling water temperature on the system productivity is shown in Figure 6. By decreasing the cooling water temperature at a constant cooling water flow rate, the amount of heat transferred to the cooling water and the heat transfer coefficient are increased, resulting in a significant drop in the pipe surface temperature. Consequently, the condensation rate of the water vapor on the cooling coil surface is increased too and, thus, the unit gives higher yield. For autumn and summer designs, higher values of productivity are noticed when compared to spring and winter designs. For example, the productivity of the unit reaches up to (9.1 and 9.4 kg/day) respectively at cooling water temperature of (14 °C). Whereas, these values reduce to (3.2 and 5 kg/day) for winter and spring designs at the same temperature.

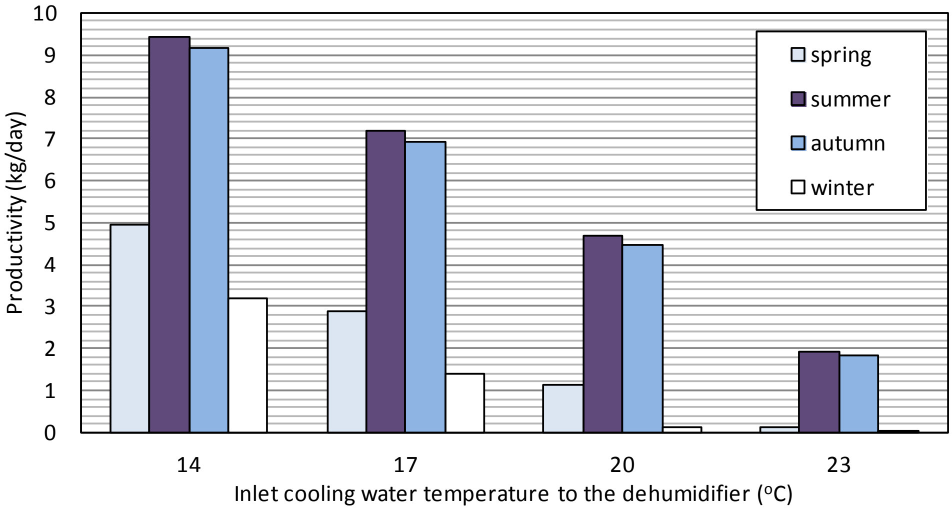

Figure (6). The effect of inlet cooling water temperature to the dehumidifier on the productivity of the unit.

3.5 The effect of the initial water temperature in the storage tank

The effect of the initial water temperature in the storage tank on the unit productivity at constant values of air and water mass flow rates (m_a = 0.027 kg/s and m_w = 0.028 kg/s) is illustrated in Figure 7. Autumn and summer designs give higher yield. They vary from about (7 and 7.2 kg/day) at water temperature of (25 °C) to about (16 and 16.7 kg/day) at water temperature of (50 oC). However, these values are decreased to about (1.5 and 3 kg/day) at (25 °C) and to about (9 and 11 kg/day) at (50 °C) for winter and spring designs. This can be attributed to the change in prevailing weather conditions among seasons. Moreover, the initial water temperature in the storage tank has a considerable influence on the unit productivity. This is due to the fact that any increase in the initial water temperature in the storage tank leads to a corresponding increase in the inlet water temperature to the humidifier which causes an increase in the moisture content of the air leaving the humidifier and, hence, increasing the quantity of fresh water that can be obtained from the unit.

3.6 The effect of the initial water mass in the storage tank

The effect of the initial water mass in the storage tank on the unit productivity is illustrated in Figure 8. The figure shows that the unit productivity is strongly influenced by the increase in the initial water mass in the storage tank, especially, when the water mass flow rate circulated through the humidifier is increased. This is due to the fact that if the initial water mass in the storage tank at a given initial temperature is increased, the water temperature in the storage tank does not change too much during the operation of the unit and, thus, the inlet water temperature to the humidifier remains comparatively higher. As a result, the unit productivity increases since the temperature of water circulated through the humidifier is higher than the wet-bulb temperature of the air at the inlet of the humidifier. Three different values of the initial water mass inside storage tank are tried, (250, 500 and 1000 kg).

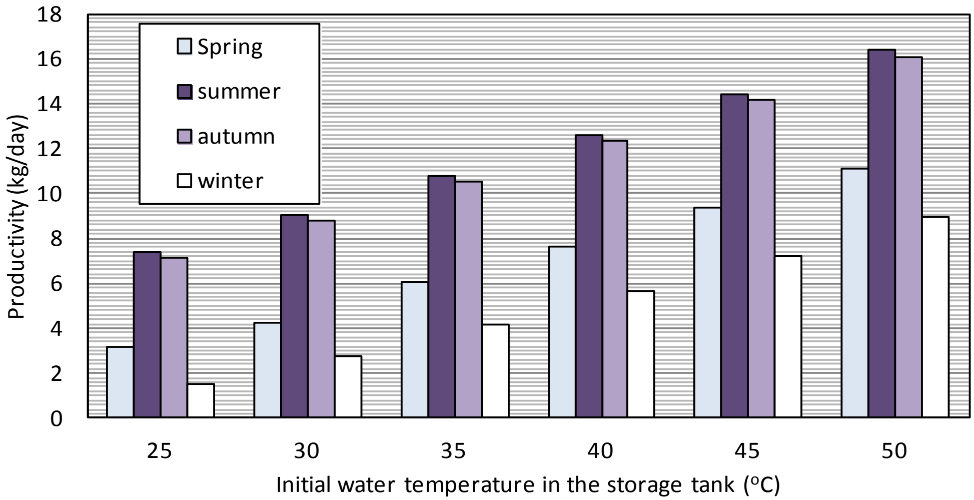

Figure (7). The effect of initial water temperature in the storage tank on the productivity of the unit.

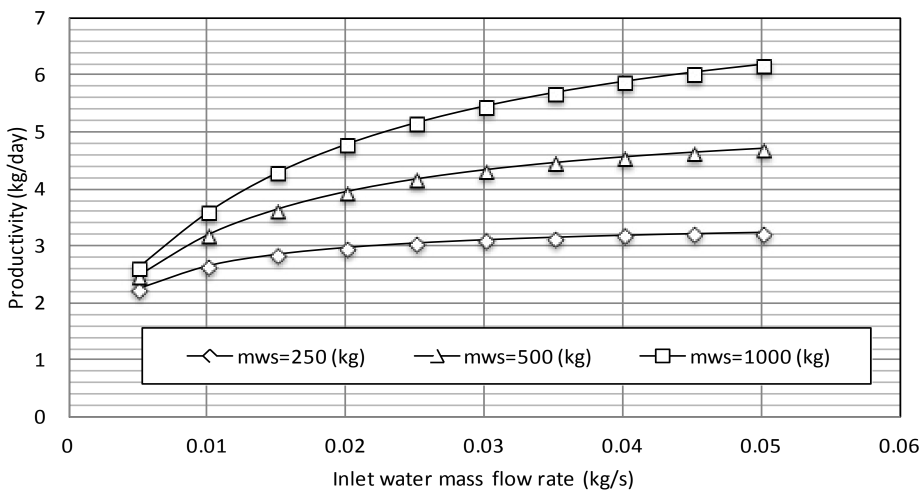

Figure (8). The effect of initial water mass in the storage tank on the productivity of the unit.

3.7 Specific water production

The amount of water produced per square meter of solar collector area per day is defined as specific water production. This parameter is an index of the solar energy efficiency of the HDH cycle, and is of great importance as the great part of the capital cost of the HDH system comes from the solar collector cost which represents (40% - 45%) for air-heated systems and (20% - 35%) for water-heated systems [13]. Figure 9 shows specific water production values at different design scenarios, spring, summer, autumn and winter. It is clear that summer and autumn designs have higher values when compared to spring and winter. This is attributed mainly to the variations in the prevailing seasonal weather conditions (solar radiation, ambient air temperature, relative humidity and wind speed). Consequently, the amount of fresh water produced at spring and winter designs is relatively low when compared with summer and autumn designs. Moreover, the values of specific water production get lower as the area of the air solar heater increases. This agrees with reference [12], where specific water production values vary between (4 to 12 kg/m^2.day) that correspond to solar heater area of (1.9 to 6 m^2) respectively.

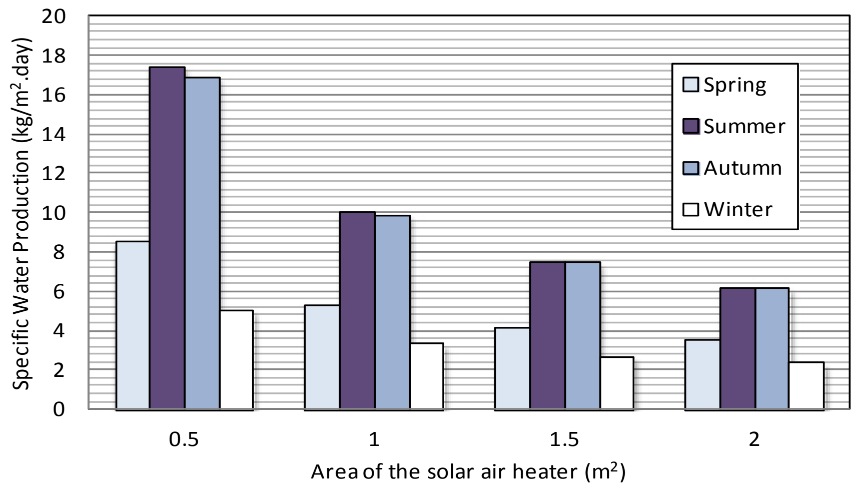

Figure (9). Specific water production at different design scenarios.

3.8 Gained Output Ratio, GOR

The Gained Output Ratio (GOR), or sometimes known as the performance ratio, is defined as the amount of product produced for a given heat input according to the following equation;

$$GOR = \frac{m_p h_{fg}}{Q_{in}} \quad \text{.......} \quad (1)$$

where, m_p is the mass flow rate of the fresh water produced, h_{fg} , is the vaporization heat evaluated at the inlet water temperature and Q_{in}, is the heat input to the unit [13]. According to literature, a solar still has a GOR of about 0.5, air-heated HDH cycles have a GOR that ranges from 1.7 to 3, whereas the water-heated cycles have a GOR that ranges between 0.3 and 4.5. GOR values for conventional desalination systems such as RO (GOR = 35 - 45), MSF (GOR of about 8), and MED (GOR up to 12) [14]. In this study, GOR values for different design scenarios (spring, summer, autumn and winter) are evaluated according to the above equation. Table 2 illustrates GOR values for different design scenarios.

According to Table 2, GOR values for summer and autumn designs are acceptable and within the margin stated in the literature above. However, for spring and winter designs, GOR values are below the suggested values in literature. This is attributed to low quantities of fresh water produced by the unit.

Table (2). GOR values for different design scenarios.

Spring	Summer	Autumn	Winter
0.79	2.75	1.94	0.27

4. CONCLUSIONS

In this paper, a theoretical investigation of a solar assisted desalination unit working on HDH principle under the prevailing conditions of Tajoura-Libya is presented. The design methodology is mainly based on studying the influence of different design scenarios; (spring, summer, autumn and winter), operating conditions and other related parameters are numerically investigated. The most important conclusions of this study can be summarized as follows;

The unit productivity increases by increasing the inlet air mass flow rates to the solar air heater and inlet water mass flow rates to the humidifier. A significant improvement in the productivity of the unit is noticed when the initial mass of water inside the storage tank is increased, especially when water mass flow rate to the humidifier is increased. Moreover, initial water temperature inside the tank has a remarkable effect on the productivity of the unit. In order to obtain a reasonable amount of fresh water, the water temperature inside the tank should be increased. Increasing the cooling water mass flow rate to the dehumidifier and reducing its temperature lead to a sizable decrease in the surface temperature of the cooling coil and hence the productivity of the unit is improved. As the surface area of the solar air heater increases, the specific water production values tend to decrease. Moreover, summer and autumn designs show higher yield values when compared with spring and winter designs. Gained output ratio, GOR, values vary between (0.27 and 0.79) for winter and spring designs and (1.94 and 2.75) for autumn and summer designs respectively. In general, the productivity of the unit is estimated to range from a minimum of (2 ~ 4) kg/m^2.day in winter to a maximum of (10 ~ 12) kg/m^2.day in summer.

Finally, due to its design simplicity, easy maintenance in addition to low cost, this unit is recommended for rural and remote areas.

5. REFERENCES

[1]. Wheida E. and Verhoeven R., "Desalination as a water supply technique in Libya", Desalination, 165, 89–97, 2004.

[2]. Abdunnabi M.J.and.Ramadan A. M," Simulation Study of the Thermal Performance of MSF Desalination Unit Operating by Solar Vacuum Tube Collectors", Power and Energy Systems and Applications (PESA), Las Vegas, USA, Nov. 12-14, 2012..

[3]. Said Al-Hallaj, Sandeep Parekh, Farid, Selman," Solar desalination with humidification–dehumidification cycle: Review of economics", Desalination, 195, 169–186, 2006.

[4]. Yamali C. and Solmus I," Theoretical investigation of a humidification-dehumidification desalination system configured by a double-pass flat plate solar air heater", Desalination, 205, 163-177, 2007.

[5]. EL-Shazly A.H,. EL-Gohary M.M and Ossman M.E.," Performance characteristics of a solar humidification dehumidification unit using packed bed of screens as the humidifier", Desalination and Water Treatment, 16, 17-28, 2010.

[6]. Zhani K., Ben Bacha H., Damak T., "Modeling and experimental validation of a humidification-dehumidification desalination unit solar part", Energy 36, 3159-3169, 2011.

[7]. Tiwari A. K and Sachdev T.," Conceptual Analysis of Desalination System Working on Humidify and Dehumidify Technique Using Solar Air Heater", International Conference on Mechanical and Robotics Engineering (ICMRE'2012), May 26-27, 2012.

[8]. Taha E. Farrag, Mohamed S. Mahmoud and Wael Abdelmoez," Experimental Validation for Two Stages Humidification-Dehumidification (HDH) Water Desalination Unit", Seventeenth International Water Technology Conference, IWTC 17, 2013.

[9]. Simoes T., Varela L., Coelho L., Abreu A., Matias M. Abril C., Joyce A., Giestas M., Loureiro D., "Autonomous HDH Solar Seawater Desalination", EuroSun 2014, 16–19, Sep. 2014.

[10]. Mofreh H. Hamed, Kabeel A. E. , Omara Z. M., Sharshir S. W.," Mathematical and experimental investigation of a solar humidification–dehumidification desalination unit", Desalination, 358, 9–17, 2015.

[11]. Abdunnabi M.J., Algmati, F.M, Alrgehi, A, "Climatic design year of Tajoura", Journal of Solar Energy and Sustainable Development, JSESD, No 1, Vol,1, CSERS, 2012.

[12]. Prakash Narayan G.,"The potential of

solar-driven humidification-dehumidification desalination for small-scale decentralized water production", Renewable and Sustainable Energy Reviews 14.4 (2010): 1187-1201.

[13]. Kabeel A. E., Mofreh, Hamed H., Omara Z. M., Sharshir S. W., "Water Desalination Using a Humidification-Dehumidification Technique - A Detailed Review", Natural Resources, 4, 286-305, 2013.

[14]. John H. Lienhard, Mohamed A. Antar, Amy Bilton, Julian Blanco, & Guillermo Zaragoza," Solar Desalination", Chapter 9, Annual Review of Heat Transfer, 273-345, 2012.

APPENDIX (A): MAIN ENERGY BALANCE EQUATIONS

1- Energy balance equations for the solar air heater

a- Second glass cover

$$m_g C_{p\_g} \frac{dT_{g_2}(t)}{dt} = I(t)\alpha_g A_c + q_{r,g_1\_g_2} - q_{c,g_2\_amb} - q_{r,g_2\_sky} + q_{c,g_1\_g_2}$$

b- First glass cover

$$m_g C_{p\_g} \frac{dT_{g1}(t)}{dt} = I(t)\alpha_g \tau_g A_c - q_{r,g_1\_g_2} - q_{c.g_1\_a_1} + q_{r,p\_g_1} - q_{c,g_1\_g_2}$$

c- First air pass

$$\rho_a(t) A_c D\, C_{p\_a} \frac{dT_{a_1}(t)}{dt} = q_{c,p\_g_1\_a_1} + q_{c,g_1\_a_1} - M_a C_{p\_a}(T_{a_{1e}}(t) - T_{a_j}(t))$$

d. Absorber plate

$$m_p C_{p\_p} \frac{dT_p(t)}{dt} = I(t)\alpha_p \tau_g^2 A_c - q_{c,p-a_2} - q_{c,p-a_1} - q_{r,p\_g_1} - q_{r,p\_b}$$

e- Second air pass

$$\rho_a(t) A_c D\, C_{p\_a} \frac{dT_{a_2}(t)}{dt} = q_{c,p\_a2} + q_{c,b\_a_2} - M_a C_{p\_a}(T_{a_2\_e}(t) - T_{a_1\_e}(t))$$

f- Base plate

$$m_b C_{p\_b} \frac{dT_b(t)}{dt} = q_{r,p\_b} - q_{r.b\_a_2} - q_{l,b\_amb}$$

2- Energy balance equations for water storage tank, humidifier and dehumidifier

a- Water storage tank

$$m_{w_1} C_{p\_w} \frac{dT_{w_1}(t)}{dt} = M_{w_2}(t) C_{p\_w} T_{w_2}(t) - M_{mw}(t) C_{p\_w} T_{mw} - M_{w_1} C_{p\_w} T_{w_1}(t) - q_{w_1\_amb}$$

b- Humidifier

$$M_a\left(h_{a_3}(t) - h_{a_2\_e}(t)\right) = M_{w_1} C_{p\_w} T_{w_1}(t) - M_{w_2}(t) C_{p\_w} T_{w_2}(t)$$

c- Dehumidifier

$$M_a\left(h_{a_3}(t) - h_{a_4}(t)\right) = M_{w_3} C_{p\_w}(T_{w_4}(t) - T_{w_3}) + M_c(t) C_{p\_w} T_{w_5}(t)$$

Note: other basic and empirical relations for computing other parameters such as heat transfer rates, heat transfer coefficients, physical properties…etc are illustrated in detail in references [4&7].