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Theoretical and experimental investigations on solar distillation of IZTECH campus area seawater

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Abstract

The world demand for potable water is increasing steadily with growing population. Desalination using solar energy is suitable for potable water production from brackish and seawater. In this paper, we present a theoretical and experimental study of solar distillation in a single basin constructed at İzmir Institute of Technology Urla Campus. The still has a base area of 2100 mm × 700 mm with a glass cover inclined at 38°. In order to obtain extra solar energy, an aluminium reflector (2100 mm × 500 mm) is also assembled to the still. We model the still and conduct its energy balance equations under minor assumptions. We take into account the temperatures of glass cover, seawater interface, moist air and bottom in theoretical calculations and measurements. The comparison of the theoretical and experimental results highlights the benefits of the proposed model of the still and the efficacy of its energy balance equations.

Keywords: Solar still; Solar still with reflector; Solar energy; Solar desalination; Energy balance equations

1. Introduction

Desalination is a process that removes dissolved minerals from seawater, brackish water, or treated wastewater. It has become important in providing a solution to the problem of decreasing fresh water resources.

Today, the demand for fresh water is continuously increasing due to industrial development, intensified agriculture, improvement of life standard and increase of the world population. The amount of potable water involves only about 3% of the world water and it is not evenly distributed on the earth.

Solar desalination is a process where solar energy is used to distill fresh water from saline, brackish water. It is particularly important for locations where solar intensity is high and there is a scarcity of fresh water.

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Solar desalination has taken much attention from many researchers over the last decades. For example, Minasian et al. [1] studied the productivity of the conventional basin type solar still by using a stainless steel cylindrical parabolic reflector. The reflector concentrates the incident solar radiation on the black outside surface of a tray located on the focal line of the reflector. Results of the study showed that the productivity of the new proposed still were 25–35% greater than the productivity of the conventional basin type solar still. On the other hand, Elkader represented the experimental results carried out with a solar still with inclined evaporating yute to study the effects of air gap, base slope angle and glass cover slope angle on the performance of the still [2]. In order to investigate the parameters involved in the still, three models have been designed, manufactured and tested against some experimental measurements on a still having 1 m × 1 m basin area. The models have been designed in a way that it can give different base slope angle and glass slope angle. A modeling and experimental research of solar distillation applied to the Sahara arid regions was presented by Bouchekima et al. [3].

In this paper, we study a solar still with and without reflector. We derive the amount of water both theoretically and experimentally. The still has a 2100 mm × 700 mm base area, and a glass cover inclined at 38°. We decided about the angle of the glass cover, taking into account the location of the experiment, i.e. for March and April when the experiment is performed, the optimum slop angle maximizing the absorption of the solar radiation is approximately 38° [6].

We first conducted a theoretical model of the solar still under consideration to describe the energy balances for the glass cover, seawater interface, black plate at the bottom and overall still and also to find still productivity. We also assembled an aluminium reflector (2100 mm × 500 mm) in order to obtain extra solar energy and compare the effect of the reflector on the still productivity. The main contribution of this paper is, therefore, the energy balance equations of the still. To our best knowledge, our study seems more comprehensive than other similar studies in the literature.

We performed various experiments in order to measure and record the temperatures of the glass cover, seawater inside the still, seawater interface, inside moist and ambient air and distilled water. The comparison of theoretical and experimental results highlights the superiority of the model of the still.

The organization of the paper is as follows: In section 2, we introduce the theoretical treatment of the solar still. Subsections 2.1, 2.2, 2.3 and 2.4 explain how the energy balance equations are developed. We present, in section 3, the details of the solar still used in the experiments. The explicit solutions of the equations, measurements and their comparison are given in section 4. Finally, we conclude in section 5.

2. Theoretical treatment of a solar still

In a simple solar still, solar radiation passes through the glass cover. This solar energy is almost entirely absorbed by the black cover on the basin while it is partially absorbed in the thin seawater layer. Thus, seawater and basin are heated by the solar energy. The heat is conducted from the black surface into the seawater and the temperature of the seawater increases. Vaporization takes place at the interface, seawater surface and air inside of the solar still, at the interface temperature, t_i. Interfacial area, seawater surface is semi permeable. A plane is called semipermeable when the mass flux of one component is zero. Such a plane is, for instance, the surface of water, which evaporates into an adjoining air stream since water but no air passes through the surface. Thus, at the interface, the saturated air at the temperature, t_i, is transported by diffusion due to the partial pressure difference and convection

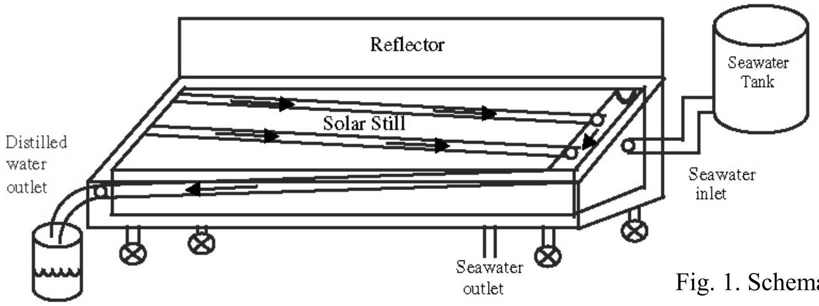

Fig. 1. Schematic view of solar still.

due to the natural convection of the humid air from the interface into the air inside of the solar still with the temperature, t_r. In steady state or quasi-steady state conditions, the air inside of the solar still is also saturated at the temperature, t_r. Therefore, the humid saturated air inside of the solar still will condense at the glass cover, which has a lower temperature, t_g then the air temperature inside solar still, t_r. Heat of condensation heats the glass cover. The glass cover is in heat exchange with surrounding and air inside of the still by convection and radiation. The condensate flows down, collecting along the glass cover and then in a channel at the end of the glass cover on the south side of the still. Finally, it collects in a storage bottle outside of the still. The schematic view of the solar still can be seen in Fig. 1.

In order to be able to calculate the daily produced condensed water, we apply the energy balance method by making the following assumptions:

- The whole system is in a quasi-steady state condition.
- Heat loss by radiation from the circumferential area is neglected.
- At the base of the still, temperatures of the walls equal to the water temperature, and the water temperature is the average of the interface temperature, t_i and the bottom temperature, t_b.
- The wind speed is assumed to be constant during the experiment.

The performance of a solar still is generally expressed as the quantity of water evaporated by unit area of the basin in one day, i.e. cubic meters or liters of water per square meter of the basin area per day. This performance of a solar still can be predicted by deploying the energy and mass balance equations on the various components of the still. The whole system is in a quasi-steady state condition and the temperatures are assumed not to change in one hour interval of time. The energy balance equations for the whole still, glass cover, seawater interface and black plate at the bottom will be given next.

2.1. Energy balance for the whole still

We first consider the energy balance of the whole still. The energy input to the still and the heat transfer from the still to the atmosphere are illustrated in Fig. 2. Hence, the energy balance equation can be written as

$$I^*_s A_g = I^*_s r_g A_g + q^*_{g,s} A_g + q^*_{h,g} A_g + q^*_{k,air} A_{k,air} + q^*_{k,l} A_{k,l} + q^*_{k,b} A_{k,b} + (m^*_{cw} h_{sat,g}) \text{ [J/s]} \quad (1)$$

$$I^*_s A_g/A_b = I^*_s r_g A_g/A_b + q^*_{g,s} A_g/A_b + q^*_{h,g} A_g/A_b + q^*_{k,air} A_{k,air}/A_b + q^*_{k,l} A_{k,l}/A_b + q^*_{k,b} A_{k,b}/A_b + (m^*_{cw} h_{sat,g})/A_b \text{ [J/m}^2\text{ s]} \quad (2)$$

In Eqs. (1) and (2), I^*_s is the solar radiation intensity, r_g is the reflectivity of the glass cover for visible light, $h_{sat,g}$ is the enthalpy of water at saturation temperature t_g, m^*_{cw} is the mass flow

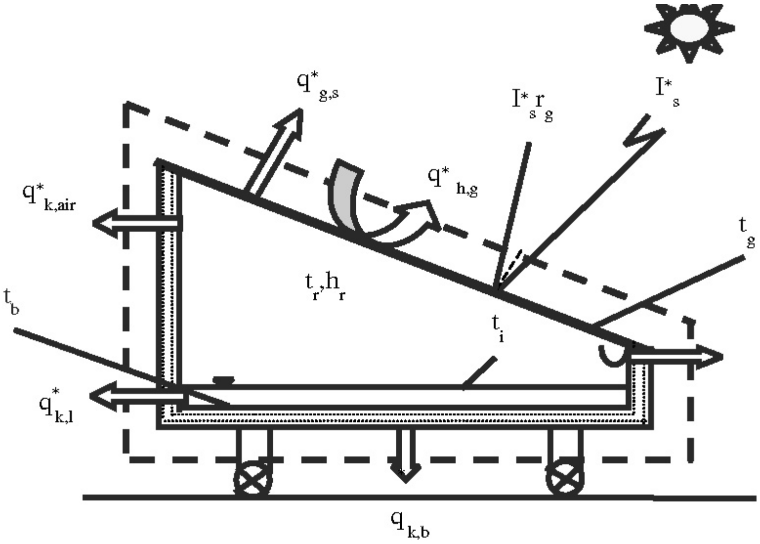

Fig. 2. Illustration of the overall energy balance.

rate of condensed water, A_g is the glass cover surface area and A_b is the bottom area covered by the seawater. By considering the heat transfer from the cover to the atmosphere by convection, we derive

$$q^*_{h,g} = h_g(t_g - t_a) \tag{3}$$

where t_g is the glass temperature and t_a is the ambient temperature. h_g is the convective heat transfer coefficient and is given by the following formula [4]:

$$h_g = 5.7 + 3.8w \tag{4}$$

where the forced convection coefficient depends on the wind velocity, w (m/s). Radiative heat transfer from the glass cover to the atmospheric air is given as

$$q^*_{g,s} = \varepsilon_g C_s[(T_g/100)^4 - (T_{sky}/100)^4] \tag{5}$$

where ε_g is the emissivity of the glass for infrared radiation, C_s is 5.667 W/m^{2}K^4, T_{sky} is the sky temperature and T_g is the glass cover temperature in K. C_s is derived from Stefan-Boltzmann constant such that $C_s = \sigma 100^4$ [6]. In practice, we may assume that the average sky temperature during operation hours is 20°C below the ambient temperature, i.e. $T_{sky} = T_a - 293$ K [5]. The conductive heat transfer from the bottom to the atmosphere can be formulated as

$$q^*_{k,b} = k_b(t_b - t_a) \tag{6}$$

where

$$\frac{1}{k_b} = \frac{1}{h_i} + \sum\frac{\delta_i}{\lambda_i} + \frac{1}{h_a}$$

considering the heat transfer from the circumferential area of the still by conduction. From inside moist air to the atmosphere,

$$q^*_{k,air} = k_r(t_r - t_a) \tag{7}$$

where

$$\frac{1}{k_r} = \frac{1}{h_r} + \sum\frac{\delta_i}{\lambda_i} + \frac{1}{h_a}$$

From liquid to atmosphere,

$$q^*_{k,l} = k_l(t_l - t_a) \tag{8}$$

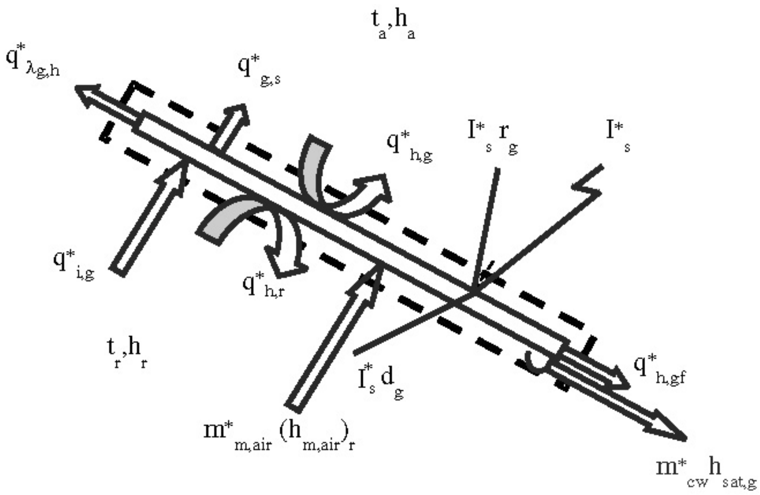

Fig. 3. Energy balance for glass cover.

where

$$\frac{1}{k_l} = \frac{1}{h_i} + \sum \frac{\delta_i}{\lambda_i} + \frac{1}{h_a}$$

and $t_l = (t_i + t_b)/2$ where t_l is the liquid seawater temperature.

2.2. Energy balance for glass cover

By following Fig. 3, the energy balance equation of glass cover is written as

$$I^*_s A_g + q^*_{i,g} A_g \cos\beta + q^*_{h,r} A_g + m_{m,air} h_{m,air,r} = I^*_s r_g A_g + I^*_s d_g A_g + q^*_{g,s} A_g + q^*_{h,g} A_g + q^*_{\lambda,gh} A_{\lambda,gh} + q^*_{h,gf} A_{h,gf} + m^*_{cw} h_{sat,g} \text{ [J/s]} \tag{9}$$

$$I^*_s A_g/A_b + q^*_{i,g} A_g/A_b \cos\beta + q^*_{h,r} A_g/A_b + (m_{m,air} h_{m,air,r})/A_b = I^*_s r_g A_g/A_b + I^*_s d_g A_g/A_b + q^*_{g,s} A_g/A_b + q^*_{h,g} A_g/A_b + q^*_{\lambda,gh} A_{\lambda,gh}/A_b + q^*_{h,gf} A_{h,gf}/A_b + (m^*_{cw} h_{sat,g})/A_b \text{ [J/m}^2\text{s]} \tag{10}$$

where $q^*_{i,g}$ is the radiative heat flux from the water surface to the glass, $q^*_{h,r}$ is the convective heat flux from the water surface to the glass and $h_{m,air,r}$ is the enthalpy of the water at saturation temperature t_r and $m_{m,air}$ is the mass flow rate of the moist air. $q^*_{\lambda,gh}$ and $q^*_{h,gf}$ are the heat losses from the back and front borders of the glass and they can be neglected for the sake of clarity.

Between the optical permeable and non-permeable walls (water surface is assumed to be a non permeable wall because transmittance of water equals to zero for infrared radiation), radiative heat transfer from the water to the glass is [6]

$$q^*_{i,g} = \frac{\left(\left(\frac{T_i}{100}\right)^4 - \left(\frac{T_g}{100}\right)^4\right) + \left((1-\varepsilon_g)/\varepsilon_g\right) d_g \left(\frac{T_i}{100}\right)^4}{\frac{1}{\varepsilon_g c_s} + \frac{1}{\varepsilon_w c_s} - \frac{1}{c_s} + \frac{1-\varepsilon_w-\varepsilon_g}{\varepsilon_w \varepsilon_g c_s} d_g} \tag{11}$$

where ε_w is the emissivity of water and d_g is the transmittance of glass for infrared radiation.

Convective heat transfer from the water to the glass is

$$q^*_{h,r} = h_r (t_r - t_g) \tag{12}$$

where h_r is the convective heat transfer coefficient. According to Churchill, Nusselt number is formulated as [7]

$Nu = 0.766\ [Ra\, f_2(Pr)]^{1/5}$ for $Ra\, f_2(Pr) \leq 7\times10^4$ (13)

and

$Nu = 0.15\ [Ra\, f_2(Pr)]^{1/3}$ for $Raf_2(Pr) \geq 7\times10^4$ (14)

where $f_2(Pr) = [1+(0.322/Pr)^{11/20}]^{-20/11}$.

It can be tabulated for different Prandtl numbers as

Pr	0.01	0.7	7	100	∞
$f_2(Pr)$	0.024	0.401	0.736	0.927	1

Grashof number, Gr, is given by the following formula:

$$Gr = \frac{g\ell^3}{\nu^2}\beta\Delta T \quad (15)$$

where g is the gravitational acceleration in m/s^2, ν is the momentum diffusivity in m^2/s, ℓ is the characteristic length in m, and it can be calculated by

$$\ell = \frac{A}{U} \quad (16)$$

where A is the surface area and U is the circumference of the participating heat transfer area projected in down stream direction. Rayleigh number is calculated by $Ra = GrPr$. Convective heat transfer coefficient can be calculated by

$$h_r = \frac{Nu\lambda}{\ell} \quad (17)$$

where λ denotes the thermal conductivity of air [7].

2.3 Energy balance for seawater interface

By referring to Fig. 4, the energy balance for seawater interface can be written as

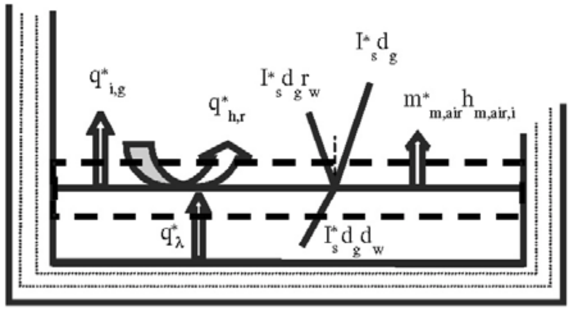

Fig. 4. Heat transfer modes for seawater interface.

$$I^*_s\, d_g\, A_g + q^*_\lambda\, A_b = I^*_s\, d_g\, r_w\, A_b + I^*_s\, d_g\, d_w\, A_b + q^*_{i,g}\, A_b + q^*_{h,r}\, A_b + m_{m,air}\, h_{m,air,i}\ \text{[J/s]} \quad (18)$$

$$I^*_s\, d_g\, A_g/A_b + q^*_\lambda = I^*_s\, d_g\, r_w + I^*_s\, d_g\, d_w + q^*_{i,g} + q^*_{h,r} + (m_{m,air}h_{m,air,i})\,/A_b\ \text{[J/m}^2\text{s]} \quad (19)$$

where d_w is the transmittance of water, r_w is the reflectivity of water for visible light. $h_{m,air,i}$ is the enthalpy of water at saturation temperature t_i and $m_{m,air}$ is the mass flow rate of the moist air. The conductive heat transfer from the bottom to the seawater interface is

$$q^*_\lambda = \lambda_w/\delta_w(t_b - t_i) \quad (20)$$

and the heat transfer from seawater interface to the glass cover is

$$q^*_{h,r} = h_r\,(t_i - t_r) \quad (21)$$

2.4. Energy balance for black plate

According to Fig. 5, the energy balance for black plate can be written as

$$I^*_s\, d_g d_w\, A_b - I^*_s\, d_g\, d_w r_b\, A_b = q^*_\lambda\, A_b + q^*_{k,b}\, A_b + q^*_{c,b} A_b\ \text{[J/s]} \quad (22)$$

$$I^*_s\, d_g\, d_w\, A_b/A_b - I^*_s\, d_g\, d_w\, r_b\, A_b/A_b = q^*_\lambda\, A_b/A_b + q^*_{k,b}\, A_b/A_b - q^*_{c,b} A_b/A_b\ \text{[J/m}^2\text{s]} \quad (23)$$

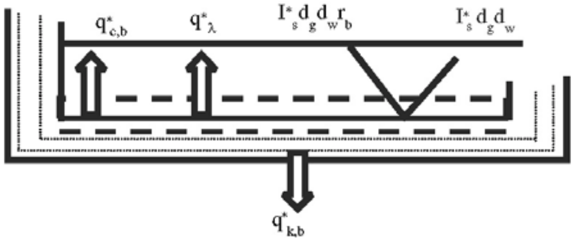

Fig. 5. Energy balance for black plate.

where r_b is the reflectivity of the black plate. $q^*_{c,b}$ is the convective heat transfer from bottom to the seawater, i.e.

$$q^*_{c,b} = h_{cb}(t_b - t_i) \quad (24)$$

Since the seawater depth in the still is very thin, the convective heat transfer from the black plate to the seawater can be neglected. The mass transfer coefficient is

$$h_M = \frac{h}{\rho c_p} \quad \text{[m/s]} \quad (25)$$

The mass flow rate of the evaporated water mass will be calculated according to the semi-permeable plane theory. For this purpose, first of all, we calculate the mass flux for a non-permeable plane according to the Reynolds analogy. By using this value, the mass transfer coefficient for semi permeable plane becomes

$$h_{M,h} = h_M \frac{P}{p_i - p_r} \ell n \frac{P - p_r}{P - p_i} \quad (26)$$

Hence, the mass flow rate of the evaporated water mass is [8]

$$m^*_{vapour} = h_{M,h} \frac{1}{RT}(p_{v,w} - p_{v,air})A_b \text{ [kg/s]} \quad (27)$$

R denotes the universal gas constant in Nm/kgK and T denotes the moist air temperature in K.

3. Solar still used in experimental study

We realized the basin type solar still in IZTECH Urla campus. The still was built with chrome sheet and galvanized iron. The bottom and sides of the chrome box are well insulated by polystyrene insulation material. The thickness of polystyrene insulation is 5 cm in east and west sides, 10 cm in other parts of the still and it is covered by a galvanized sheet. The thickness of the glass cover is 4 mm. A chrome channel is placed under the lower side of the glass cover to collect the condensed water. The channel is ended with a small plastic pipe in order to drain the fresh water into an external vessel. The glass is mounted at an angle of 38° to the solar still in order to ensure that the condensate will run down the glass in the condensate-collecting channel. Moreover, two aluminium collecting channels are stuck to the glass cover to collect condensate flowing down. The absorbing plate is also made of chrome in the dimensions of 2 m × 0.5 m. The effective area of the solar still is therefore 1 m^2. The absorbing plate of the solar still has 1-meter square area and it is painted with black dye. Three holes are also drilled to the solar still, one of which is for seawater inlet, the second one for condensed water outlet and the third one for discharging of remaining seawater. A metallic reflector 2100 mm × 500 mm in dimension (made of aluminium, having the reflectivity of about 90%) is attached to the still to concentrate the incident solar radiation.

Ni-Cr and K type thermocouples are located in different places of the still before fixing the glass cover. They are used to record the temperatures, such as outside glass cover, solar basin water, inside moist air temperature and seawater interface. A data logger allows us to record the temperature values. The long side of the solar still is aligned in north-south direction. Technical specifications of solar still are given in Table 1. Fig. 6 is a photograph of the solar still.

Fig. 6. Photograph of the solar still.

Table 1
Technical specifications of the solar still

Width	0.5 m
Length	2 m
Cover inclination	38°
Glass area	1.4 m^2
Base area	1 m^2
Glass depth	4 mm

4. Results and discussion

In this part, we present the theoretical and experimental results of the solar distillation in a single basin proposed in the previous sections.

As already explained, an aluminium reflector is assembled to the still and the effect of the reflector on the still productivity is examined. The numerical solution of the basic heat and mass transfer equations are established as well, in order to make comparisons with the experimental results.

In the theoretical part, we developed a computer program that implements appropriate numerical analysis methods in order to solve the energy balance equations. This program allows us to calculate the temperatures of the glass cover, the inside moist air, the bottom and the interface, depending on the amount of the radiation and various other parameters given as the input to the program. Temperature values also help us to evaluate theoretically the amount of evaporated water for a given solar radiation and ambient temperature. In this program, we utilize a gradient-based optimization method to solve four non-linear equations defined in the previous section. According to this method, in order to be able to find a solution set for our energy balance equations, we first derive the cost function E as follows:

$$E = f_1^2 + f_2^2 + f_3^2 + f_4^2$$

where

$$f_1(t_b, t_i, t_r, t_g) = 0,$$
$$f_2(t_b, t_i, t_r, t_g) = 0,$$
$$f_3(t_b, t_i, t_r, t_g) = 0,$$
$$f_4(t_b, t_i, t_r, t_g) = 0.$$

The above functions, f_1, f_2, f_3 and f_4 are obtained by assigning the known constant values in equations 2, 10, 19 and 23. Then, the unknown temperature values (glass cover temperature t_g, inside moist air temperature t_r, bottom temperature t_b and interface temperature t_i) are found by minimizing the function E according to the steepest descent method as follows:

$$t_b^+ = t_b - \frac{\partial E}{\partial t_b}\eta$$

$$t_i^+ = t_i - \frac{\partial E}{\partial t_i}\eta$$

$$t_r^+ = t_r - \frac{\partial E}{\partial t_r}\eta$$

$$t_g^+ = t_g - \frac{\partial E}{\partial t_g}\eta$$

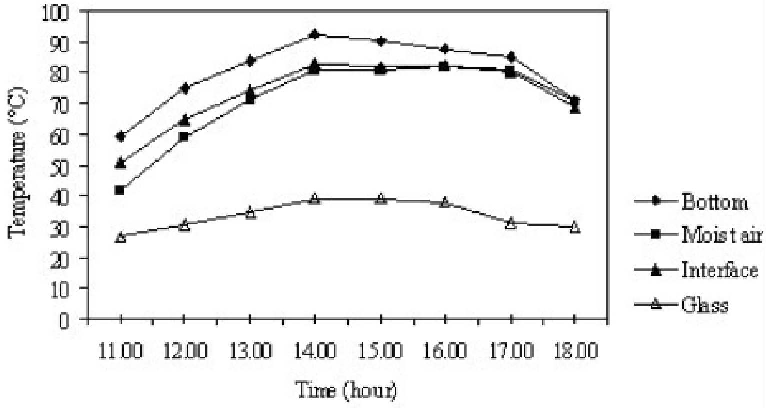

Fig. 7. Hourly variation of theoretical temperature values on solar still with reflector.

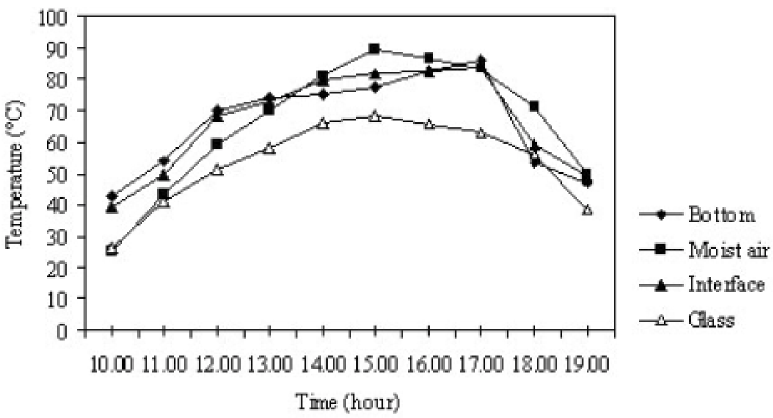

Fig. 8. Hourly variation of experimental temperature values on solar still with reflector.

where η denotes the step length. The solution set is approximated by applying the above formulae iteratively.

In the experimental part of the study, the temperature values and the amount of distilled water were recorded in each hour. The total solar radiation on tilted surface is 20.7 MJ/m^2 during the experiment and it was calculated from the meteorological measured solar radiation value on horizontal surface [4]. The reflector provides 6.2 MJ/m^2 extra energy as well. The amount of extra energy was calculated by using fictive sun method [6]. Thus, the total solar energy received during the experiment is 26.9 MJ/m^2. Fig. 7 presents the theoretical hourly variation of temperatures on the 29th day of April. These calculations were performed with known solar radiation and ambient air temperature.

Fig. 8 presents the hourly variation of the temperatures of the experiment. Hourly variation of the experimental and theoretical distilled water values is given in Fig. 9. According to the figure, the amount of distilled water calculated theoretically is 3.26 l/m^2 and the amount of distilled water measured experimentally is 3.040 l/m^2. The difference between the theoretical and experimental results is due to the leakage from the still. Figs. 10 and 11 present the hourly variation of theoretical and experimental temperatures on 31st of March on the solar still without reflector. In this experiment, the total solar energy coming from the sun is 21.45 MJ/m^2. The amount of distilled water obtained from the still without reflector was experimentally measured as 1.430 l/m^2.

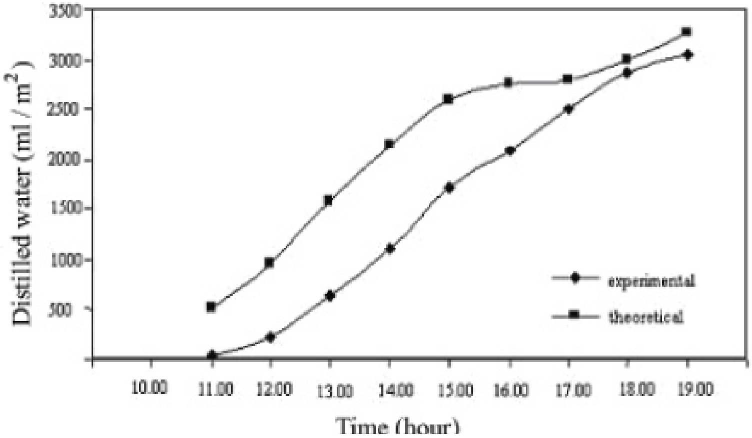

Fig. 9. Comparison of theoretical and experimental distilled water values on solar still with reflector.

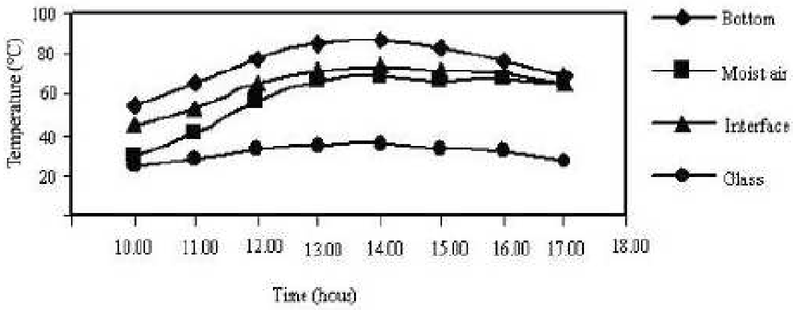

Fig. 10. Hourly variation of theoretical temperatures on solar still without reflector.

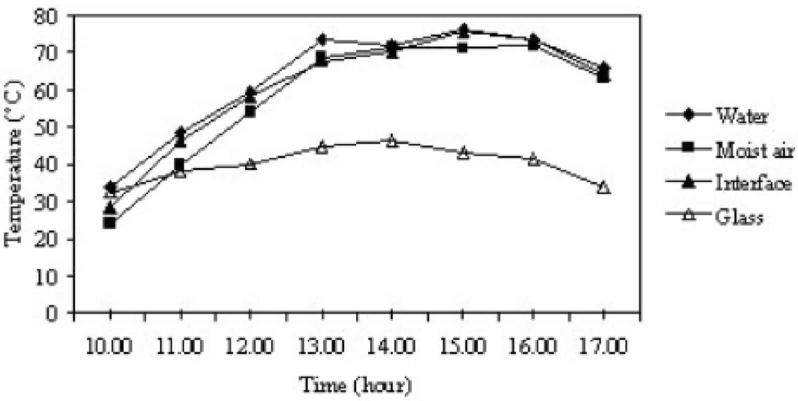

Fig. 11. Hourly variation of experimental temperatures on solar still without reflector.

The efficiency of a still without reflector can be calculated by the following equation:

$$\eta = \frac{\text{Water output} \times \text{latent heat of evaporation of water}}{\text{Daily solar radiation}}$$

$$= \frac{m^* \times L_v}{I_{s.t} \times A_g} \tag{28}$$

The efficiency can be found by using the following formula for the system with reflector:

$$\eta_R = \frac{m^* \times L_v}{I_{s.t} \times A_g + I_{s,b} \times A_{she}} \tag{29}$$

where m^* is the daily output (kg/m^{2}s), L_v is the latent heat of evaporation of water (kJ/kg) and $I_{s,t}$

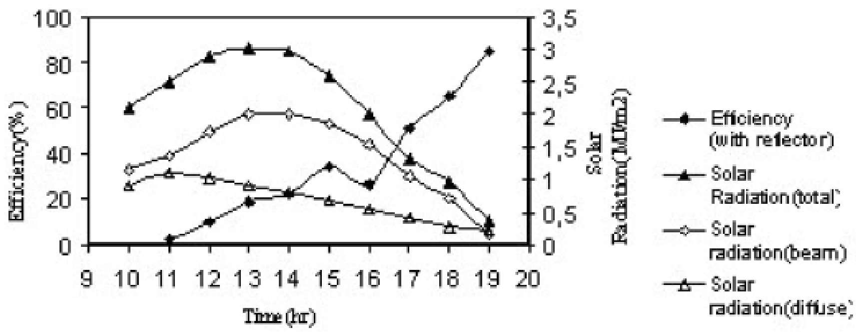

Fig. 12. Experimental hourly efficiency of still and variation of solar radiation for 29th of April.

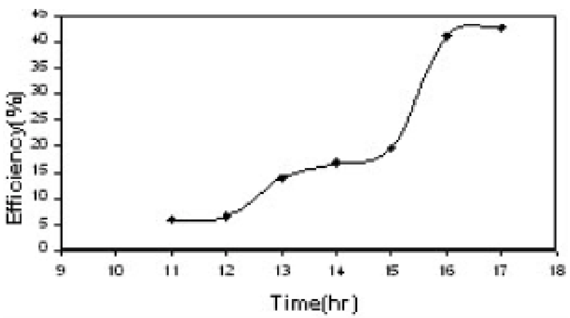

Fig. 13. Experimental hourly efficiency of still for 31st of March.

is the daily total solar radiation (kJ/m^{2}s), $I_{s,b}$ is the beam radiation reflected from the reflector (kJ/m^{2}s) and A_{she} is the effective shaded area on the glass cover. Fig. 12 shows hourly variation of efficiency for 29th of April. The efficiency is calculated by using Eq. (29).

The hourly variation of the still efficiency for 31st of March is shown in Fig. 13, which indicates that the efficiency of the still increases to a maximum value late in the afternoon. The efficiency is calculated by using Eq. (28) (for more explanation, please refer to [9]).

5. Conclusion

In this work, we developed a theoretical framework of a single basin solar still constructed in the university. This framework enabled us to establish the numerical solutions of the basic heat and mass transfer equations of the solar still. The still was equipped with a reflector and the effect of the reflector was studied as well. We compared the theoretical solutions with the experimental results. These comparisons show that the proposed model of the still is quite accurate and it provides satisfactory results. Therefore, we highlight that the developed model can be very useful for designing and analyzing the solar stills. More comparisons using more experiments that should be performed systematically are still of interest in order to prove the accuracy of the model, which may lead to major generalization of the proposed model.

6. Symbols

t_g — Temperature of glass cover, °C
t_b — Temperature of bottom, °C
t_r — Temperature of inside moist air, °C
t_i — Seawater interface temperature, °C
t_l — Temperature of the liquid at the bottom, °C
t_a — Temperature of ambient air, °C
T_i — Seawater interface temperature, K
T_g — Temperature of glass cover, K
T_{sky} — Sky temperature, K
I_s — Solar intensity, W/m^2

$q_{k,l}$ — Heat transfer from the liquid at the bottom into the atmosphere through the bottom area, W/m^2
$q_{k,b}$ — Heat transfer from the bottom into the atmosphere, W/m^2
$q_{k,air}$ — Heat transfer from the inside moist air through the circumferential area into the atmosphere, W/m^2
$q_{h,g}$ — Convective heat transfer from the glass cover into the atmosphere, W/m^2
$q_{g,s}$ — Net radiative heat transfer from the glass cover into the sky, W/m^2
d_g — Transmittance of glass
r_g — Reflectivity of glass
m^*_{cw} — Mass flow rate of condensed water, kg/m^{2}s
$m^*_{m,air}$ — Mass flow rate of moist air, kg/m^{2}s
h_r — Convective heat transfer coefficient at moist air, W/m^2 K
h_a — Convective heat transfer coefficient at ambient air, W/m^2 K
k_l — Overall heat transfer coefficient for heat transfer from the liquid at the bottom into the atmosphere through the bottom area, W/m^2 K
k_r — Overall heat transfer coefficient for heat transfer for heat transfer from the inside moist air through the circumferential area into the atmosphere, W/m^2 K
k_b — Overall heat transfer coefficient for heat transfer from the bottom into the atmosphere, W/m^2 K
A_b — Bottom area covered by the seawater, m^2
A_g — Glass cover surface area, m^2
$A_{k,air}$ — Circumferential area of the solar still covered by the inside moist air, m^2
$A_{k,l}$ — Circumferential area of the solar still covered by the seawater, m^2
w — Wind velocity, m/s
h_M — Mass transfer coefficient for non-permeable plane, m/s
$h_{M,h}$ — Mass transfer coefficient for semi-permeable plane, m/s
L_v — Latent heat of evaporation of water, J/kg

Greek

λ — Thermal conductivity, W/mK
ρ — Density, kg/m^3
β — Thermal expansion coefficient, K^{-1}
σ — Stefan-Boltzman constant, W/m^2 K^4

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