THEORETICAL AND EXPERIMENTAL INVESTIGATION ON THE INFLUENCE OF STILL GLASS COVER COOLING ON WATER PRODUCTIVITY

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Abstract

This paper presents a theoretical and experimental investigation on the optimization of a solar still. The theoretical approach is based on the simulation of the solar still system, the influenced parameters were isolated and their corresponding effects were numerically evaluated. According to these results, the optimization methodology is based on the concept of recuperation of the condensed heat. Two solar stills having the same dimensions were constructed. A descending film of salt water-cools the glass cover of the first still. The cooling water is then recycled and fed into the supply of the solar still. The other still is simple and conventional; it is used only for comparison and references purposes. Both stills are similar in geometry, and inclination angle. The theoretical and experimental results show that the productivity increases with cooling the glass cover. The experimental results show important increment in productivity, reaches in some cases 60%. The influence of cooling water flow rate was also investigated. The experimented investigation has shown the non-linear behavior of the flow rate on the performance.

The objective of the present research work is to reach an optimum system configuration in order to provide a simple and high effect solar distillation unit.

Keywords: Seawater Desalination, Solar Desalination, Water Desalination, Solar Still

Introduction

The increasing populations, coupled with the industrial and agricultural development of rural areas are creating an unbalance between supply and demand. The recourse to non-conventional water resource is a must in order to respond to this situation. The desalination of brackish and seawater is suitable solution for many cases. The desalination technologies have reached an advanced commercial maturation stage, which put it as a potential competitor to other low price water treatment technologies. Various types of desalination plants have been built and developed world wide, according to recent IDA inventory more than 17 millions liter/day is desalting around the world. However, most existing desalination units use fossil fuels as a source of energy.

Modern civilization is completely depending on the cheap and abundant energy. The economic wealth and material standards of living of a country are determined by the technologies and fuels consumption. The cost of energy has shown to be a major component of the cost of distilled water. The actual low price of fossil energy sources, have encouraged the dependence on fossil energy. The solar has a definite advantage over the fossil, which is its adaptability for small stand-alone units in rural and isolated areas.

Solar stills, in many cases, might be an ideal source of fresh water for drinking and agriculture, in arid isolated zones. The first conventional solar still (4700 m^2 in area) was built in 1872. Since then a great number of manuscript, as well as excellent monographs has been published [1], [2]. Recently [3], [4], [5] a different model has been presented in order to model the connective heat transfer and consequently the productivity of still. The basic idea is how to recycle the recuperated heat from the vapor condensation. Gyorgy et al. [7] has presented a schema based on the air blown method for heat recycle. In the present investigation the improvement methodology is based upon the cooling of collector cover with a thin layer of saline water before its introduction into the solar still.

The effect of cooling water flow rate on productivity was also studied. A simple mathematical model is used for the solution of the general equation. A finite difference numerical technique is used for the solution of the governing equations. A comparison between the theoretical model results and experimental results is presented. The results have shown reasonable accordance between theoretical and experimental finding.

The objective of this research work is to improve the solar still productivity through the modification of the system configuration, based on energy recovery approach.

Formulation of the Problem (Theoretical Model)

The theoretical model depends on the energy balance of each part of the used system. The following assumptions have been assumed:

1- There is no leakage in the still.
2- The temperature gradient along the cold water thickness, the glass covers thickness and water depth has been assumed negligible.

Energy balance of cold water and glass cover are calculated from the following equations:

$$M_c\left(T_{wco}-T_{wci}\right)+h_2\left(T_g-T_{wc}\right)-h_3\left(T_{wc}-T_a\right)+\alpha_c H_s=0 \tag{1}$$

$$M_g\frac{dT_g}{dt}=\tau_1 H_s+h_1\left(T_w-T_g\right)-h_2\left(T_g-T_{wc}\right) \tag{2}$$

where:

$$\tau_1=\left(1-R_C\right)\left(1-\alpha_C\right)\alpha_g \tag{3}$$

h_1 is combined convective, radiative and evaporative heat transfer coefficient from water to glass cover still, (W/m^2.C), h_2 is heat transfer coefficient from glass cover to cold water, (W/m^2.C) and h_3 is heat transfer coefficient from cold water to ambient air, (W/m^2.C).

Energy balance of water content in the still basin can be expressed as:

$$M_w\frac{dT_w}{dt}=\tau_2 H_s-h_1\left(T_w-T_g\right)+h_3\left(T_b-T_w\right) \tag{4}$$

where:

$$\tau_2=\left(1-R_C\right)\left(1-\alpha_C\right)\left(1-\alpha_g\right)\alpha_w \tag{5}$$

H_s = total solar radiation on the still, (W/m^2)

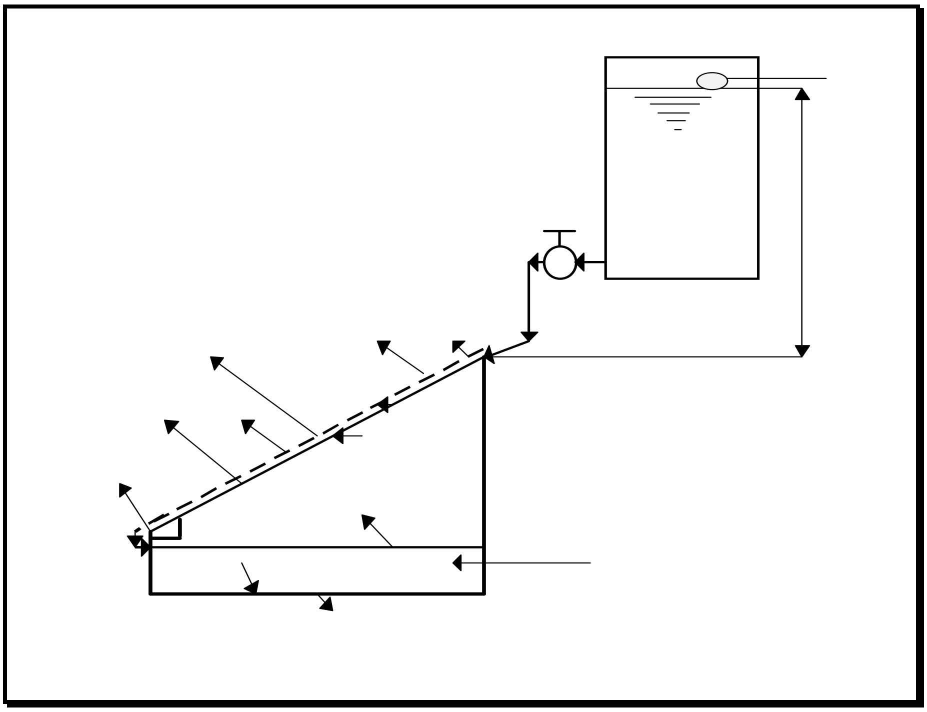
Energy balance of the basin liner can be expressed as:

$$\tau_3 H_s=h_4\left(T_b-T_w\right)+h_b\left(T_b-T_a\right) \tag{6}$$

where:

$$\tau_3=\left(1-R_C\right)\left(1-\alpha_C\right)\left(1-\alpha_g\right)\left(1-\alpha_w\right)\alpha_b \tag{7}$$

$$\frac{1}{h_b}=\frac{t_1}{K}+\frac{1}{h_5} \tag{8}$$



Flow diagram of the solar distillation system

h_4 is heat transfer coefficient from the saline water to the basin liner, (W/m^2.C), h_5 is heat transfer coefficient from the bottom of solution to ambient air, (W/m^2.C) and h_b is overall heat transfer coefficient from the bottom to the ambient, (W/m^2.C).

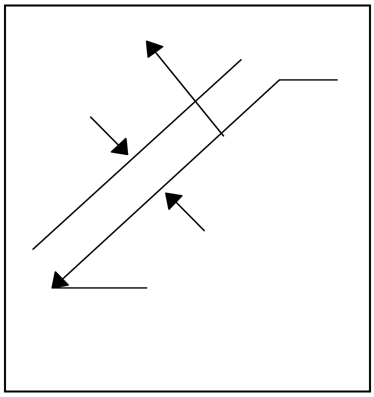
From equations (4) and (7)

$$M_w \frac{dT_w}{dt} = \tau_4 H_s - h_1\left(T_w - T_g\right) - U_b\left(T_w - T_a\right) \tag{9}$$

where:

$$\tau_4 = \tau_2 + U_b\left(\frac{t_1}{K} + \frac{1}{h_5}\right)\tau_3 \tag{10}$$

$$\frac{1}{U_b} = \left(\frac{1}{h_4} + \frac{1}{h_b}\right) \tag{11}$$



To calculate the thickness of the film of the cold water, assuming laminar flow

$$V_x = \frac{g\sin\theta}{\upsilon}\left[t.y - \frac{y^2}{2}\right]$$

$$Q = \int_0^t V_x\, b.dy = \frac{g\sin\theta}{\upsilon}.\frac{t^3}{3}.b \qquad (12)$$

The thickness is calculated from the following relation:

$$m = \frac{\rho g\sin\theta}{\upsilon}.\frac{t^3}{3}.b \qquad (13)$$

where *m* is the cold water flow rate, (kg/s), *t* is the thickness of cold water and *b* is the still width

The absorbtivity and reflection of the cold water will be changed with the film thickness of the cold water; there are poor information about the small thickness. The only available data is presented in Table (1).

Table (1) Absorbtivity and reflection of the solar flux

Water thickness, t (m)	0.1	0.5	1.0	1.5
Absorbtivity and reflection of the solar flux (kW/m^2)	0.541	0.384	0.322	0.301

In the present work, the cold water layer thickness is less than 0.1 m. A linear relation is assumed from Table (1), in order to retrieve the necessary data of absorbitivity and reflection for small thickness ranging from 0 to 0.1m. It is assumed that the absorbtivity and reflection at 0, thickness equal 0.95. This relation can be obtained from the previous table.

$$\alpha_c = 0.95 - 4.1.\,t \qquad (14)$$

The heat transfer coefficient from the water surface to the glass cover (h_1) is calculated from the following equation, [7]:

$$h_1 = h_c + h_r + h_e \qquad (15)$$

From references [6], [7], [8], the following relation are presented

$$h_c = 0.8831\left[\left(T_w - T_g\right) + \left(\frac{P_w - P_g}{0.265 - P_w}\right)\left(T_w + 273\right)\right]^{\frac{1}{3}} \qquad (16)$$

$$h_e = 0.0061\left[\left(T_w - T_g\right) + \left(\frac{P_w - P_g}{0.265 - P_w}\right)\left(T_w + 273\right)\right]^{\frac{1}{3}}\frac{\left(P_w - P_g\right)L}{T_w - T_g} \qquad (17)$$

$$h_r = \sigma.F_{wg}.\left[\left(T_w + 273\right)^4 - \left(T_g + 273\right)^4\right] \qquad (18)$$

where h_e is evaporation heat transfer coefficient, (W/m^2.C), h_c is convection heat transfer coefficient, (W/m^2.C) and h_r is radiative heat transfer coefficient, (W/m^2.C).

The heat transfer from glass cover to cold water is calculated from the following relation after [9]:

$$N_u = 0.664 R_e^{0.5} P_r^{0.33} \tag{19}$$

The heat transfer from cold water to ambient air is calculated from this the following relation after [7]:

$$Nu = c(Gr.\mathrm{Pr})^n \tag{20}$$

The value of c and n depend on the value of Gr.

1- for $Gr < 10^3$, $c = 1$, $n = 0$,
2- for $10^3 < Gr < 3.2 \times 10^5$, $c = 0.21$, $n = 1/4$,
3- for $3.2 \times 10^5 < Gr < 10^7$, $c = 0.075$, $n = 1/3$.

Productivity can be calculated from the following equation:

$$D = \frac{q_e}{L} = \frac{h_e(T_w - T_g)}{L} \tag{21}$$

Solution of the theoretical model:

The cold water temperature, glass temperature, still water temperature and solar radiation change with time. The finite difference numerical method has been used for their calculation. In this calculation, the time interval (Δt) has been taken as 0.5 hour. The heat transfer coefficient is calculated using the initial values of temperatures (T_{wci} = T_g = 20°C and T_w = 20.01°C), then calculate the new values from the model and then calculate the new heat transfer coefficient. For each time interval the amount of water in the basin is considered as the initial amount of fed water due to the cooling water recycled fed into the supply of the solar still.

Experimental set up

Two single basins solar still are constructed. The two solar stills have the main shape dimensions and construction. One of this still is conceived in such a manner to allow the flow of thin layer of water on the outer surface. The flow is then recycled and fed into the basin inlet. The basin area of each still is 0.5 x 1 m; heights of the front and back walls are 0.24 m and 0.5 m, respectively. The glass cover slope was 15°. The water thickness in the two stills is 0.03 m. A rectangular 4 mm thick glass sheet was fitted in an iron farm. The rubber gasket was used as tightened between glass cover and frame to avoid vapor leak.

The cold water flows over the glass cover was kept uniform and constant with the help of a regulator and a constant head tank 0.35 m above the level of still. The dimensions of the tank are square base of 0.63*0.63 m and 0.72 height. The main tank has the same dimension. The evaporated water contacts the glass cover, and condenses then runs down, and collected. An inclined rectangular channel was welded at the short side wall of each still from inside surface to collect the condensate. The two stills were insulated from the two sides and bottom by using a wood powder with a thickness 0.05 m. A wooden box was used to contain the still and insulation. The glass cover temperatures, water temperatures, cold water inlet and exit temperatures and ambient air temperature, also solar radiation intensity measured of two stills temperatures are measured at different. The stills and all system were manufactured in the Faculty of Engineering, Tanta University, Egypt.

Photograph of the system used

Results and Discussion

The theoretical results as well as the experimental results were presented together through Figures 1 to 14. The figures from 1 to 7 show the variation of glass cover temperature, cold water temperature, and brine water temperature during the day without glass cover cooling and with glass cooling, at different amount of cold water descending over the glass cover. The amount of cold water changed from 0 (no glass cover cooling) to 0.12 kg/s. The experimental tested were carried out at no cooling, at 0.01 kg/s and at 0.03 kg/s. It can be seen that the temperatures of glass and cold water decrease with increase of cold water flow rate, also the brine temperature decreases with increasing the cold water flow rate. It can be observed that the temperature difference between the brine temperature and glass temperature increase with increasing the amount of flow rate of cold water. The following table shows this difference at deferent amount of cold water at noon. Comparison of the experimental data is presented in Figures 1, 3, 4. There are some overestimation (average estimation is about 9%) for the water temperature and glass temperature. This deviation is due to the effect of change of the solar radiation and the condensate temperature also the effect of inlet cold water temperature.

Table (2) Temperature difference due to glass cooling

Cold water flow rate (kg/s)	0.0	0.005	0.01	0.03	0.06	0.09	0.12
Temperature difference (°C)	17	21	24	28.6	31	32	33

Table (2) shows that the increasing value of temperature difference due to glass cooling reached 200%, and the difference is increasing until 0.06 kg/s and then the increment value is small (asymptote). Also, it can be seen that the temperature difference between the cold water temperature and glass temperature decreases with increasing the cold water flow rate.

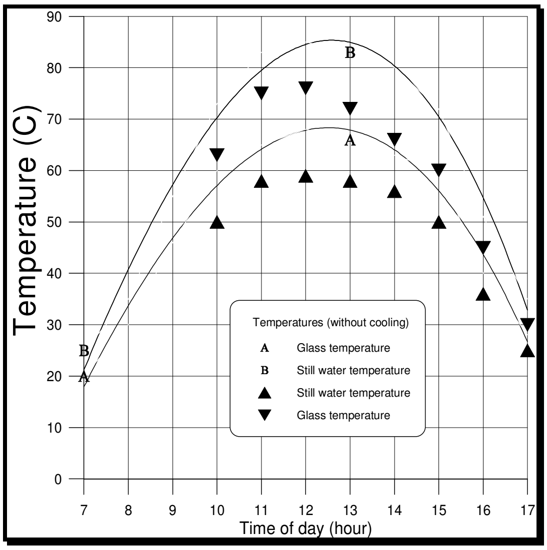


Fig. 1: Variation of temperature during the day at no glass cooling

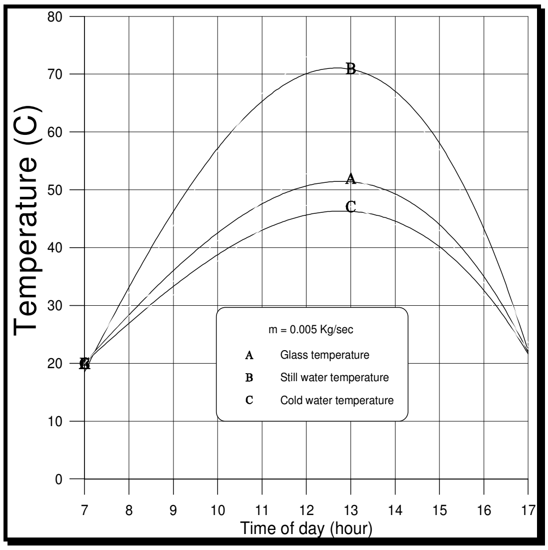


Fig. 2: Variation of temperature during the day at 0.005 kg/s cold water

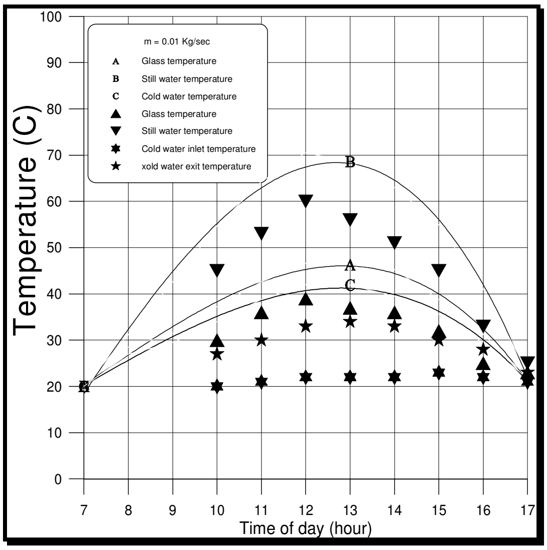


Fig. 3: Variation of temperature during the day at 0.01 kg/s cold water

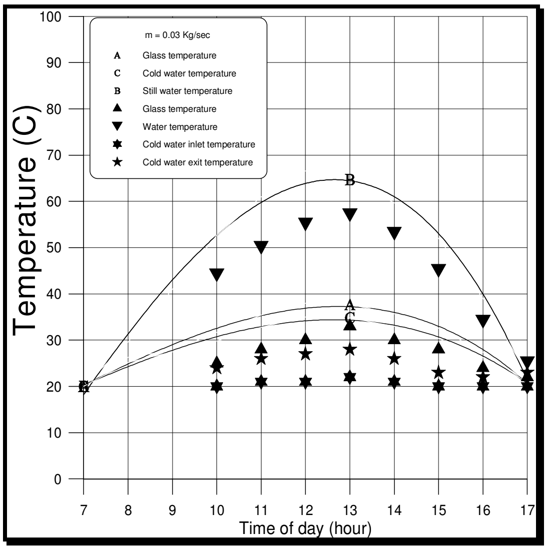


Fig. 4: Variation of temperature during the day at 0.03 kg/s cold water

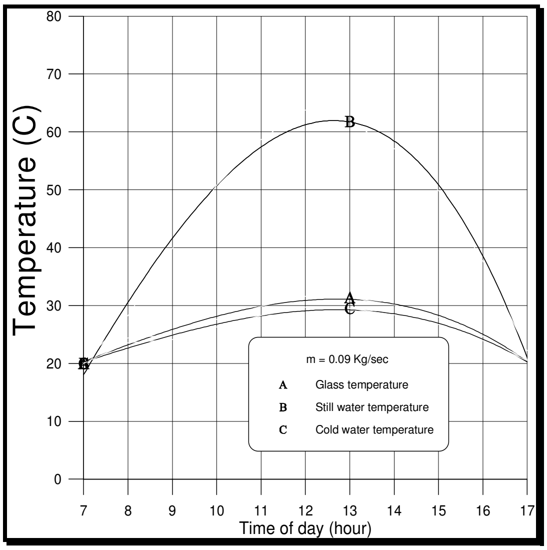


Fig. 5: Variation of temperature during the day at 0.06 kg/s cold water

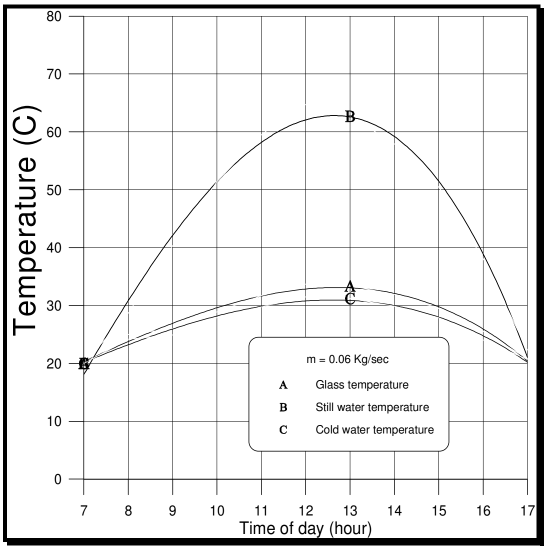
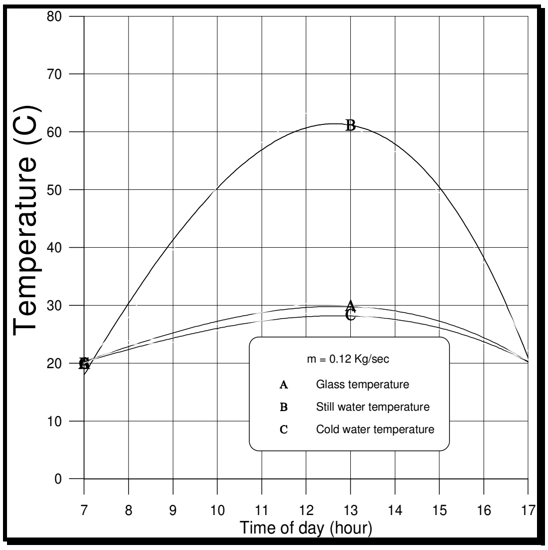


Fig. 6: Variation of temperature during the day at 0.09 kg/s cold water

Figure 8 shows the variation of the productivity during the day at different flow of cooled water. It can be observed that the maximum productivity without glass cover cooling reached to 0.38 $kg/m^2.hr$. The productivity increases with increment of the cold water flow. At cold water flow of 0.005 kg/s the productivity reaches 0.53 kg/hr. The productivity increases to reach its maximum at 0.59 kg/hr at flow rate of 1.2 kg/s.



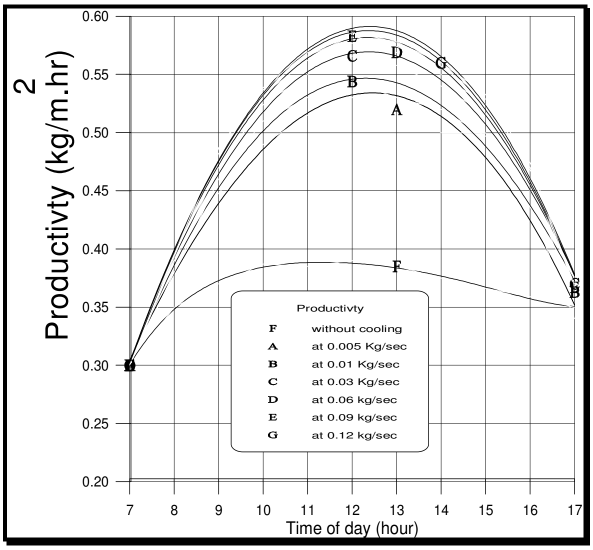


Fig. 7: Variation of temperature during the day at 0.12 kg/sec cold water

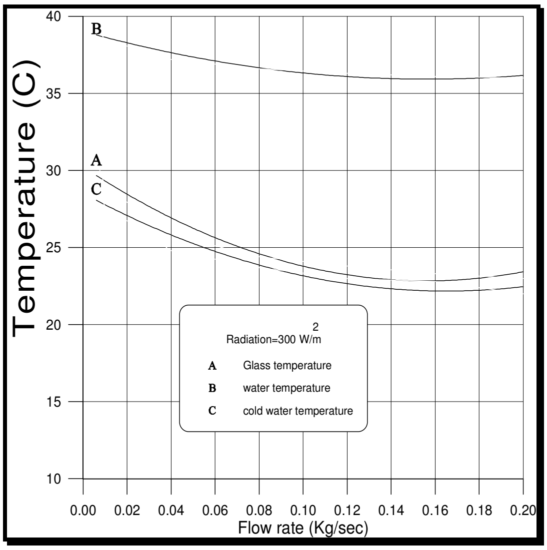
Fig. 8: Variation of productivity during the day at different flow rates

Figure 9 shows the variation of solar radiation during the day, these values of the radiation was used in the theoretical model of calculations.

Figures 10, 11, and 12 show the variation of temperatures at different solar radiations (300, 600, 900 W/m^2) for different flow rates of cold water. It can be seen from figures that the variation of the temperatures depends on the values of the flow rate and solar radiation. At the high values of flow rate (0.2 kg/s) the temperature difference reaches 13°C at solar radiation equals 300 W/m^2. The temperature difference at solar radiation 600 W/m^2 and 900 W/m^2 is the same value and equals 35°C. At cooling flow rates equal 0.04 kg/s, the temperature difference is 12°C at solar radiation equals 300 W/m^2, equals 21°C at 600 W/m^2, and equals 30°C at 900 W/m^2.

Figure 13 shows the variation of productivity with the flow of cold water at different values of solar radiation. It can be seen from the figure that the productivity increases with increment of the cold water flow rate until 0.12 kg/ s and then the increment is asymptotic.

Figure 14 shows the theoretical and experimental results for the different cases without glass cooling, at cold water flow rate 0.01 kg/s and at cold water flow rate 0.03 kg/s. It can be seen that there are small difference between the experimental and theoretical results. This discrepancy is explained by some experimental uncertainty, in fact not all of the distilled water is collected, some distilled water drops falling back into the basin still, it is difficult to estimate this amount.



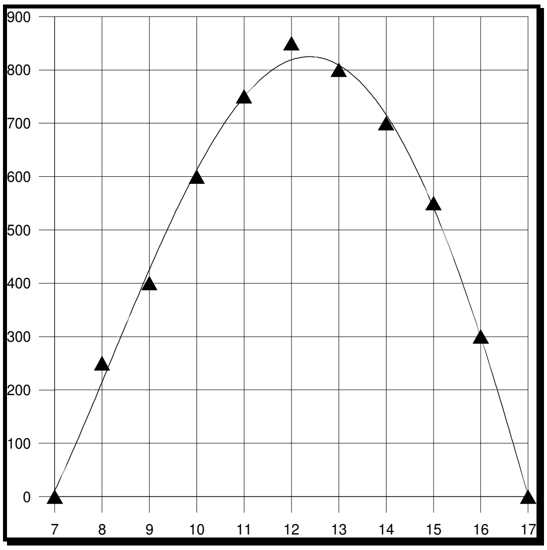
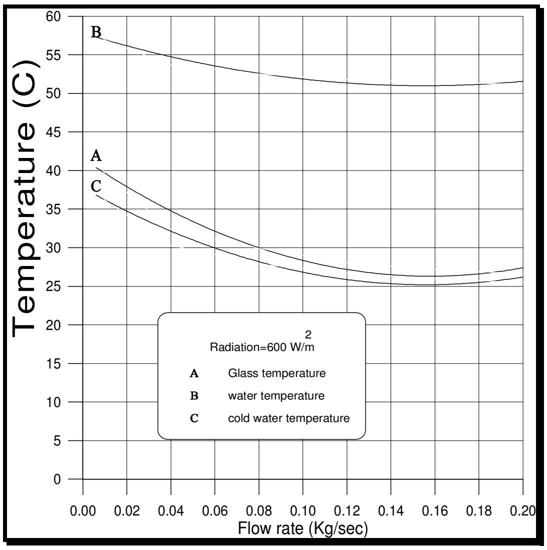


Fig. 9: Variation of solar radiation during the day.

Fig. 10: Variation of temperature with the flow rates.



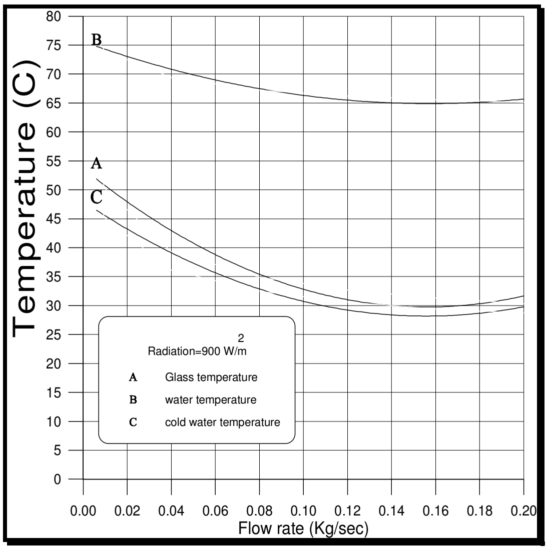


Fig. 11: Variation of temperature with the flow rates.

Fig. 12: Variation of temperature with the flow rates.

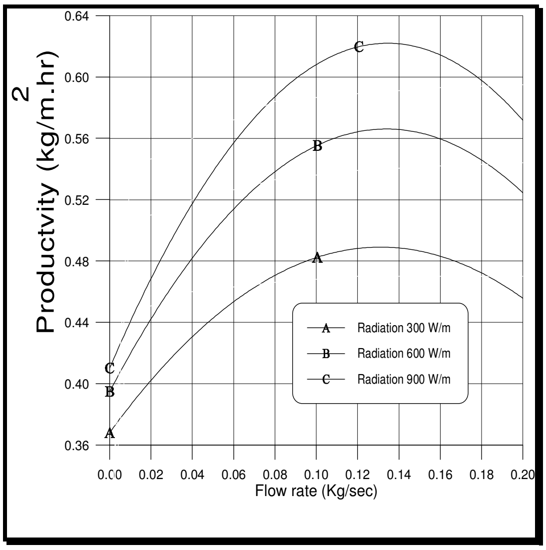


Fig. 13: Variation of Productivity with the flow rates at different solar radiation

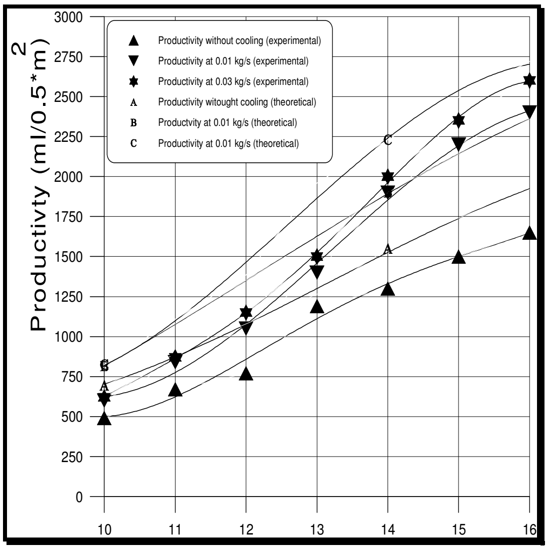


Fig. 14: Variation of productivity during the day at the different cooling flow rates

The productivity increases in a linear manner with cooling water flow rate. It reaches its maximum at 0.03 kg/s, then the productivity increment is not linear and reaches a steady state. This is certainly indicates that the system has attained its thermal stability and the increment of cold water flow rate will not be sensible regarding the cover temperature. In the optimum case the increment in productivity reaches 70% over the conventional system.

In fact the cover cooling has proved to be an adequate and simple tool towards the improvement of solar still. The theoretical model helps in forecasting the necessary cooling water flow in order to reach the optimum productivity. There is additional effort required to couple this model with turbulence, and convective heat transfer model inside the still, in order to improve the thermo-fluid conception of the system.

Conclusions

Based on the present investigation, the following recommendations and conclusions are offered:

1. The presented theoretical model predicts with a reasonable approximation the productivity of solar still.

2. The cover cooling technique has proved to be an adequate and simple tool towards the improvement of solar still productivity. The present experimental results show an increment in productivity up to 60%.

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Nomenclature

A = still surface area, (m^2)
D = daily output of the still per unit area, ($kg/m^2.hr$)
D_{eq} = Equivalent diameter (m)
F_{wg} = radiation shape factor from brine surface to inner cover surface (--)
L = latent heat of evaporation, (J/kg)
l = still length, (m)

M_c = heat capacity of cold water per unit area, (J/m^2.°C
M_w = heat capacity of water per unit area, (J/m^2.°C)
M_g = heat capacity of glass cover per unit area, (J/m^2.°C)
Nu = Nusselt number, (--)
q_e = heat transfer rate in the still by evaporation per unit cover area, (W/m^2)
p_w = partial pressure of water vapor at T_w,, (MN/m^2)
p_g = partial pressure of water vapor at T_g, (MN/m^2)
R_g = glass cover reflectivity
R_c = cold water reflectivity
R_w = water reflectivity
Ra = Rayleigh number, (--)
T_w = water temperature, (°C)
$T_{w\,c}$ = cold water temperature, (°C)
$T_{w\,ci}$ = inlet cold water temperature, (°C)
$T_{w\,co}$= outlet cold water temperature, (°C)
T_g = glass temperature, (°C)
T_b = basin linear temperature, (C)
t_1 = insulation thickness, (m)
ιg = transmativity of the glass, (--)
t = time, hr
V = velocity of cold water, (m/sec).
Gr = Grashof number, (--)
Pr = Prandtl number, (--)
α_w = water absorbtivity, (--)
α_g = glass cover absorbtivity, (--)
α_c = cold water absorbtivity, (--)
α_b = bottom surface absorbtivity, (--)
β = coefficient of thermal expansion, (K^{-1})
ρ = density of water, (kg/m^3)
σ = Stefan-Boltzman constant, 5.6697x10^{-8}, (W/m^2.C)