Thermal modelling of a double condensing chamber solar still: an experimental validation

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Abstract

In this communication, a thermal model of a double condensing chamber solar still (DCS) has been presented. The thermal model is based on energy balance equations for the different components of a double condensing chamber solar still (DCS), namely the water mass, the first and second condensing covers and the basin liner, including the reflecting mirror. Experiments were conducted for both the single slope conventional solar still (CSS) and double condensing chamber solar still (DCS) on an hourly basis for comparison of their performance. Numerical computations for evaluating the hourly temperatures and yield have been performed for Delhi climatic conditions. The obtained results have been compared with the experimental observations. It is observed that there is a fair agreement between the theoretical and experimental observations.

Keywords: Solar stills; Thermal modelling; Solar energy; Distilled water

Nomenclature

\[ A \] Area, m\(^2\)

\[ c \] Conductance of air

\[ C_d \] Coefficient of diffusion

\[ C_w \] Specific heat, J/kg °K

\[ g \] Acceleration due to gravity, m/s\(^2\)

\[ h \] Heat transfer coefficient, W/m\(^2\) °K

\[ h_{bw} \] Convective heat transfer coefficient from basin liner to water, W/m\(^2\) °K

\[ h_b \] Conductive heat transfer coefficient from basin liner to ambient, W/m\(^2\) °K

\[ h_p \] Convective heat transfer coefficient from condensing cover-I to ambient, W/m\(^2\) °K
$h_4$ Convective heat transfer coefficient from condensing cover-II to ambient, W/m$^2$ °K
$h_5$ Total heat loss coefficient from condensing cover-I to outer glass cover W/m$^2$ °K
$G$ Solar intensity at time $\theta$, W/m$^2$
$k$ Thermal conductivity, W/m °K
$L$ Thickness, m
$\mathcal{L}$ Latent heat of vaporization, J/kg
$m_{w}$ Hourly distillate output, kg/m$^2$/h
$M_w$ Water mass in basin liner, kg
$P$ Saturated partial pressure, N/m$^2$
$q$ Rate of heat transfer, W/m$^2$
$q_u$ Rate of energy transferred from chamber-I to chamber-II, W/m$^2$
$R$ Reflectivity of surface
$T$ Temperature, °K
$\theta$ Time, s
$U_{3b}$ Overall heat transfer coefficient from water surface to ambient air, W/m$^2$ °K
$U_{10}$ Overall top loss coefficient from water surface to ambient air, W/m$^2$ °K
$v$ Wind velocity, m/s

Greek symbols
$\alpha$ Absorptance
$\tau$ Effective transmittance after reflection and absorption from double glass
$\alpha'_b$ Total absorptance at basin liner
$\alpha'_w$ Total absorptance at water mass
$\varepsilon$ Emittance
$\varepsilon_{eff}$ Effective emittance of water and glass surface
$\rho$ Density, kg/m$^3$
$\rho'$ Reflectance of mirror
$\sigma$ Stefan–Boltzmann constant W/m$^2$ K$^4$

Subscripts
$a$ air
$b$ basin liner
$c$ metallic sheet
$cw$ convection from water surface to glass
$c_1$ condensing cover-I
$c_2$ condensing cover-II
$ev$ evaporation from water surface to glass
$g_1$ inside glass cover
$g_2$ outside glass cover
$g$ condensing cover-I
$1$ wood
$2$ polyurethane
$rw$ radiative heat transfer from water to glass
$w$ water
1. Introduction

A review of the work done in the field of conventional solar stills (CSS) has been performed by Malik [14] and updated by Tiwari [22]. It has been reported that the performance of a conventional solar distillation (CSS) system can be predicted by using theoretical methods of solutions. These include:

(i) computer simulation (Cooper [3, 4]);
(ii) thermic circuits and the Sankey diagrams (Frick [8]);
(iii) periodic and transient insolation (Hirshman and Roelof [9]; Baum et al. [2]; Nayak et al. [16]; Sodha et al. [19]; Tiwari and Madhuri [21]; and Lawrence et al. [12]);
(iv) iteration methods (Sharif and Kiss [18]; and El-Refaie et al. [7]);
(v) numerical methods (Kamal [10]; and Sartori [17]).

In their analyses, the basic internal heat and mass transfer relations given by Dunkle [6] have been used.

Further, it has been observed that the performance of conventional solar stills (kg/m²/day) can be improved by using the following methods:

(a) use of black dye in basin (Sodha et al. [19]; Tiwari and Madhuri [21]; Lawrence et al. [11] and Tamini [20]);
(b) reducing bottom heat loss coefficient (Cooper [3]; and Tiwari and Madhuri [21]);
(c) reducing water depth in basin (Tiwari and Madhuri [21]; and Lawrence et al. [13]);
(d) using reflector (Wibulswas and Tadtiam [23] and Tamini [20]);
(e) using internal condenser (Ahmed [1]);
(f) using back wall with cotton cloth (Wibulswas and Tadtiam [23]);
(g) regenerative effect in back wall (Wibulswas [24]).

To achieve a higher water temperature, some of the above mentioned modifications are used in designing the double condensing chamber solar still (DCS). The upward heat loss is reduced by using double glazing. Chamber-II is created behind the partition wall using stainless steel as condensing cover-II. Condensing cover-II is exposed to the ambient which helps in fast release of the latent heat of condensation. Vapours are transferred from chamber-I to chamber-II through a vent which is provided at the top of the partition wall.

The proposed model of the double condensing chamber solar still (DCS) has the following advantages.

(i) The heat capacity of the metallic condensing cover (Condensing cover-II) is a minimum, unlike the glass cover of the conventional solar still (CSS).
(ii) The condensing cover-II of the double condensing chamber solar still (DCS) acts only as a condenser, unlike the conventional solar still (CSS).
(iii) The water temperature in the double condensing chamber solar still (DCS) becomes much higher than in the single slope conventional solar still due to a lower value of the top heat loss coefficient (double glass cover is used).
In the present paper, a thermal model of a double condensing chamber solar still has been developed. The mathematical model is based on energy balance equations of the different components of a new design solar still (DCS). The results have been compared with experimental results, and it has been inferred that there is a fair agreement between the theoretical and experimental observations. The performance of a conventional solar still under similar environmental conditions has also been obtained and compared with the performance of a double condensing chamber solar still.

2. Working principle

The cross-sectional view of a double condensing chamber solar still (DCS) is shown in Fig. 1. The solar radiation after reflection, absorption and transmission from the double glass cover enters chamber-I of a distiller unit. This transmitted radiation is further partially reflected and absorbed by the water mass. The attenuation of solar flux in the water mass depends on the absorptivity and depth, respectively. The solar radiation finally reaches the blackened basin, where it is mostly absorbed. After absorption of the solar radiation at the basin liner, most of the thermal energy is convected to the water mass and the water gets heated. The heat loss below the basin liner is reduced by providing an insulation and vapour below it. The temperature of the inner glass cover is increased significantly due to the reduced heat loss coefficient from the top of the inner glass cover. Hence, the overall temperature difference between the water and the inner glass cover is reduced, and the pressure in chamber-I is increased significantly due to its higher operating temperature range. At the same time, the pressure in the second chamber is relatively low due to its direct exposure to ambient through a metallic sheet, generally known as the condensing cover-II. The vapour transferred from chamber-I to chamber-II due to the difference in pressure between the two chambers. Most of the transferred vapour is condensed on the vertical walls, after releasing its latent heat of condensation, and is collected through distillation output-2 and distillate output-3. The condensation at the back of the partition wall, which is collected from the distillate output-3, is only 4–5% of the total output. It is, therefore, neglected for calculation purposes. Further, there is also an output, collected through distillate output-1, from the inner glass cover because of its lower temperature than the water mass.

3. Experimental setup and observations

Fig. 1 shows a pictorial view of a double condensing chamber solar still. The still, with effective area of $1.290 \text{ m} \times 1.145 \text{ m}$, is made of timber material of thickness $0.015 \text{ m}$, and it is painted outside and inside with waterproof paint. The back wall is made of stainless steel, measuring $1.145 \text{ m} \times 0.455 \text{ m}$. The enclosure of the solar still is divided into two chambers by providing a partition wall $0.070 \text{ m}$ away from a metallic back wall from the inside. A metallic tray ($1.004 \text{ m} \times 1.000 \text{ m} \times 0.040 \text{ m}$), which is blackened from inside, is used as a basin, and it is placed over the horizontal platform. A double glass cover ($0.005 \text{ m}$), fitted in an aluminium frame ($1.142 \text{ m} \times 1.289 \text{ m} \times 0.018 \text{ m}$) is placed over the structure, having waterproof foam of
Fig. 1. (a) Cross-sectional view of double condensing chamber solar still (DCS). (b) Pictorial view of a double condensing chamber solar still with glass cover.
0.023 m thick fixed at the top along the periphery of a solar still. A provision for filling the water into the blackened tray has been made. Further, there is an outlet for the tray, generally used during the cleaning period. The distillate output has been collected through outlets provided below the lower ends of the inner glass cover and the metallic wall. The photograph of the system has been shown in Fig. 2. The water, about 20 litres, is filled through the inlet before the start of the experiment. Temperatures of the water, inside glass, outside glass and metallic sheet were recorded on an hourly basis using calibrated thermocouples and a sensitive digital thermometer. One end of each thermocouple is fixed at the center of the metallic sheet and the inside and outside glass covers with the help of Araldite (adhesive). The water temperature is measured by dipping the thermocouple in the water mass. The ambient temperature is measured by using a mercury thermometer. The solar intensity is measured by using a solarimeter placed on the surface of the inclined glass cover. Experiments were conducted during 15–29 December, 1995 for both the double condensing chamber solar still (DCS) and the single slope conventional solar still (CSS). The results for a typical day (23 December) have been given in Table 1 and Table 2, respectively.

4. Energy balance

In order to write the energy balance equations, the following assumptions have been made.

(i) The heat capacity of the glass covers, metallic condensing cover, insulating material and tray has been neglected.

Fig. 2. Photograph of a double condensing chamber solar still.
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Comparison between experimental and theoretical values of temperatures and output of double condensing chamber solar still (DCS)

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<th>Output back (kg/m²h)</th>
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Total output Experimental(kg/m²·day) 0.706 0.733 Theoretical(kg/m²·day) 0.705 0.753
Table 2
Comparison between experimental and theoretical values of temperatures and output of conventional solar still (CSS)

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</table>

Total output

Experimental: 0.983 kg/m² day
Theoretical: 1.0915 kg/m² day
(ii) There is no temperature gradient along the water depth and glass cover thickness.
(iii) The system is in a quasi-steady state condition.
(iv) The side losses from the tray have been neglected due to the shallow water depth (0.02 m).
(v) The inclination of the glass cover is small.
(vi) The solar distiller unit is vapour leakage proof.
(vii) The evaporating surface area and inner glass cover area are approximately the same.

The energy balance equations are as follows:

**Basin Liner:**

\[ \alpha' G = h_{bw}(T_b - T_w) + h_b(T_b - T_a) \]  

- the rate of energy absorbed by basin liner
- the rate of energy lost by convection to water
- the rate of energy lost by conduction to ambient through bottom

**Water Mass:**

\[ h_{bw}(T_b - T_w) + \alpha' G = M_wC_w\left(\frac{dT_w}{d\vartheta}\right) + q_w + q_{n} \]  

- the rate of energy convected from basin liner
- the rate of energy absorbed
- the rate of energy stored
- the rate of energy lost to glass cover by radiation, convection and evaporation
- the rate of energy transferred to chamber II

**Inner glass (condensing cover-I):**

\[ \dot{q}_w = h_{g}(T_{g1} - T_{g2}) \]  

- the rate of energy received from water mass by convection, radiation and evaporation
- the rate of energy lost to outer glass cover

**Outer glass:**

\[ h_{g}(T_{g1} - T_{g2}) = h_{A}(T_{g2} - T_a) \]  

- the rate of energy received by inner glass cover
- the rate of energy lost to ambient
Eqs. (3) and (4) can be combined and written as
\[ \dot{q}_w = h_s(T_{g1} - T_a) \]  
the rate of energy received from water mass by convection, radiation and evaporation
\[ \dot{q}_w = \frac{C_d A_s}{h_s \sqrt{2(dP/\rho)} \sqrt{dP}} \]  
the rate of energy received from chamber I
\[ \dot{q}_w = h_A(T_c - T_a) \]  
the rate of energy lost to air

Second condensing cover (Metallic sheet):
\[ \dot{q}_w = C_d A_s / A_c \sqrt{2(dP/\rho)} \sqrt{dP} = h_s(T_{g1} - T_a) \]  
the rate of energy received from glass cover temperature \( T_{g2} \) can be obtained by using Eqs. (3) and (4).

Eq. (2), after eliminating \( T_b \) and \( T_{g1} \) from Eqs. (1) and (5), can be rewritten as
\[ M_w C_w dT_w / d\theta = h_s \dot{G} + \dot{G}' \]  
and
\[ m_{c1} = A_b h_{ew} \frac{(T_w - T_a)}{L} \]  
where
\[ L = 2.4935 \times 10^6 \left[ 1 - 9.4779 \times 10^{-4} T + 1.3132 \times 10^{-7} T^2 - 4.7974 \times 10^{-9} T^3 \right] \]

5. Numerical procedure and computation

The following procedure is adopted in solving Eqs. (6) and (7).
Eq. (7) can be written as
\[ dT_w = (1/M_w C_w) (h_s \dot{G} + \dot{G}' - (U_{3b} + U_{10})(T_w - T_a) - \dot{q}_w) d\theta \]
\[ T_w(\theta + 1) = T_w(\theta) + (1/M_w C_w) (h_s \dot{G} + \dot{G}' - (U_{3b} + U_{10})(T_w - T_a) - \dot{q}_w) d\theta \]  
where \( \dot{G} \) and \( T_{g1} \) represent the average values of intensity and ambient temperature for the
interval of one hour. The unknown variables in Eqs. (6) and (10) are $T_c$ and $T_w$. First, the value of $T_c$ (as a root) for given $T_w$ and $T_{g1}$ is found from Eq. (6) using the Newton Raphson method. That value of $T_c$ is taken for which $F(T_c)$ is less than the given value of accuracy.

$$F(T_c) = C_d A_r \sqrt{2 \frac{dP}{\rho} \frac{dP}{\rho} - h_d A_r (T_c - T_v)}$$

Using this value of $T_c$ in Eq. (10), the value of $T_w$ is evaluated by the finite element method for an interval of one second. Knowing $T_c$ and $T_w$, $T_{g1}$ and $T_{g2}$ are calculated using Eqs. (5) and (4). The obtained value of $T_w$ is again put in Eq. (6) to evaluate $T_c$. This procedure [Fig. 3] is repeated for one hour with an interval of one second. This interval is chosen for better
accuracy. Further, the output from chamber-I and chamber-II is calculated using Eqs. (8) and (9).

The total output for one hour is obtained by adding the output for each second.

\[ \dot{m}_{c1}(\theta + 1) = \dot{m}_{c1}(\theta) + A_{h}h_{w}(T_{w} - T_{g1})/L \]
\[ \dot{m}_{c2}(\theta + 1) = \dot{m}_{c2}(\theta) + A_{c}d/\rho C_{p} \]

Hence, after an hour, the values of the following parameters are available theoretically, \( T_{w}, T_{g1}, T_{c1}, T_{g2}, \dot{m}_{c1}, \dot{m}_{c2} \), which are compared with the experimental values. The comparison has been given in Table I, and Figs. 5-7.

For the conventional solar still under similar experimental conditions, the comparison of theoretical and experimental values of water temperature, glass temperature and output is given in Table 2 and Fig. 8.

6. Numerical results and discussion

The different design and climatic parameters used in the study have been reported in Table 3 and Fig. 4, respectively. The hourly average intensity and ambient temperature have been taken from Table I for the numerical computations, which are average values for one hour.

For the double condensing chamber solar still (DCS), the obtained hourly values of various temperatures have been shown in Figs. 5 and 6. Experimental values corresponding to the theoretical values have been shown in the same figure. It is observed that the values of \( T_{g1} \) are much higher than the condensing cover \( (T_{c1}) \) due to the minimum heat loss from the inner glass cover to ambient due to the double glazed cover, as expected. It is also important to mention here that, due to higher values of the inner glass cover and water temperatures, the operating temperatures of the first chamber become higher than the temperature of the second condensing chamber. This creates a higher pressure difference between the two chambers, as
reported earlier, and hence, diffusion of the vapors takes place. The diffusion of the vapors has been considered in the thermal modelling by using the diffusion coefficient \( C_d \). It is very difficult to measure the diffusion coefficient \( C_d \) due to the enclosed chambers. However, the best value of the diffusion coefficient \( C_d \) has been used for validation of the theoretical model.

This experiment can also be used for finding the value of this coefficient \( C_d \). It has also been observed experimentally that the diffusion coefficient varies between 0.002–0.006 from morning to evening due to the variation in the operating temperature range of both chambers.

Further, the hourly variation of yield obtained from both chambers has been shown in Fig. 7. It is further to be noted that the hourly yield from chamber-II is always higher than
Fig. 7. Comparison between theoretical and experimental values of output from chamber-I and chamber-II on hourly basis for DCS.

from chamber-I. It is due to the fact that the pressure difference between the chambers is more than that indicated by the water and inner glass temperatures.

For the conventional solar still (CSS), the theoretical and experimental values of water temperature, glass temperature and output is given in Table 2. Fig. 8 shows the comparison of these parameters. The experimental output of the double condensing chamber solar still (1.439 kg/m² day) is 46 % higher than the conventional solar still (0.983 kg/m² day) for 23 December, 1995.

Fig. 8. Comparison between theoretical and experimental values of water temperature, glass temperature and output of conventional solar still on hourly basis for 23 December, 1995.
Table 3
Design parameters and their values

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<th>Parameters</th>
<th>Value</th>
<th>Parameters</th>
<th>Value</th>
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<tr>
<td>p'</td>
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</table>

Finally, it is observed that there is a fair agreement between the theoretical and experimental observations.

7. Experimental uncertainties

The external and internal uncertainties have been calculated for the solar stills.

\[
\% \text{ Internal Uncertainty} = \frac{\sqrt{\Sigma \sigma_i^2 / N}}{\text{Average of total number of observations}}
\]

(Nakra and Choudhury[15]

\[= 13.5\%\]

where

\[\sigma = \text{ standard deviation} = \frac{\sum (X - \bar{X})^2}{N_0} .\]

\[X - \bar{X} = \text{ deviations of the observations from the mean and } N_0 = \text{ number of observations taken to find the mean.}\]

External uncertainty is measured by taking into account the least count of the various instruments used. In our case, it comes out to be \% external uncertainty = 1.5 % and total uncertainty = 15 %. 
8. Conclusions

The performance of the double condensing chamber solar still (DCS) for a typical winter day (23 December, 1995) of Delhi gives about 46% higher output than the single slope conventional solar still (CSS). Further, there is a fair agreement between the experimental results and theoretical results obtained by using the present model. It can also be predicted that the double condensing chamber solar still will be more economical on large scale production to provide drinking water in remote areas.

Appendix A

\[ \tau = (1 - R_{g1})(1 - R_{g2})(1 - \alpha_{g1})(1 - \alpha_{g2}) \]

\[ \alpha_{e} = 1/2(1 + \rho)^{2}(1 - \alpha_{e})(1 - \alpha_{w})\alpha_{e}\tau \]

\[ \alpha_{w} = 1/2(1 + \rho)^{2}(1 - \alpha_{e})(1 - \alpha_{w})(1 - \alpha_{s}) + \alpha_{w} \]

\[ q_{w} = q_{rw} + q_{cw} + q_{ew} = h_{1}(T_{w} - T_{g}) \]

\[ h_{1} = h_{rw} + h_{cw} + h_{ew} \]

\[ h_{rw} = \epsilon_{rw}(T_{w} + 273)^{2} + (T_{g} + 273)^{2}(T_{w} + T_{g} + 546) \]

\[ h_{cw} = 0.884 \left[ \frac{P_{e}(T) - P_{g}(T)}{268.9 \times 10^{3} - P_{e}(T)} \right]^{1/3} \]

\[ h_{ew} = 16.273 \times 10 - 3h_{cw} \frac{P_{e}(T) - P_{g}(T)}{T_{w} - T_{g}} \]

\[ P(T) = \exp \left\{ 25.317 - \frac{5144}{T + 273.15} \right\} \]

\[ \rho = \frac{353.43}{T_{l} + 273.15} \]

\[ dP = P_{e}(T) - P_{g}(T) \]

\[ \epsilon_{eff} = \left[ 1/\epsilon_{e} + 1/\epsilon_{w} - 1 \right]^{-1} \]

\[ h_{4} = 5.7 + 3.8 \nu \quad \text{(Duffie and Beckman[5])} \]

\[ h_{h} = [L_{1}/k_{1} + 1/c + L_{2}/k_{2} + l/h_{0}]^{-1} \]

\[ h_{g} = [L_{g1}/k_{g1} + l/c + L_{g2}/k_{g2}]^{-1} \]
\[ h_o = \left[ L_{b1}/k_{c1} + l/c + L_{b2}/k_{c2} + l/h_o \right]^{-1} \]

\[ h = \frac{h_{bor}}{h_{bor} + h_o} \]

\[ U_{3b} = \frac{h_{bor}}{h_{bor} + h_b} \]

\[ U_{1b} = \frac{h_b h_o}{h_t + h_o} \]

**References**
