THERMAL EFFICIENCY OF DOUBLE SLOPE FRP SOLAR DISTILLER: AN ANALYTICAL AND EXPERIMENTAL STUDIES.

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ABSTRACT An analytical expression for thermal efficiency of Double Slope Fiber Re-enforced Plastic (FRP) solar still has been derived by incorporating the uniform water flow in both slopes due to capillary action as well as water pressure in a pipe provided above the reservoir. In order to validate the theoretical results obtained by present model, have been compared with the experimental results of Tiwari and Salim[2]; it was observed that due to proposed modification in solar distiller for continuous water flow, the thermal efficiency is significantly increased for least water flow velocity and is in accordance with the results of Tiwari and Salim [2].

Key words: Solar distillator, Distiller unit, Solar still.

INTRODUCTION: A review on historical background, importance, different designs and economic aspects of solar distillation system have been carried out by Malik et al [1]. It was observed that multiwicd solar still is more efficient than conventional distillation unit. Based on multiwick solar still, a double slope multiwick made up with fiber re-enforced plastic (FRP) material were designed, constructed and tested by Tiwari and Salim [2]; which is found to be more efficient than single slope during summer due to maximum solar radiation
available for east-west orientation of double slope solar still. During the operation of double slope FRP multiwrick solar still, the following problems were noticed:

(i) for least water flow due to capillary action, the jut cloth got dried during peak sun-shine hours and

(ii) wastage of excess hot water during early and late working hours.

First problem was solved by immediate raising of water column in reservoir during peak hours and second problem has been solved in the proposed new design by connecting excess water outlet to the reservoir by raising the height of partition between distillate and lower end of jute cloth (Fig. 1). Since the daily raising of water column during peak hours is a very difficult task for large scale installation (Tiwari [3]), hence a new arrangement has been made for a constant flow of water over jute cloth or intermittent flow along with capillary action. In this way, the maintenance cost of the system can be minimised, in the proposed new design in comparison to earlier one.

Analytical expressions for water and glass temperatures and thermal efficiency of the system have been derived in terms of system design and climatic parameters, based on energy balance of different components of unit. The effects of mass flow rate on total as well as individual internal heat transfer coefficients have also been studied. Numerical computations were also carried out for a typical day in Delhi to predict the performance of the system.

The theoretical results were in good agreement with the experimental results carried out by Tiwari and Salim [2].

THEORETICAL ANALYSIS
Following assumptions have been made:

(i) Heat capacities of glass and flowing water are negligible,

(ii) Dunkle's relation is valid due to parallel surfaces of water and glass for close cycle system,
(iii) Flow rate is uniform from \( x = 0 \) to \( x = L \) (fig.1.b). The energy balance equations for glass cover and flowing water (fig.1.b. Tiwari et al.[6]) are:

\[
(\alpha \tau) g \ell(t) b \, dx + h_1(T_w - T_g) b \, dx = h_2(T_g - T_a) b \, dx \tag{1}
\]

\[
(\alpha \tau) b \ell(t) b \, dx = m_w c_w (dT_w / dx) \, dx + h_1(T_w - T_g) b \, dx
+ h_b(T_w - T_a) b \, dx \tag{2}
\]

Where,

\[
h_1 = h_{rw} + h_{cw} + h_{ew};
\]

\[
h_{rw} = \left[ \frac{6 \xi (T_w + 273)^4 - (T_w + 273)^4}{(T_w - T_g)} \right];
\]

\[
h_{cw} = 0.884 \frac{(T_w - T_g + [(P_w - P_g)(T_w + 273)/(268.9 \times 10^3 - T_w)])^{1/3};}
\]

\[
h_{ew} = 0.016 \frac{h_{cw} \times [(P_w - P_g)/(T_w - T_g)]}{Tiwari et al.[4]} \text{ or, Yeh et al.[5]}
\]

The initial values of \( T_w \) and \( T_g \), to calculate first value of \( h_1 \) are:

\[
T_w = T_a + 5.0 \quad \text{and} \quad T_g = T_a + 2.0
\]

\[
h_b = \left[ \left( L_i / K_i \right) + (1/h_1) \right]^{-1}
\]

and,

\( m_w \) is constant mass flow rate maintained by special arrangement shown in fig.1.a.

\[
h_2 = 5.7 + 3.8 \, \text{V}
\]

from equations (1) & (2),

\[
(dT_w / dx) + a \, T_w = f(t) \tag{3}
\]

Where,

\[
a = b (U_{12} + h_b) / m_w c_w
\]

\[
U_{12} = h_1 h_2 / (h_1 + h_2)
\]

and,

\[
f(t) = \left[ b (\alpha \tau)_{\text{eff}} l(t) + b(U_{12} + h_b)T_a / (m_w c_w) \right]
\]

\[
(\alpha \tau)_{\text{eff}} = (\alpha \tau)_b + (\alpha \tau)_g \left[ h_1 / (h_1 + h_2) \right]
\]
Figure 1a. Cross-sectional view of the double slope FRP multilick solar still.
- Black jute cloth
- Black polythene

Figure 1b. Elemental length $dx$ of flowing water.

Figure 1c. Photograph of the system (Tiwari and Salim (2))
The solution of this equation (3) can be written by using boundary condition i.e. \( T_w|_{x=0} = T_w \) (\( = T_a + 5.0 \) approximately due to feeding of excess water) as,

\[
T_w = (f(t)/a)[1 - \exp(-ax)] + T_w e^0 \exp(-ax) \quad (4)
\]

Now the average water temperature over 0 to L length can be written as,

\[
\bar{T}_w = \frac{1}{L} \int_0^L T_w \, dx; \text{ or,}
\]

\[
\bar{T}_w = (f(t)/a)\{1 - (1 - \exp(-aL))/aL\} + T_w e^0 \{1 - \exp(-aL)/aL\} \quad (4a)
\]

Similarly, the average glass temperature over 0 to L length can be written as,

\[
\bar{T}_g = [\omega(T)g / (l(t) + h_1 T_w + h_2 T_a)] / (h_1 + h_2) \quad (4b)
\]

The values of \( \bar{T}_w \) and \( \bar{T}_g \) calculated from Eqs. 4a and 4b are used to calculate next set of \( h_1 \).

And, the instantaneous efficiency can be defined by using the above equation (4a) as follows:

\[
n = \frac{h_{ew} (\bar{T}_w - \bar{T}_g) / l(t)}{(h_2/(h_1 + h_2)){[1 - (1 - \exp(-aL))/aL]}^{1}} + [h_2/(h_1 + h_2)]{{(T_w - T_a)/l(t)}} \quad (5)
\]

Case (i):

When \( \frac{aL}{l(t)} \ll 1 \)

Equation (4a) becomes,

\[
\bar{T}_w = T_w \quad (5a)
\]

And, equation (5) becomes,

\[
n = \frac{h_{ew}(\bar{T}_w - \bar{T}_a) / l(t)}{h_{ew}[(h_2/(h_1 + h_2))(T_w - T_a)/l(t)]}\quad (5b)
\]
Case (ii):

When \( aL \gg 1 \), equation (4a) becomes,

\[
\bar{T}_w = \left( \frac{f(t)}{a} \right) + T_0(1/aL)
\]

and, equation (5) becomes,

\[
n = \frac{h_2}{(h_1+h_2)} \left\{ (\alpha\tau)_{\text{eff}}/(U_{12}+h_b) + \left[ (T_{w0}-T_a)/(1/(t)) \right](1/aL) \right\} - (\alpha\tau)_g/(h_1+h_2)
\]

For this system, it is clear from equation (5) that the efficiency is dependent on mass flow rate along the length and the term \( aL \).

**NUMERICAL RESULTS AND DISCUSSIONS**

In order to appreciate the numerical results, computations have been made for one of the typical hot day at Delhi.

The following parameters are used to evaluate the efficiency of the double slope FRP solar still with different mass flow rates as shown in different figures:

\[
(\alpha\tau)_b = 0.8; \quad L = 0.87 \text{ m. to 3.5 m.}
\]

\[
(\alpha\tau)_g = 0.05; \quad V = 5 \text{ m/s}
\]

\[
h_b = 0.74 \text{ W m}^{-2} \text{ \circ C}^{-1};
\]

\[
ce = 4190 \text{ J kg}^{-1} \text{ \circ C}^{-1}; \quad b = 1.1 \text{ m.}
\]

\[
\epsilon = 0.9; \quad \sigma = 5.66 \times 10^{-8} \text{ W m}^{-2} \text{ K}^{-4}
\]

The \( h_1 \) has been calculated separately for every set of observations by using preceding values of average water temperature \( (T_w) \) and average glass temperature \( (T_g) \) as mentioned earlier. The calculated variation of \( h_1 \) for this system with time is shown in (Fig.2). It is clear from Fig.2 that when the mass flow rate tends to zero, the effect of \( h_1 \) is enough to notice. While Fig.3 shows the effect of mass flow...
Figure 2. The variation of total heat transfer coefficient $h_1$ with time (hrs).

Figure 3. The variation of different heat transfer coefficients $h_{rw}$, $h_{cw}$ and $h_{sw}$ with time (hrs.)

Figure 4. The variation of efficiency $n$ with logarithm of mass flow rate, log m.

Figure 5. The variation of efficiency $n$ with length L.
rate on internal heat transfer coefficients namely \( h_{rw}, h_{ow} \) and \( h_{ow} \). In this total heat transfer coefficient, the most significant contribution is of evaporative heat transfer coefficient for any mass flow rate within the range considered here. The effects of mass flow rate on radiative and convective heat transfer coefficients are almost constant throughout the day and negligibly small in comparison with evaporative heat transfer coefficient from water to glass at mid-day. Considering the effect of evaporative heat transfer coefficient from water to glass, it decreases as the mass flow rate increases at any particular time because of decrease in water temperature \( T_w \) due to large mass flow rate.

The calculated variation of efficiency of this system with mass flow rate \( (m_w) \) is shown in fig.(4) by using both expressions of efficiency (Eq.5) from fig.(4), it is clear that:

(i) when \( aL \gg 1 \) (i.e. \( m_w \) is small) \( \rightarrow 0; L \) is constant) there is a sharp variation in efficiency and

(ii) when \( aL \ll 1 \) (i.e. greater values of \( m_w; L \) is constant) the efficiency shows almost independence on the mass flow rate \( (m_w) \). This trend is obvious from the theoretical results (equations (5), (5b) & (5d)), because when mass flow rate is high, mass does not get sufficient time to get heat for being evaporated but when mass flow rate tends to zero it gets sufficient time for being evaporated and to show the variation in efficiency.

The computation has also been made to check the variation in efficiency with length \( (L) \) when \( m_w \) is constant for three cases namely when (i) \( aL \gg 1 \); (ii) \( aL = 1 \) and (iii) \( aL \ll 1 \) which has been depicted in fig.(5). In two extreme cases i.e. \( (aL \gg 1) \) & \( aL \ll 1 \) the efficiency does not show any dependence on length. It is because when \( aL \gg 1 \) (i.e. very low rate of mass flow), the value of \( a \) is much greater than that of length \( L \) does not contribute in its lower rate and when \( aL \ll 1 \) (i.e. very high rate of mass flow), the value of \( a \) is much lower than that of \( L \) and again \( L \) does not contribute. But when \( aL=1 \), \( a \) & \( L \) are of the same order so the length
does contribute, which is clear from fig. (5). The curve for \( aL \sim 1 \) tends to the upper curve for \( aL \gg 1 \) and lower curve for \( aL << 1 \) in fig. (5).

REFERENCES


NOMENCLATURE

\( A_b \) --- Area of the basin ( \( m^2 \) )
\( b \) --- Breadth of the basin ( \( m \) )
\( C_w \) --- Specific heat of water ( \( J \ Kg^{-1} \ O\C^{-1} \) )
\( dx \) --- Elemental thickness along \( x \) ( \( m \) )
\( h_1 \) --- Total heat transfer coefficient from water to glass ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_2 \) --- Heat transfer coefficient from glass to ambient ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_b \) --- Heat transfer coefficient from water to ambient through insulation ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_{cw} \) --- Convective heat transfer coefficient from water surface to glass ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_{ew} \) --- Evaporative heat transfer coefficient from water surface to glass ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_i \) --- Heat transfer coefficient from bottom insulation to ambient ( \( W \ m^{-2} \ O\C^{-1} \) )
\( h_{rw} \) --- Radiative heat transfer coefficient from water surface to glass ( \( W \ m^{-2} \ O\C^{-1} \) )
\( I(t) \) --- Solar intensity \( (W \ m^{-2}) \)
\( K_i \) --- Thermal conductivity of insulation \( (W \ m^{-1} \ \degree C^{-1}) \)
\( L_i \) --- Thickness of insulation \( (m) \)
\( L \) --- Length of each side of double slop FRP body \( (m) \)
\( m_w \) --- Mass flow rate along \( L \) \( (Kg \ s^{-1}) \)
\( P_w \) --- Partial pressure of water vapor at water temperature \( (N \ m^{-2}) \)
\( P_g \) --- Partial pressure of glass at ambient temperature \( (N \ m^{-2}) \)
\( T_a \) --- Instantaneous ambient temperature \( (^\circ C) \)
\( T_g \) --- Instantaneous glass temperature \( (^\circ C) \)
\( T_w \) --- Instantaneous water temperature \( (^\circ C) \)
\( T_{w0} \) --- Instantaneous water temperature at \( x=0 \) \( (^\circ C) \)
\( V \) --- Wind speed \( (m \ s^{-1}) \)
\( X \) --- Space coordinate along length \( (m) \)
\( n \) --- Efficiency of the system
\( (\alpha \tau)_b \) --- Absorptivity of blackened surface
\( (\alpha \tau)_g \) --- Product of absorptivity and transmittivity of glass cover
\( \varepsilon \) --- Emissivity of the water surface
\( \sigma \) --- Stefan-Boltzman constant