HEAT AND MASS TRANSFER IN TUBULAR SOLAR STILL UNDER STEADY CONDITION

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A quasi steady heat and mass transfer model of a Tubular Solar Still taking account of humid air properties inside the still is presented in this study. Time variations of saline water, humid air, cover and trough temperatures and production and condensation fluxes are derived under a steady meteorological condition. In order to validate the proposed model, an indoor production experiment on a Tubular Solar Still was carried out in a thermostatic room at the University of Fukui. A special technique to investigate the water-vapor movement in the still was also developed and could detect a time lag between the evaporation flux and the condensation flux. It is found from the production experiment that the analytical solutions derived from the present model could reproduce the experimental results on the saline water temperature, the humid air temperature, the cover temperature and production and condensation fluxes.

Key Words : *Tubular Solar Still, solar energy, heat and mass transfer, evaporation, convection, condensation, radiation, distilled water*

1. INTRODUCTION

The human and economical activities, particularly in arid and remote areas, may depend on desalination performance to meet fresh water demand produced from brackish or saline water. Solar distillation is the simplest desalination technique, compared with other types, e.g., multiple-effect distillation, multi-stage flash, reverse osmosis, electro-dialysis and biological treatment. Solar distillation may be also one of viable options for providing drinking water for a single house or a small community in arid regions. A basin-type solar still is the most popular method of solar distillation but unfortunately has not been fully advanced yet. The major reasons for this fact may be attributed to low productivity of distillate and the difficulty of rapid and easy removal of salt accumulated in the basin. To enhance distilled water productivity, a new type of solar distillation, i.e., Tubular Solar Still (TSS) was designed by the authors and has been tested (since 2001) as well as the basin type still in the United Arab Emirates (UAE).

Conventionally, most of the heat and mass transfer models of the solar still have been described using temperature and vapor pressure of saline water surface and glass cover, neglecting the presence of intermediate medium, i.e., humid air (Dunkle[1)], Kumar et.al.[2)], Tiwari et.al.[(3)]). Nagai et.al[4)], however, found that the relative humidity of the humid air is by no means saturated in the daytime and proposed a heat transfer model on a basin type solar still. The production mechanism and heat and mass transfer parameters of a solar still can be judged more preciously under the steady meteorological condition and the results can apply to more accurate unsteady production model.

This paper aims to formulate the heat and mass transfer phenomenon in a TSS taking account of the humid air properties under a steady meteorological condition and to derive analytical expressions of saline water temperature, humid air temperature, tubular cover temperature, trough temperature and evaporation and condensation fluxes.

2. THEORY

Production principle of a TSS is illustrated in **Fig. 1**. The solar radiation, after transmission through a transparent tubular cover, is mainly

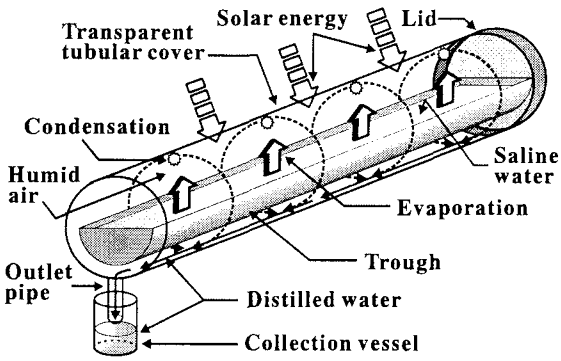

Fig. 1 Production principle of TSS

absorbed by saline water in the trough. The remaining small amount of the solar energy is absorbed by the tubular cover and trough. Thus, the water in the trough is heated and then begins to evaporate. There happen many types of heat transfer inside the tubular cover and outside, e.g., convective and evaporative heat transfer from the saline water to the humid air, convective and condensative heat transfer from the humid air to the tubular cover, radiative heat transfer between the water surface and the tubular cover, convective and radiative heat transfer from the tubular cover to the atmosphere. Convective heat transfer also occurs between the trough and the water and between the trough and the humid air. The evaporated water vapor is transferred to the humid air and then finally condensed on the tubular cover inner surface, releasing its latent heat of vaporization. The condensed water trickles down the bottom of the tubular cover inner surface due to gravity and is stored in a collection vessel through a pipe provided at the lower end.

(1) Mass and Energy balance equations

The proposed mass and energy balance equations are made up under the following assumptions:

i) Water temperature is uniform in the trough.
ii) Water vapor near the water surface is saturated.
iii) There is no leakage of the water vapor in the TSS.
iv) The absorption coefficient of the humid air is neglected, i.e., $\alpha_{ha}=0$.

The mass balance equation for water in the trough can be written as:

$$\frac{dV_W}{dt}=-\frac{m_e A_W}{\rho_W} \tag{1}$$

Fig. 2 shows the energy balance of TSS. All terms in equations in the present paper are explained in Nomenclature. Energy balance equations of the TSS are expressed for the following four different components.

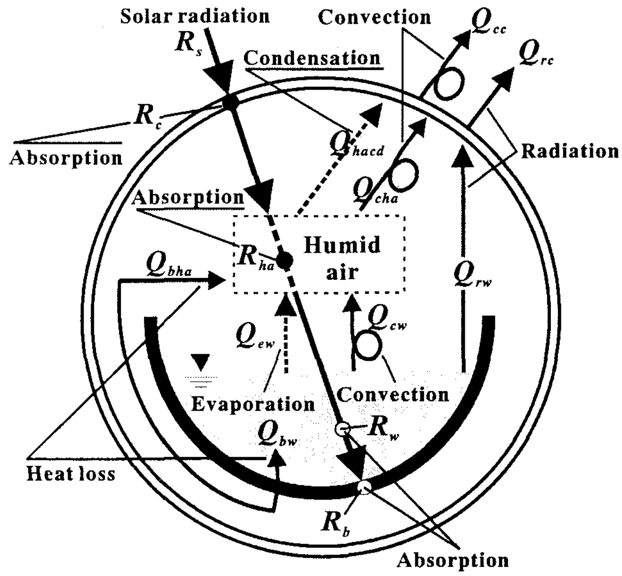

Fig. 2 Energy balance of TSS

Saline water:

$$(\rho C)_W \frac{\partial (V_W T_W)}{\partial t}=R_W+Q_{tW}-Q_{eW}-Q_{cW}-Q_{rW} \tag{2}$$

Trough:

$$(\rho CV)_t \frac{\partial T_t}{\partial t}=R_t-Q_{tW}-Q_{tha} \tag{3}$$

Humid air:

$$(\rho CV)_{ha}\frac{\partial T_{ha}}{\partial t}=R_{ha}+Q_{eW}+Q_{cW}+Q_{tha}-Q_{cha}-Q_{cdha} \tag{4}$$

Tubular cover:

$$(\rho CV)_c \frac{\partial T_c}{\partial t}=R_c+Q_{cha}+Q_{cdha}+Q_{rW}-Q_{cc}-Q_{rc} \tag{5}$$

where,

$Q_{cc} = h_{cc}(T_c-T_a)A_c$
$Q_{cdha} = Lh_{cdha}(\rho_{vha}-\rho_{vc})A_{ha} = Lh_{cdha}(\gamma T_{ha}-\psi T_c)A_{ha}$
$Q_{cha} = h_{cha}(T_{ha}-T_c)A_{ha}$
$Q_{cw} = h_{cw}(T_w-T_{ha})A_w$
$Q_{ew} = Lh_{ew}(\rho_{vw}-\rho_{vha})A_w = Lh_{ew}(\beta T_w-\gamma T_{ha})A_w$
$Q_{rc} = h_{rc}(T_c-T_a)A_c$
$Q_{rw} = h_{rw}(T_w-T_{ha})A_w$
$Q_{tha} = h_{tha}(T_t-T_{ha})A_{tha}$
$Q_{tw} = h_{tw}(T_t-T_w)A_{tw}$
$R_t = \tau_{t1}R_sA_w+\tau_{t2}R_s(2r_tl_t-A_w)$
$R_c = \tau_cR_s(2r_c)$
$R_{ha} = \tau_{ha}R_sA_{ha}$
$R_w = \tau_wR_sA_w$
$\tau_c = (1-al_c)\alpha_c$
$\tau_{ha} = (1-al_c)(1-\alpha_c)\alpha_{ha}$
$\tau_{t1} = (1-al_c)(1-\alpha_c)(1-\alpha_{ha})(1-al_w)(1-\alpha_w)(1-al_t)\alpha_t$
$\tau_{t2} = (1-al_c)(1-\alpha_c)(1-\alpha_{ha})(1-al_t)\alpha_t$
$\tau_w = (1-al_c)(1-\alpha_c)(1-\alpha_{ha})(1-al_w)\alpha_w$

(2) Steady state model and analytical solutions

Assuming that the energy balances of the trough, humid air, and tubular cover are in a quasi steady state, then the left hand term of Equations (2) to (4) will be neglected. The solutions of Equations (2) to (4) are eventually written in terms of water and air temperature as follows:

$$T_{ha} = JT_w + KT_a + M \quad (6)$$

$$T_t = NT_w + PT_a + Q \quad (7)$$

$$T_c = ST_w + UT_a + V \quad (8)$$

where,

$H_{2\text{-}c} = (h_{rc} + h_{cc})A_c$

$H_{2\text{-}ha} = Lh_{cdha}A_{ha}$

$H_{3\text{-}ha} = h_{cha}A_{ha}$

$H_{cw\text{-}w} = h_{cw}A_w$

$H_{ew\text{-}w} = Lh_{ew}A_w$

$H_{rw\text{-}w} = h_{rw}A_w$

$H_{tha\text{-}t} = h_{tha}A_{tha}$

$H_{tw\text{-}t} = h_{tw}A_{tw}$

$$B = \frac{H_{tw-t}}{H_{tha-t} + H_{tw-t}}$$

$$D = \frac{H_{tha-t}}{H_{tha-t} + H_{tw-t}}$$

$$E = \frac{R_t}{H_{tha-t} + H_{tw-t}}$$

$$F = \frac{-\left(H_{cw-w} + \beta H_{ew-w} + BH_{tha-t}\right)}{\psi H_{2-ha} + H_{3-ha}}$$

$$G = \frac{H_{3-ha} + H_{cw-w} + \gamma\left(H_{2-ha} + H_{ew-w}\right) + (1-D)H_{tha-t}}{\psi H_{2-ha} + H_{3-ha}}$$

$$I = \frac{-\left(R_{ha} + EH_{tha-t}\right)}{\psi H_{2-ha} + H_{3-ha}}$$

$$J = \frac{(1-F)H_{rw-w} - F\left(\psi H_{2-ha} + H_{3-ha} + H_{2-c}\right)}{(G-1)H_{3-ha} + (\psi G - \gamma)H_{2-ha} + G\left(H_{2-c} + H_{rw-w}\right)}$$

$$K = \frac{H_{2-c}}{(G-1)H_{3-ha} + (\psi G - \gamma)H_{2-ha} + G\left(H_{2-c} + H_{rw-w}\right)}$$

$$M = \frac{R_c - I\left(\psi H_{2-ha} + H_{3-ha} + H_{2-c} + H_{rw-w}\right)}{(G-1)H_{3-ha} + (\psi G - \gamma)H_{2-ha} + G\left(H_{2-c} + H_{rw-w}\right)}$$

$N = B + DJ$

$P = DK$

$Q = E + DM$

$S = F + GJ$

$U = GK$

$V = I + GM$

Substituting the expressions for T_{ha}, T_t and T_c i.e., Equations (6) to (8) into Equation (2), it can be rewritten as:

$$(\rho C)_w\left(T_w \frac{\partial V_w}{\partial t} + V_w \frac{\partial T_w}{\partial t}\right) = XT_w + YT_a + Z \quad (9)$$

where,

$X = (N-1)H_{tw-t} + (J-1)H_{cw-w} + (\gamma J - \beta)H_{ew-w} + (S-1)H_{rw-w}$

$Y = PH_{tw-t} + K\left(H_{cw-w} + \gamma H_{ew-w}\right) + UH_{rw-w}$

$Z = R_w + QH_{tw-t} + M\left(H_{cw-w} + \gamma H_{ew-w}\right) + VH_{rw-w}$

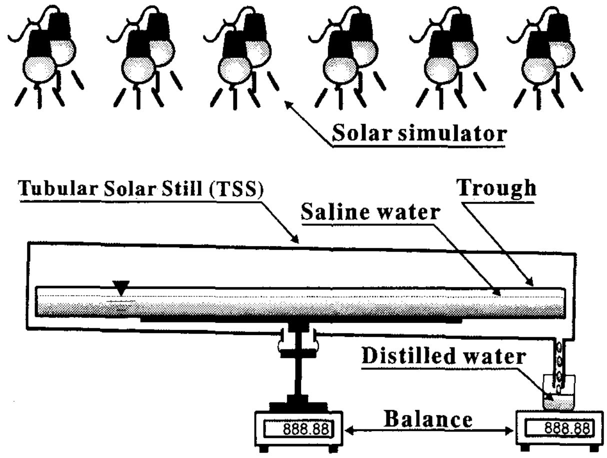

Fig. 3 Schematic diagram of the experiment

Substituting Equation (1) into Equation (9) yields the following differential equation:

$$\frac{dT_w}{dt} = aT_w + b \quad (10)$$

where, $a = \dfrac{X}{(\rho CV)_w} + \dfrac{m_e A_w}{(\rho V)_w}$

$$b = \frac{YT_a + Z}{(\rho CV)_w}$$

An analytical solution of the above differential equation for $t = 0$; $T_w = T_{w0}$ is

$$T_w = \frac{1}{a}\left[\left(aT_{w0} + b\right)e^{at} - b\right] \quad (11)$$

The production flux per unit area per unit time may be calculated as:

$$m_p = h_{ew}\left(\beta T_w - \gamma T_{ha}\right) \quad (12)$$

The condensation flux per unit area per unit time may be given by:

$$m_c = h_{cdha}\left(\gamma T_{ha} - \psi T_c\right) \quad (13)$$

3. INDOOR EXPERIMENTS

In order to examine the quasi steady state model for water distillation presented in Chapter 2, indoor simulating experiments were carried out in a thermostatic room to keep a constant temperature and a constant relative humidity at the University of Fukui. The schematic diagram of the experiment is shown in **Fig. 3**. The TSS is comprised of a tubular cover and a semicircular trough. The tubular cover is made of a curled transparent vinyl chloride sheet of 0.5mm in thickness and a transparent polyvinyl chloride bottle at both ends. The tubular cover is 0.52m in length and has an outside diameter of 0.13m. The black trough for storing water in the TSS is made of vinyl chloride with 1.0mm in thickness, 0.1m in outside diameter and 0.49m in length. A solar simulator is comprised with 12

Table 1 Estimated values of heat and mass transfer coefficients

h_{ew} (m/s)	h_{cdha} (m/s)	h_{cw} (W/m^2)	h_{cc} (W/m^2)	h_{rw} (W/m^2)	h_{rc} (W/m^2)
0.0067	0.0023	4.43	3.89	7.59	5.69

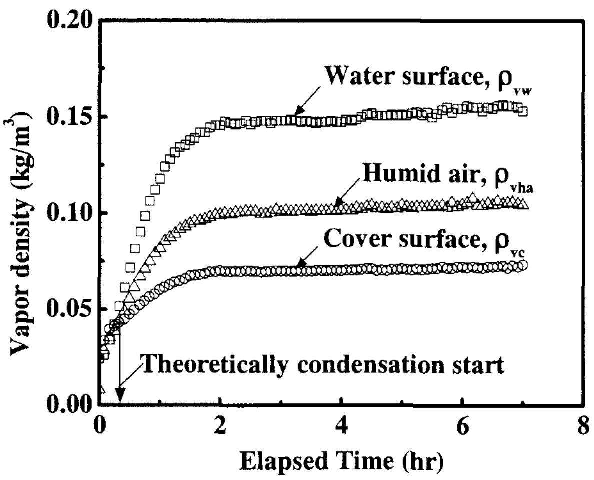

Fig. 4 Time variations of observed vapor densities

incandescent spotlights (125W) arranged in six rows of two lights each, and is used as the energy source to produce the distilled water. All lights were set at a height of 0.3m above the tubular cover top. In this experiment temperatures, relative humidities, and radiation flux were measured by eight thermo-couples, two thermo-hygrometers and a pyranometer, respectively. The data were automatically recorded to a data logger at five minute intervals. The evaporation flux across the water surface in the trough and the production flux from the TSS were measured separately and simultaneously using two electric balances with a minimum reading of 0.01g. A special experimental technique to measure the evaporation flux was developed using a support of the trough on the balance, which was set independently from the other compositions of the TSS. A lid, attached at the end of the tubular cover, is screwed open and the trough can be promptly taken out and easily inserted back into the TSS again after flushing the accumulated salt.

Another experiment was also carried out to evaluate the heat transfer coefficient between the trough and the humid air (h_{tha}). A 10mm thickness vinyl chloride plate (same material as the trough, 161.4 in weight, and a surface area of 268.0cm^2) was prepared for the experiment and heated to 70°C in an oven. The heated plate was then placed in the thermostatic room. Temperature variation in the plate over time was recorded at intervals of five seconds.

4. RESULTS AND DISCUSSIONS

All data of the production experiment were obtained under the constant air temperature and relative humidity of 25°C and 35%, respectively and constant short-wave radiative flux of 1200 W/m^2. It was observed that the evaporation started after twenty minutes elapsed and then forty minutes later, the production followed the evaporation. Quasi steady production was achieved for the last four hours of the experiment duration. Then the values of heat transfer parameters were evaluated using the equations given in Appendix A and shown in **Table 1.**

The heat transfer coefficient between the trough and the humid air (h_{tha}) became as 12.06 W/m^2 °C from an analytical solution derived from the heat energy conservation equation of the heated plate given in Appendix B. The heat transfer coefficient between the trough and the water (h_{tw}) is assumed same as h_{tha}.

Fig. 4 shows the time variations of vapor density at the saline water surface and the tubular cover inner surface and in the humid air. As shown by the arrow in the figure, the beginning of the condensation on the tubular cover inner surface may be detected after twenty minutes elapse when the condensation condition, $\rho_{vha} > \rho_{vc}$ was satisfied. On the other hand, water droplets were visually observed at the almost same time. From this agreement, it was confirmed that the measurement accuracy was satisfactory. The vapor densities ρ_{vw}, ρ_{vha} and ρ_{vc} were high in order after the occurrence of the condensation takes place. It was observed that the relative humidity of the humid air inside the TSS was not saturated since it almost kept 72.0% during the experiment.

T_{ha}, T_c and T_w were analytically calculated using Equations (6), (8) and (11), respectively. These calculated temperatures were compared with the experimental temperatures. The analytical solutions were calculated using the following design parameters:
T_{w0} = 25.8°C; T_a = 25°C; α_b = 1.0; α_c = 0.2; α_{ha} = 0.0; α_w = 0.98; al_b = 0.05; al_c = 0.05; al_w = 0.02; the initial mass of water in the trough = 0.607kg. The reduction in the volume and surface area of the saline water associate with the evaporation was taken account in the model.

Fig. 5 shows the comparison of the calculated temperatures with the observed temperatures. It is seen that the analytically obtained temperatures

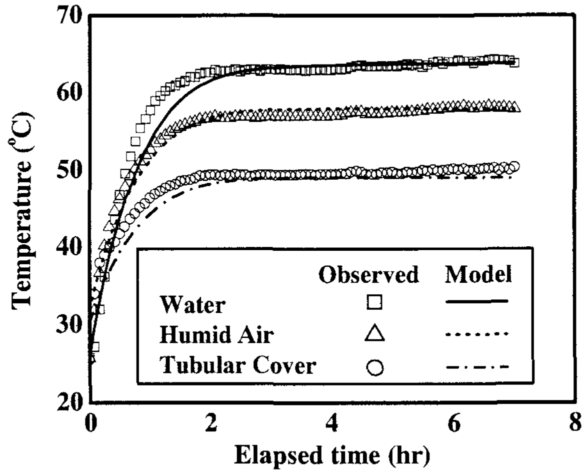

Fig. 5 Comparison of analytically evaluated temperatures with observed temperatures

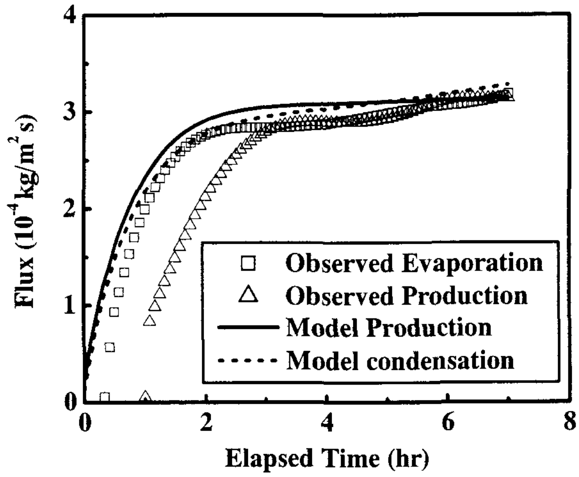

Fig. 6 Comparison of evaporation, production and condensation fluxes

gave a good agreement with the observed temperatures for the period of quasi thermodynamically equilibrium state.

Production and condensation fluxes were analytically evaluated using Equations (12) and (13), respectively and then compared with observed ones (see **Fig. 6**). The evaporation started at early time of the experiment (after twenty minutes) and reached the quasi steady state approximately after two hours elapse. The condensation started after the occurrence of evaporation when the humid air was satisfied the condensation condition, $\rho_{vha} > \rho_{vc}$. The production was observed followed by the occurrence of the condensation when condensed (distilled) water got stored in a collection vessel. A quasi steady production appeared approximately after about three hours elapse. After all, the moisture movement in the TSS brought a time lag of about forty minutes was observed between the beginning of the evaporation and the production. Finally, the observed evaporation and condensation fluxes became equal for the quasi state period and this fact was well reproduced by the proposed model.

5. CONCLUSIONS

A quasi steady heat and mass transfer model of a Tubular Solar Still taking account of humid air properties inside the still is presented in this study. In order to validate the proposed model, an indoor production experiment on a Tubular Solar Still was carried out in a thermostatic room at the University of Fukui. The proposed model can predict the saline water temperature, the humid air temperature, the tubular cover temperature and the production flux under a quasi steady state condition.

APPENDIX A

The various heat and mass transfer coefficients of the TSS are defined as follows:

i) $$h_{ew} = \frac{m_e}{(\rho_{vw} - \rho_{vha})} = \frac{m_e}{(\beta T_w - \gamma T_{ha})}$$

ii) $$h_{cw} = \frac{L h_{ew}(\rho_{vw} - \rho_{vha})}{C(P_{vw} - P_{vha})} = \frac{L h_{ew}(\beta T_w - \gamma T_{ha})}{C(P_{vw} - P_{vha})}$$

where, the coefficient C for the TSS is given by

$$C = 2.92\times10^{-2} - 1.37\times10^{-3}T_w + 5.07\times10^{-5}T_w^2 - 8.08\times10^{-7}T_w^3 + 5.11\times10^{-9}T_w^4$$, Islam et.al.[5]

iii) $$h_{cdha} = \frac{m_p}{(\rho_{vha} - \rho_{vc})} = \frac{m_p}{(\gamma T_{ha} - \psi T_c)}$$

iv) $$h_{rw} = \frac{\sigma\left(T_w'^2 + T_c'^2\right)\cdot\left(T_w' + T_c'\right)}{\frac{1}{\varepsilon_w} + \left(\frac{1}{\varepsilon_c} - 1\right)\cdot\frac{A_w}{A_c}}$$

v) $$h_{rc} = \sigma\varepsilon_g\left(T_c'^2 + T_a'^2\right)\cdot\left(T_c' + T_a'\right)$$

vi) $$h_{cc} = \frac{k}{d}\left(Nu_L^{3.3} + Nu_T^{3.3}\right)^{1/3.3}$$, Raithby et.al.[6]

where,

$$Nu_L = \frac{2f}{ln\left(1 + \frac{2f}{Nu_1}\right)}$$

$$Nu_T = C_3 Ra^{1/3}$$

$$f = 1 - \frac{0.13}{(Nu_1)^{0.16}}$$

$$Nu_1 = 0.772 C_1 Ra^{1/4}$$

APPENDIX B

Energy balance of a trough plate exposed in air with a constant temperature and relative humidity is

$$(\rho CV)_t \frac{\partial T_t}{\partial t} = -Q_{tw} - Q_{tha}$$

An analytical solution of the above differential equation for $t = 0$; $T_t = T_{t0}$ is

$$T_t = T_a + (T_{t0} - T_a)e^{-\frac{h_t A_t t}{(\rho CV)_t}}$$

NOMENCLATURE

A	:	area	(m^2)
A_{tha}	:	contact surface area between trough and humid air	(m^2)
A_{tw}	:	contact surface area between trough and water	(m^2)
al	:	albedo	(–)
C	:	specific heat capacity	(J/kg °C)
h_{cc}	:	convective heat transfer coefficient from tubular cover to atmosphere	(W/m^2 °C)
h_{cdha}	:	condensative mass transfer coefficient from humid air to tubular cover	(m/s)
h_{cw}	:	convective heat transfer coefficient from water surface to humid air	(W/m^2 °C)
h_{ew}	:	evaporative mass transfer coefficient from water surface to humid air	(m/s)
h_{rc}	:	radiative heat transfer coefficient from tubular cover to atmosphere	(W/m^2 °C)
h_{rw}	:	radiative heat transfer coefficient between water surface and tubular cover	(W/m^2 °C)
h_{tha}	:	convective heat transfer coefficient between trough and humid air	(W/m^2 °C)
h_{tw}	:	convective heat transfer coefficient between trough and water	(W/m^2 °C)
l	:	length	(m)
L	:	latent heat of vaporization	(J/kg)
m_c	:	condensation flux	($kg/m^2 \cdot s$)
m_e	:	evaporation flux	($kg/m^2 \cdot s$)
m_p	:	production flux	($kg/m^2 \cdot s$)
Nu	:	Nusselt number	(–)
Q_{cc}	:	convective heat transferred from tubular cover to atmosphere	(J/s)
Q_{cdha}	:	condensative heat transferred from humid air to tubular cover	(J/s)
Q_{cha}	:	convective heat transferred from humid air to tubular cover	(J/s)
Q_{cw}	:	convective heat transferred from water surface to humid air	(J/s)
Q_{ew}	:	evaporative heat transferred from water surface to humid air	(J/s)
Q_{rc}	:	radiative heat transferred from tubular cover to atmosphere	(J/s)
Q_{rw}	:	radiative heat transferred between water surface and tubular cover	(J/s)
Q_{tha}	:	convective heat transferred between trough and humid air	(J/s)
Q_{tw}	:	convective heat transferred between trough and water	(J/s)
R	:	radius	(m)
Ra	:	Rayleigh number	(-)
R_s	:	solar radiation	(W/m^2)
T	:	temperature	(°C)
T'	:	temperature	(°K)
V	:	volume	(m^3)
α	:	absorption coefficient	(–)
ε	:	emissivity	(–)
ρ	:	density	(kg/m^3)
σ	:	Stefan-Boltzman constant, 5.6697×10^{-8}	(W/m^2 °C^4)

Subscripts:

a	:	atmosphere
t	:	trough
c	:	tubular cover
ha	:	humid air
w	:	water

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(Received September 30, 2004)