THERMAL ANALYSIS OF SOLAR DRYERS

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ABSTRACT

A general purpose solar crop dryer for drying various agricultural products such as coffee, fruits, vegetables, medicinal plants, etc. is simulated. The simulated solar collector consists of the solar air heater (solar collector) which uses low-emissivity glass cover and an integral dryer chamber attached to the collector where the products to be dried are placed, Fig. 1. A thermal solar-collector model was developed to determine the available useful energy for heating the ambient air from the available meteorological data such as global and diffuse radiation and ambient air temperature. Basic heat transfer equations for single-plane glass glazing are derived and techniques for the solution of these equations are presented. A computer programme was written to predict the collector outlet temperature, mass flow rate and other engineering variables from the input of the metearologicol data and collector parameters.

Results of the system simulation are presented in graphical form suitable for system performonce determination. From the incident solar flux, ambient air temperature and solar collector parameters, the average yearly values of useful energy, the collector outlet air-temperature and air volume flow-rate are also predicted and presented graphically. In this paper, the theoretical analysis of the drying chamber was effected through an example, as it very much depends on experimentally determined parameters which also depend on the particular type of product to be dried.

INTRODUCTION

Sun drying is still the most common method of preserving agricultural products in most tropical and sub tropical countries. Due to lack of sufficient preservation methods, farmers have to spread the agricultural products in thin layers on paved grounds or on mats where they are exposed to sun and wind. Considerable losses occur during natural sun drying due to various influences such as rodents, birds, insects, rain, storm and micro-organisms. The quality of dried products may also be lowered significantly. Overdrying, contamination by dust and insect infestation are typical examples for natural sun dried products.

The transient nature of solar radiation is a major factor in determining the success, or otherwise, of solar energy devices. This has prompted the development of design methods that attempt to account for the transient nature. Prominent amongst these are methods relying on the direct simulation of the solar conversion system. In this paper, a simulation programme for modeling the performance of solar dryers has been developed.

Referring to Fig. 1, the simulated solar dryer is naturally divided into two parts - the solar air collector system, in which ambient air is heated by solar energy to increase its drying potential, and the drying chamber (or silo), where the crop to be dried is spread out on racks during drying. The first part of simulation programme deals with the air collector model where the input meteorological data are processed for a given collector system to predict the properties of the hot air leaving the collector. The second part of the programme models the drying chamber.

The dryer analysed in this paper uses solar energy for the purpose of supplying hot air to the dryer-cabinet. The temperature, mass flow rate, specific humidity of the supplied hot air to the chamber depend on the design and type of solar collector. The type of solar collector with the integrated dryer-chamber is indicated in Fig. 1. The thermal analysis of the solar-collector is performed using the available solar radiation data - global & diffuse radiation intensities, ambient temperature, etc.

THERMAL ANALYSIS OF FLAT PLATE COLLECTOR

The thermal performance of any type of solar collector can be evaluated by energy balance that determines the portion of the incoming radiation delivered as useful energy to the working fluid, here air.

In order to construct a model suitable for a thermal analysis of a flat-plate collector, the following simplifying assumptions are made.

- The temperature drop between the top and the bottom of the absorber plate is negligible.
- Heat flow is one-dimensional through the cover as well as through the back and side insulations.
- The sky is treated as a black body source for infrared radiation at an equivalent sky temperature.
- The irradiation on the collector plate is uniform.

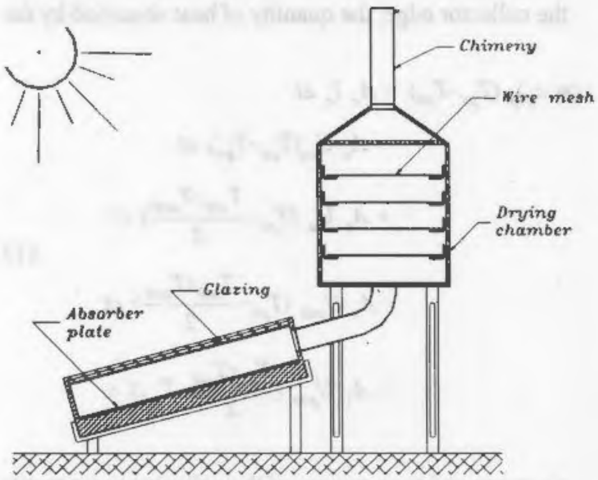

Figure 1 Integrated solar collector and drying chamber

The solar radiation energy I_N incident on the blackened collector surface which is inclined at an angle θ to the horizontal, defined in terms of the global radiation G_r, the diffuse radiation D_r, the beam radiation factor R_b and the ground reflectivity factor ρ, is given by:

$$I_N = R_b\ (\ G_r\ -\ D_r\ )\ +\ 0.5\ (\ 1\ +\cos\theta \qquad + 0.5\ \rho\ (\ 1 - \cos\theta\ )\ G_r \tag{1}$$

The flux collected per unit time which depends on the transimissivity τ of the glass cover (glazing) and the absorbtivity α of the absorber plate is:

$$I_c\ =\ I_N\ (\ \tau\ \ \alpha\ ) \tag{2}$$

The heat exchange processes between the various solar collector components and ambient are represented by the thermal resistance of the simplified collector model Fig. 2 to solve the set of differential equations. These differential equations are derived from simple energy balance on the absorber plate, on the air stream and the glass cover. In its simplest form, a single glass cover is considered for the system simulation.

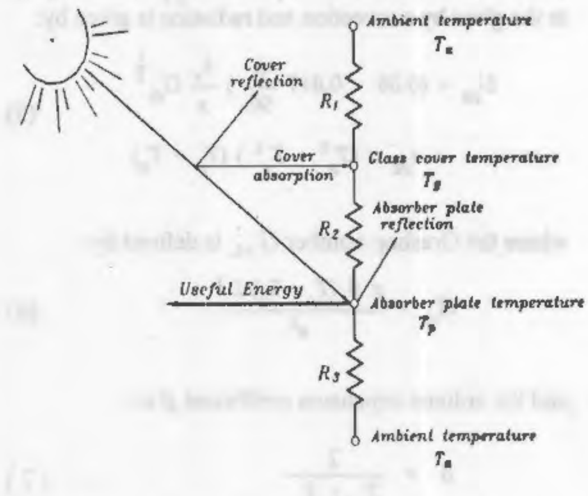

Figure 2 Thermal circuit for solar collector

Due to the temperature dependent radiant and convective heat transfer coefficients, these sets of differential equations are non-linear. An integration scheme which linearises the given set of eqnations within a given time step is applied for solving these equations. The differential equations are evaluated for the various components of the solar collector at time $(t + \Delta t)$ based on available ˈata at time t. The time interval Δt for the system simulation is set to 60 seconds.

ENERGY BALANCE ON THE ABSORBER PLATE

Energy balance on the absorber plate yields the expression:

$$\begin{aligned}(m\ c_p)_p \frac{dT_p}{dt} &= A_c\ I_c - A_c\ U_{pg}\ (T_p - T_g) \\ &- A_c\ U_a\ (T_p - \frac{T_{ai} + T_{ao}}{2}) \\ &- A_c\ U_{pab}\ (T_p - T_a) \\ &- A_1\ U_{poe}\ (\frac{T_p + T_{ai}}{2} - T_{ai})\end{aligned} \tag{3}$$

In the above equation, the coefficient A_1 is defined in terms of the collector perimeter p and the depth of the edge d_e by:

$$A_1\ =\ p\ d_e \tag{4}$$

The overall heat transfer coefficient U_{pg} from the plate to the glass by convection and radiation is given by:

$$U_{pg} = (0.06 - 0.017\,\frac{\theta}{90})\,\frac{k_a}{x}\,G_{rL}^{\frac{1}{3}} + \epsilon_{pg}\,\sigma\,(T_p^2 + T_g^2)\,(T_p + T_g) \tag{5}$$

where the Grashop number G_{rL} is defined by:

$$G_{rL} = \frac{g\,\beta\,(T_p - T_g)\,L^3}{\nu^2} \tag{6}$$

and the volume expansion coefficient β as:

$$\beta = \frac{2}{T_{ai} + T_{ao}} \tag{7}$$

The overall emittance ϵ_{pg} for the absorber plate and the glass cover is obtained from the relation:

$$\epsilon_{pg} = \frac{1}{(\frac{1}{\epsilon_p} + \frac{1}{\epsilon_g} - 1)} \tag{8}$$

The heat transfer coefficient to the air is given by:

$$U_a = 0.664\,\frac{k_a}{L}\left|\frac{P_r\,G_{rL}\,\cos\theta}{P_r + 0.9524}\right|^{0.25} \tag{9}$$

The Prandtl number P_r in Eq. (9) is:

$$P_r = \frac{\nu}{\alpha} = \frac{c_p\,\nu}{k} \tag{10}$$

All properties in the above expressions are evaluated at the film temperature T_f given by:

$$T_f = \frac{T_{ao} + T_{ai}}{2} \tag{11}$$

The heat transfer coefficient from the plate to the edge of the collector cover, defined in terms of the thermal conductivity k_i of the insulation and the edge insulation thickness t_e, is:

$$U_{pae} = \frac{k_i}{t_e} \tag{12}$$

The plate temperature after time interval Δt is determined solving the differential equation given by Eq. (3), with the respective heat transfer coefficients defined in Eqs. (5), (9) and (12):

The temperature of the plate at time $(t + \Delta t)$ in relation to the absorbed useful incident radiation on the surface of the plate, the heat losses through the glass cover and the collector edge, the quantity of heat absorbed by the

$$\begin{aligned}(m\,c_p)_p\,(T_{p1} - T_{p0}) &= A_c\,I_c\,\Delta t \\ &- A_c\,U_{pg}(T_{p0} - T_{g0})\,\Delta t \\ &+ A_c\,U_a\,(T_{p0} - \frac{T_{ai0} + T_{ao0}}{2})\,\Delta t \\ &- A_c\,U_{pab}\,(T_{p0} - \frac{T_{ai0} + T_{ao0}}{2})\,\Delta t \\ &- A_1\,U_{pae}\,(\frac{T_{p0} + T_{ai0}}{2} - T_{ai0})\,\Delta\end{aligned} \tag{13}$$

air stream and temperatures of the collector components at time t is given by:

$$\begin{aligned}T_{p1} &= \frac{A_c\,I_c}{(m\,c_p)_p}\,\Delta t + T_{p0} \\ &- \frac{A_c\,U_{pg}}{(m\,c_p)_p}\,(T_{p0} - T_{g0})\,\Delta t \\ &+ \frac{A_c\,U_a}{(m\,c_p)_p}\,(T_{p0} - \frac{T_{ai0} + T_{ao0}}{2})\,\Delta t \\ &- \frac{A_c\,U_{pab}}{(m\,c_p)_p}\,(T_{p0} - \frac{T_{ai0} + T_{ao0}}{2})\,\Delta t \\ &- \frac{A_c\,U_{pae}}{(m\,c_p)_p}\,(T_{p0} - \frac{T_{ai0} + T_{ao0}}{2})\,\Delta t\end{aligned} \tag{14}$$

ENERGY BALANCE ON THE AIR STREAM

Considering heat transfer from the collector plate to the air-stream and heat transfer from the air-stream to the glazing, energy balance on the stream yields:

$$(m\ c_p)_a \frac{dT_{ao}}{dt} = A_c U_a \left(T_p - \frac{T_{ai}+T_{ao}}{2}\right)$$

$$- A_c U_a \left(\frac{T_{ai}+T_{ao}}{2} \quad T_g\right) \qquad (15)$$

$$(\dot m\ C_p)_a\ (T_{ao} - T_{ai})$$

The temperature of the air stream entering the solar collector at the ambient conditions at time t gains energy from the incident radiant energy on the collector plate. In relation to the inlet air-stream temperature, the heat losses from the plate to the glass cover and the collector edge, the temperature of the air-stream at outlet from the collector at $(t + \Delta t)$ is given by:

$$T_{ao1} = T_{ao0} \left\{ 1.0 - \frac{\Delta t}{(m\ c_p)_a} [\ A_c\ U_a + (\dot m\ c_p \right.$$

$$+ T_{ai0} \frac{\Delta t}{(m\ c_p)_a} \left\{ (\dot m\ c_p)_a - A_c\ U_a \right\} \qquad (16)$$

$$+ T_{p0} \frac{A_c\ U_a}{(m\ c_p)_a} \Delta t + T_{g0} \frac{A_c\ U_a}{(m\ c_p)_a} \Delta t$$

ENERGY BALANCE ON THE GLAZING

From energy balance on the glazing:

$$(m\ c_p)_g \frac{dT_g}{dt} = A_c\ I_N\ (1 - \tau)$$

$$+ A_c\ U_{pg}\ (T_p \quad T_g) \qquad (17)$$

$$- A_c\ U_{ga}\ (T_g - T_a)$$

The glazing temperature at time $(t + \Delta t)$ in relation to the absorbed incident radiation on the surface of the glass, the heat losses from the collector plate to the glazing and heat losses from the glazing to the surrounding at time t is given by:

$$T_{g1} = T_{g0} \left\{ 1.0 - \frac{\Delta t}{(m\ c_p)_g} [\ A_c\ U_{pg} + A_c \right.$$

$$+ \frac{A_c\ I_N\ (1 - \tau)}{(m\ c_p)_g} \Delta t + T_{p0} \frac{A_c\ U_{pg}}{(m\ c_p)_g} \qquad (18)$$

$$+ T_{a0} \frac{A_c\ U_{ga}}{(m\ c_p)_g} \Delta t$$

MASS FLOW RATE OF AIR

The determination of the mass flow rate at exit from the collector would constitute boundary conditions at inlet to the drying chamber along with the exit temperature and specific humidity. The mass flow rate of the air stream at exit from the collector is determined from the flux collected per unit time I_c, the temperature difference between the heated air T_{ao} and the ambient (incoming) air temperature T_{ai} as:

$$\dot m_t - (\dot V)_a\ \rho_a$$

$$- \frac{A_c}{c_{pa}} \left\{ \frac{I_c}{(\ T_{ao} - T_{ai}\ )} - U_a \right\} \qquad (19)$$

THE DRYING MECHANISM

In the drying process, water is removed from material to be dried until its required final moisture content for longer period of storage is reached. The material to be dried is placed on a screen (wire-mesh) in the drying chamber (cabinet), Fig.1, of the drier. The heated air from the solar collector passes upward through the bed of the material to be dried.

During the evaporation of water from the surface of the material to be dried into the air stream, a quantity of heat, equal to the latent heat of evaporation plus the sensible heat necessary to briag the water to the temperature of the air is abstracted from the surrounding. It is only when, the two types of transport phenomena - the heat flow into the material surface and the mass flow of water from the surface of the material into the air stream, are in balance that a steady state is observed.

The system analysis is simplified by considering the air stream supplied from the solar collector as the only source of heat for the drying process.

The following simplifying assumptions are made in the derivation of the governing equations of drying in the separate drying chamber.

- The system is assumed adiabatic.
- The drying is for the unbound moisture in the wet product.
- A bed of uniform cross-sectional area A_b [m ^{2}] is considered.
- Air at flow rate of $\dot m_a$ [kg dry air/ hr] per m^2 of

cross-sectional area enters the chamber of the dryer at a specified specific humidity ω_1.

As the heated air from the solar collector moves upward through the bed, its temperature T as well as the specific humidity ω vary through the bed, Fig. 3. The air stream which enters the packed bed at temperature T_1 and specific humidity ω_1 leaves at T_2 and ω_2. Hence, the temperature T and specific humidity ω vary along the bed. The determination of ω_2 and hence temperature T_2 depends on which one of the two types of drying periods - *the constant-rate drying period or the falling-rate drying period* - prevail, on the mass flow rate of the heated air stream, its temperature and specific humidity at section 1 - 1, Fig. 3.

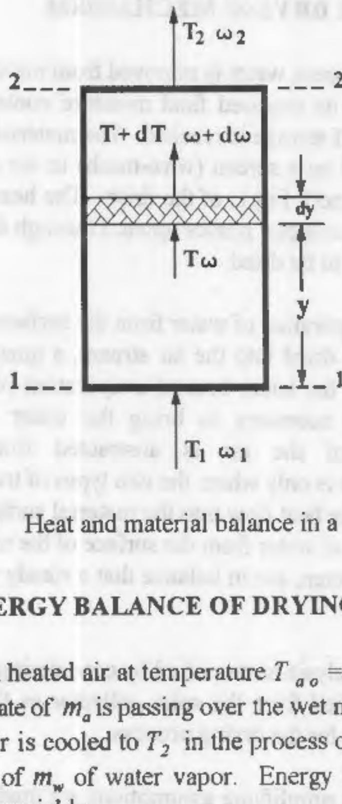

Figure 3 Heat and material balance in a dryer

ENERGY BALANCE OF DRYING PROCESS

When heated air at temperature $T_{ao} = T_1$ with a mass flow rate of $\dot{m}_a$ is passing over the wet material, Fig. 3, the air is cooled to T_2 in the process of evaporating a mass of $\dot{m}_w$ of water vapor. Energy balance for the process yields:

$$\dot{m}_w \, h_{fg} = \dot{m}_a \, c_{pa} \, (T_1 - T_2)$$

$$where \quad \dot{m}_a = \frac{\dot{m}_1}{(1 + \omega)} \tag{20}$$

In Eq. (20), the latent heat of evaporation h_{fg} and the specific heat capacity c_{pa} of the air stream are evaluated at the mean value of T_1 and T_2. The temperature of moist air leaving the first tray and entering the second tray T_2 can be determined from Eq. (20) if $\dot{m}_w$ - the rate at which water evaporates from the material to be dried - is known. Conversely, if T_2 - the temperature at which the air leaves the first tray is known, then $\dot{m}_w$ - the rate of evaporation of water from the product - can be determined. Similar procedure is applied to the rest of the trays.

DISCUSSION OF RESULTS

The thermal analysis of the solar dryer has been carried out taking Addis Ababa as the simulation site. The dimensions of a 2 m^2 solar-collector, the edge-depth of the collector, the insulation thickness, etc. have been optimized for the thermal analysis of the solar dryer. Meteorological data of Addis Ababa, such as ambient temperature, global and diffuse radiation, for 5 years have been averaged and used in the analysis. January 1st has been taken as the start of the thermal simulation with the effects of energy losses during the nights also considered.

Results of the thermal analysis of the solar collector are represented in Fig. 4 to Fig. 7. In these figures, the solar flux $\dot{Q}_{in}$, the temperature of the plate T_p, the ambient (collector inlet) air temperature T_a, the glass temperature T_g and the collector outlet air-stream temperature T_o are indicated for the months of January, April, July and October as functions of solar-time.

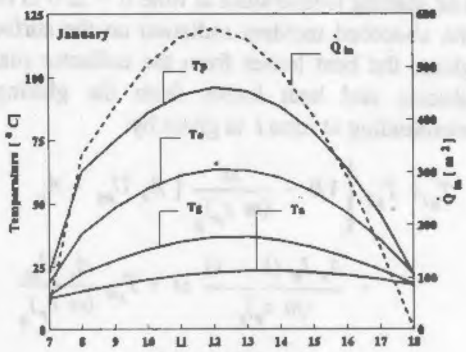

Figure 4 Ambient, glass, collector outlet and plate temperatures for January

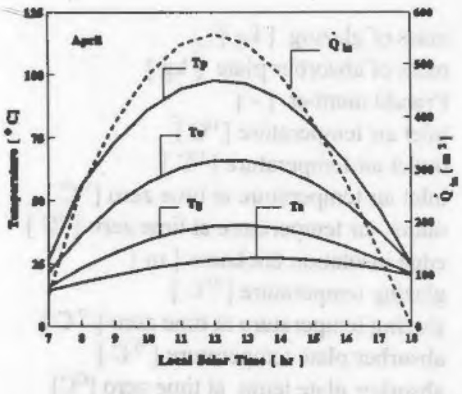

Figure 5 Ambient, glass, collector outlet & plate temperatures for April

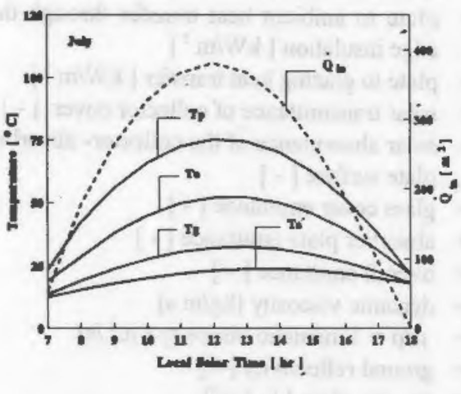

Figure 6 Ambient, glass, collector outlect & plate temperatures for July

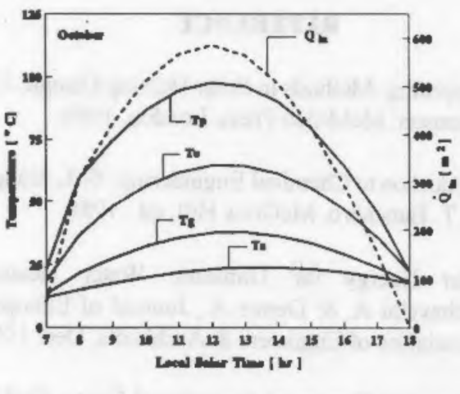

Figure 7 Ambient, glass, collector outlet & plate temperature for October

Averages of these values over the year are represented in Figure 8.

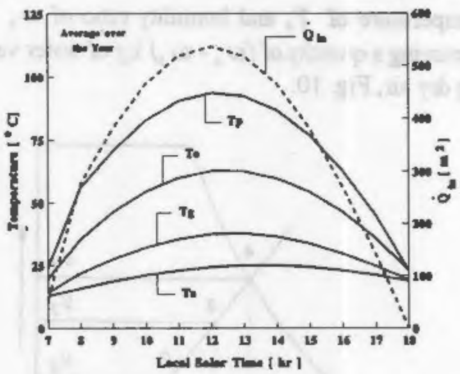

Figure 8 Average yearly values of temperatures and solar flux

In Figure 9, the yearly average outlet temperature T_o and the mass flow rate m_o as well as the mean of these yearly average values are represented as T_a and m_{av} respectively.

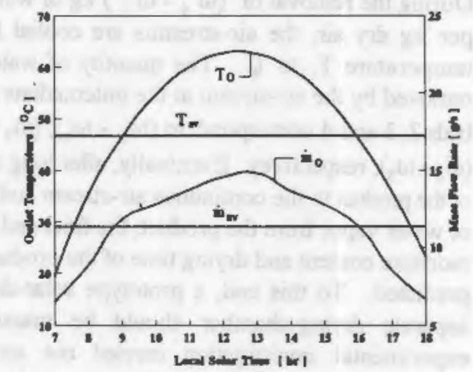

Figure 9 Yearly average & mean values of mass flow rate and temperature

The drying mechanism in the separate drying chamber depends, among other parameters, on the prevailing drying period, on the type of product to be dried, on the original water content of the product, the final and required water content of the product, etc. These parameters would require intensive experimental investigation on the particular product under consideration. The theoretical analysis of the drying process in the separate drying-chamber is demonstrated with the help of an example. The drying chamber is assumed to be built with 4 packed beds. The air-stream leaving the solar collector is assumed to enter the

drying-chamber at state 1 at a temperature of T_1 and **humidity ratio of** ω_1 and leave as saturated air at **temperature of** T_4 and humidity ratio of ω_4 , after **removing** a quantity of $(\omega_4 - \omega_1)$ kg of water vapor per **kg dry air**, Fig. 10.

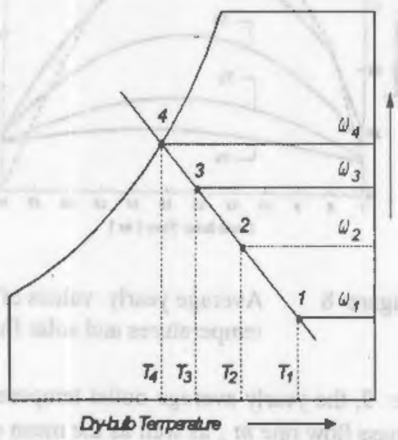

Figure 10 Psychrometric representation of drying mechanism

During the removal of $(\omega_4 - \omega_1)$ kg of water vapor per kg dry air, the air-streamis are cooled from the temperature T_1 to T_4. The quantity of water vapor removed by the air-stream at the imtemediate packs of beds 2, 3 and 4 correspond to $(\omega_2 - \omega_1)$, $(\omega_3 - \omega_2)$ and $(\omega_4 - \omega_3)$, respectivey. Eventually, after long exposure of the product to the continuous air-stream and removal of water vapor from the product, the final and required moisture content and drying time of the product can be predicted. To this end, a prototype solar-dryer with separate drying-chamber should be manufactured, experimental investigation carried out on various agricultural products such as coffee, vegetables, etc. and the drying period estimated.

NOMENCLATURE

A_c = area of a collector [m^2]
c_{pg} = specific heat of glazing [kJ/kg K]
c_{pp} = specific heat of plate [kJ/kg K]
c_{pa} = specific heat of air [kJ/kg K]
g = gravity of the earth [m / s^2]
G_{rL} = Grashof number [-]
I_c = I_n $(\tau\ \alpha)$ = solar irradiation on collector surface [Wh/m^2]
k_a = thermal conductivity of air [kJ/kg K]
k_i = thermal conductivity of insulation [kJ/kg K]
L = length of absorber plate [m]
m_a = mass of air [kg]
m_g = mass of glazing [kg]
m_p = mass of absorber plate [kg]
P_r = Prandtl number [-]
T_{ai} = inlet air temperature [^oC]
T_{ao} = outlet air temperature [^oC]
T_{ai0} = inlet air temperature at time zero [^oC]
T_{ao0} = outlet air temperature at time zero [^oC]
t_e = edge insulation thickness [m]
T_g = glazing temperature [^oC]
T_{g0} = glazing temperature at time zero [^oC]
T_p = absorber plate temperature [^oC]
T_{p0} = absorber plate temp. at time zero [^oC]
Δt = time interval [sec]
U_{ga} = glazing to ambient heat transfer [kW/m^2]
U_{pab} = plate to ambient heat transfer through the back insulation [kW/m^2]
U_{pae} = plate to ambient heat transfer through the edge insulation [kW/m^2]
U_{pg} = plate to glazing heat transfer [kW/m^2]
τ = solar transmittance of collector cover [-]
α = solar absorptance of the collector- absorber plate surface [-]
ϵ_g = glass cover emittance [-]
ϵ_p = absorber plate emittance [-]
ϵ_{pg} = overall emittance [-]
μ = dynamic viscosity [kg/m s]
ν = μ/ρ = kinematic viscosity [m^2 /s]
ρ = ground reflectivity [-]
ρ_a = density of air [kg/ m^3]
σ = Stefan Botzman constant [5.67×10^{-8} W/m^2 K]
θ = inclination of the solar collector [^{o}]

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