ANALYTICAL STUDIES OF CROP DRYING CUM WATER HEATING SYSTEM

Centre for Energy Studies, Indian Institute of Technology, Hauz Khas, New Delhi 110 016, India

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Abstract—In this communication, a new design of thin layer bed crop drying cum water heater has been proposed and analysed for making the whole system operate throughout the year. Energy balance equations for each component of the system have been used to predict the analytical results. The system can be used to provide hot water in case the drying system is not in operation. The water heater below the air heater systems will act as a storage material for drying the crop during off sunshine hours.

Solar energy  Air heater  Drying system  Water heater

NOMENCLATURE

$A_s$ = Area of solar collector (m$^2$)
$A_r$ = Area of reflecting surface (m$^2$)
$A_w$ = Side area of water tank (m$^2$)
$A_k$ = Side area of walls of drying chamber (m$^2$)
$b$ = Breadth of collector (m)
$c$ = Constant for drying product
$C_p$ = Specific heat of air (J/kg °C)
$C_v$ = Specific heat of humid air (J/kg °C)
$C_w$ = Specific heat of drying product (J/kg °C)
$C_f$ = Specific heat of water (J/kg °C)
$t_d$ = Time interval (s)
$dx$ = Sectional element along length of collector
$E$ = Equivalent moisture content left in drying product (kg)
$h_c$ = Convective heat transfer coefficient between blackened surface and air (W/m$^2$ °C)
$h_t$ = Overall heat transfer coefficient from water to air through bottom insulation (W/m$^2$ °C)
$h_n$ = Convective heat transfer coefficient from product to enclosed air (W/m$^2$ °C)
$h_l$ = Convective heat transfer coefficient between blackened surface and water (W/m$^2$ °C)
$h_h$ = Overall heat transfer coefficient from enclosed air of drying chamber to air through drying walls (W/m$^2$ °C)
$h_l$ = Overall heat transfer coefficient from flowing air to ambient through glass cover (W/m$^2$ °C)
$I(t)$ = Solar radiation incident on solar collector (W/m$^2$)
$I'$ = Solar radiation incident on reflector (W/m$^2$)
$L$ = Length of solar collector (m)
$M_s$ = Mass of material to be dried (kg)
$M_w$ = Mass of water (kg)
$m_a$ = Mass flow rate of air through collector (kg/s)
$m_w$ = Mass flow rate of air through chamber (kg/s)
n = Constant for drying product
$p(T)$ = Partial vapor pressure at temperature $T$ (N/m$^2$)
$t$ = Time interval (s)
$T_a$ = Ambient air temperature (°C)
$T_{in}$ = Temperature of air inside collector (°C)
$T_{in}$ = Temperature of air inside collector at $x = 0$ (°C)
$T_{out}$ = Temperature of air inside collector at $x = L$ (°C)
$T_b$ = Temperature of blackened surface (°C)
$T_h$ = Temperature of humid air in drying chamber (°C)
$T_p$ = Temperature of product in drying chamber (°C)
$T_w$ = Temperature of water (°C)
$t = Position coordinate along flow direction (m)
INTRODUCTION

The work reported for an open sun dryer has been done by various workers [1, 2], and it has been reported that the open sun dryer has the following disadvantages.

(i) drying time is large,
(ii) crops get damaged due to rain,
(iii) contamination of crop because of dust,
(iv) attack by insects and pests.

Some of the above problems have been solved by designing a solar cabinet crop dryer [2-8]. Further, the cabinet dryer has the following disadvantages.

(i) It is suitable only for small capacity of the crop, hence cannot be used for commercial purposes.
(ii) It can be used only during sunny periods.
(iii) Due to evaporation of moisture and its condensation on the glass cover, the transmittivity of the glass cover is reduced.
(iv) Overheating of the crop may take place due to direct exposure to sunlight, and hence, the quality of the product may deteriorate.

In order to solve the above problems, various designs of a passive solar crop dryer have been developed and tested [8-20]. These designs have been recommended for commercial purposes and these include:

(i) Chimney dryer,
(ii) bin-type dryer,
(iii) rack-type dryer.

In their studies, the storage effect in the drying chamber has also been included to reduce the fluctuation in outlet temperature of the hot air. Further, the stored thermal energy is also used for drying purposes during off sunshine hours. It is important to mention here that the storage material used in the drying chamber is mostly the rock bed. This makes the system very difficult to maintain in case of air blockage in the drying chamber. In order to solve this problem and to make the system more economical from the point of view of thermal energy utilization, the new design for crop drying (Fig. 1) has been proposed and analysed.

In this communication, energy balance equations of the air cum water heater and drying chamber have been written separately. Based on these energy balance equations, the performance of the air cum water heater and drying chamber have been discussed analytically. Numerical computations have also been performed for a typical set of design and climatic parameters.

WORKING PRINCIPLE OF THE PROPOSED SYSTEM

The solar radiation, incident directly as well as reflected from the reflector of the vertical wall of the drying chamber, is absorbed by the absorber after transmission through the glazed surface. Part of the absorbed radiation is convected to the water mass beneath the absorber and the rest is convected to the flowing air above the absorber. Thus, the water and the air both get heated. The hot air is allowed to pass through different layers of crop in a drying chamber. The hot air
carries away the moisture content of the crop and then the moist air is allowed to pass through the openings provided at the top of the drying chamber. In this case, the flowing air is heated at a reasonable temperature with minimum fluctuation, depending on the capacity of the water mass beneath the absorber. This temperature can be increase by reducing the water mass. The air can also be heated during off sunshine hours due to the reverse heat flow from the hot water to the flowing air above the absorber.

The advantages of the proposed system are as follows:

(i) The fluctuation and the magnitude of the hot air temperature at the outlet of a heater is reduced at the desired temperature.

(ii) The air can also be heated during off sunshine hours due to the storage effect in the water mass.

(iii) The solar radiation is enhanced due to provision of the reflector at the vertical wall of the drying chamber.

(iv) The hot water available can be used during non-operating periods of the system as a dryer, etc.

The cross-sectional view of the system has been shown in Fig. 1(a).

**THERMAL ANALYSIS**

In order to write energy balance equations for the proposed system [Fig. 1(a)], the following assumptions have been made.

(i) Thermal capacities of the glass cover, absorbing material and chimney material are negligible.

![Diagram of the crop drying cum water heating system](image)

*Fig. 1. (a) Cross-sectional view of the crop dryer cum water heater. (b) Cross-sectional view of flowing air over the water heater*
(ii) There is no stratification along the depth of the air collector, storage material (e.g. water, [21]) and the crops. This can be achieved either by keeping a small thickness of the water mass ($\approx 0.10$ m) and cross or stirring the crop from time to time.

(iii) The system is perfectly insulated.

(iv) There is no air and water leakage from the collector.

(v) The outlet temperatures of the solar collector is equal to the chimney inlet temperature.

The energy balance equations for the different components of the solar air dryer are as follows:

(a) absorber

$$\alpha \pi [I(t)A_e + \rho \Gamma(t)A_i] = h_u(T_b - T_u)A_e + h_s(T_b - T_s)A_s,$$

(b) water mass

$$h_u(T_b - T_u)A_e = M_w C_w \frac{dT_w}{dt} + (A_i + A_e)h_s(T_w - T_s),$$

(c) flowing air [Fig. 1(b)]

$$h_s(T_b - T_s)b \frac{dx}{dx} = m_A C_A \frac{dT_A}{dx} + h_s(T_A - T_s)b \frac{dx}{dx}.$$

From equation (1), an expression for the absorber temperature can be obtained, and it is given by

$$T_b = \frac{(\alpha \pi)[I(t)A_e + \rho \Gamma(t)A_i] + h_u A_i T_u + h_s A_s T_s}{h_u A_e + h_s A_s}.$$

After substitution of $T_b$ from the above equation in equation (2), one obtains

$$\frac{dT_w}{dt} + a T_w = F(t) + b T_s,$$

where $a$, $b$ and $F(t)$ can be obtained after algebraic simplification, and expressions for these are given below:

$$a = \frac{1}{M_w C_w} \left[ (A_i + A_e)h_s + \frac{h_u h_s A_e}{h_u + h_s} \right],$$

$$b = \left[ \frac{h_s A_i h_u}{h_u + h_s} \right] \frac{1}{M_w C_w},$$

and

$$F(t) = \frac{1}{M_w C_w} \left[ (A_i + A_e)h_s T_u + \frac{h_u \alpha \pi [I(t)A_e + \rho \Gamma(t)A_i]}{h_u + h_s} \right].$$

Similarly, equation (3) can be rewritten after eliminating $T_b$, and it can be expressed as follows.

$$\frac{dT_A}{dx} + a' T_A = F'(t) + b' T_w,$$

where

$$a' = \frac{b \frac{dx}{dx}}{m_A C_A} \left( h_s + \frac{h_u h_s}{h_u + h_s} \right),$$

$$b' = \frac{1}{M_A C_A} \left( h_s b \frac{dx}{dx} \right).$$
and

\[ F(t) = \frac{b}{m_A C_s} \left( h_l T_A + \frac{h_A x [f(t)A_e + \rho I(t)A_e]}{h_A A_e + h_e A_e} \right). \]

Equation (5) is a first-order differential equation which can be integrated between the 0–x interval to obtain the solution for \( T_x \). The expression for \( T_A \) is given by

\[ T_A = \frac{[F(t)+b'T_x]}{a'} \left( 1 - e^{-a't} \right) + T_{A0} e^{-a't}. \tag{6} \]

The outlet air temperature at \( X = L \) can be obtained by using the above equation as follows:

\[ T_{A0a} = T_A|_{x=L} = \frac{F(t)+b'T_x}{a'} \left( 1 - e^{-a'L} \right) + T_{A0} e^{-a'L}. \tag{7} \]

An average value of the flowing air temperature \( \bar{T}_A \) can also be used to evaluate the upward heat transfer coefficient. The expression for \( \bar{T}_A \) is given by

\[ \bar{T}_A = \frac{1}{L} \int_0^L T_A \, dx = \frac{[F(t)+b'T_x]}{a'} \left[ 1 - \frac{1 - e^{-a'L}}{a'L} \right] + T_{A0} \frac{1 - e^{-a'L}}{a'L}. \tag{8} \]

After substituting \( \bar{T}_A \) from the above equation in equation (4) and rearranging the various terms, equation (4) becomes

\[ \frac{dT_x}{dT} + aT_x = g(t). \tag{9a} \]

where

\[ a = \left[ a - \frac{bb'}{b} \left\{ 1 - \frac{1 - e^{-a'L}}{a'L} \right\} \right] \]

\[ g(t) = F(t) + b \left\{ \frac{F(t)}{a'} \left[ 1 - \frac{1 - e^{-a'L}}{a'L} \right] + T_{A0} \left\{ 1 - e^{-a'L} \right\} \right\}. \]

After integrating equation (9a) between the 0–t time interval and considering \( g(t) \) a constant for the 0–t time interval using its average value \( \bar{g}(t) \), the solution of equation (9a) can be written as

\[ T_x = \frac{\bar{g}(t)}{a} \left( 1 - e^{-a't} \right) + T_{A0} e^{-a't}. \tag{9b} \]

Now, an average value of \( T_x \), i.e., \( \bar{T}_x \), can be expressed as follows:

\[ \bar{T}_x = \frac{1}{t} \int_0^t T_x \, dt = \frac{\bar{g}(t)}{a} \left[ 1 - \frac{1 - e^{-a't}}{a_t} \right] + T_{A0} \frac{1 - e^{-a't}}{a_t}. \tag{10} \]

After substituting \( \bar{T}_x \) and \( T_x \) from equations (9b) and (10) in equations (6) and (7), an expression for \( T_A \) and \( T_{A0a} \) can be obtained in terms of design and climatic parameters.
The outlet air at \( x = L \) [equation (7)] becomes the inlet air for the drying chamber, and the flow rate of hot air entering the drying chamber can be controlled by adjusting the shutter provided below the chamber. It can be discussed in the next section.

**Drying chamber**

**Drying air heater:** Useful energy available at outlet of collector becomes the input energy for the drying chamber, and it is given by

\[
Q_a = \dot{m}_a C_A (T_{A_{in}} - T_{ao}) = \dot{m}_C C_C (T_{CH} - T_p). \tag{11}
\]

The above equation can be used to evaluate the moisture content evaporated from the crop placed in the drying chamber using the following energy balance for the drying product.

**Drying product:**

\[
\dot{m}_C C_C \left( T_{CH} - T_p \right) = \dot{M}_C C_p \frac{dT_p}{dt} + h_{eq} A_p (T_p - T_{CH}) + 0.016 h_{eq}[p(T_r) - rp(T_{CH})] A_p + h_{eq} A_i (T_{CH} - T_i), \tag{12}
\]

where \( p(T_p) \) and \( p(T_{CH}) \) are partial vapor pressures at the product and chamber temperatures, respectively, and these can be linearized for the operating temperature range as given below:

\[
p(T_p) = R_1 T_p + R_2 \tag{13a}
\]

\[
p(T_{CH}) = R_3 T_{CH} + R_4. \tag{13b}
\]

Here, \( R_1 \) and \( R_2 \) can be obtained by curve fitting from steam table data.

Equations (11) and (12) can be simultaneously solved for the crop product temperature \( (T_p) \), and its average value \( (\bar{T}_p) \) can also be obtained analytically as done earlier for \( T_A \) and \( T_{ao} \) as a function of design parameters of the drying chamber and various thermal loss coefficients. After knowing \( \bar{T}_p \), the equivalent moisture content \( (E) \) in the crop can be evaluated from the following expression:

\[
1 - r = \exp(-c_{E} \bar{T}_p E^p), \tag{14}
\]

where \( c \) and \( n \) depend on the type of crop to be dried.

**ANALYTICAL RESULTS AND DISCUSSION**

**Analytical study of air heater**

In order to study the analytical results, the following cases have been evaluated:

Case I: If \( a'L \ll 1 \) in equation (8), i.e. a small length of air collector and large flow rate, then

\[
\frac{1 - e^{-xL}}{a'L} \rightarrow 1
\]

and equation (8) reduces to \( T \rightarrow T_{ao} \). Therefore, there is no heating of the air for such condition. So, the small length of collector for large flow rate case is ruled out. These results are in accordance with expected results.

Case II: If \( a'L > 1 \) in equation (8), i.e. a larger length of air collector for small flow rate, then

\[
1 > \frac{1 - e^{-xL}}{a'L} > 0,
\]

and equation (8) becomes a strong function of the climatic and design parameters. In this case, the solar radiation is properly used for air heating which is our requirement. The hot air is then used for crop drying in a drying chamber.
Table 1. Design parameters and various heat transfer coefficients for the proposed dryer

<table>
<thead>
<tr>
<th>Sl No.</th>
<th>Design parameters</th>
<th>Heat transfer coefficients</th>
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<tbody>
<tr>
<td>1.</td>
<td>(\alpha = 0.9), (t = 0.9), (\rho = 0.8)</td>
<td>(h_a = 100, \text{W/m}^2, \text{°C}), (h_s = 25, \text{W/m}^2, \text{°C})</td>
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<tr>
<td>2.</td>
<td>(A_t = A_i = b \times L_e, b = 1, \text{m})</td>
<td>(h_a = 0.8, \text{W/m}^2, \text{°C}), (h_s = 6.0, \text{W/m}^2, \text{°C})</td>
</tr>
<tr>
<td>3.</td>
<td>(M_s = 100, \text{kg}), (C_s = 4190, \text{J/kg}, \text{°C})</td>
<td>(h_a = 0.08, \text{W/m}^2, \text{°C}), (h_s = 5.7, \text{W/m}^2, \text{°C})</td>
</tr>
<tr>
<td>4.</td>
<td>(A_s = 2.6, \text{m}^2, \dot{m}_s = 0.02, \text{kg/s})</td>
<td>(R_i = 325, \text{N/m}^2, \text{°C}, R_l = -5154, \text{N/m}^2)</td>
</tr>
<tr>
<td>5.</td>
<td>(C_s = C_s h = 1.01 \times 10^5, \text{J/kg}, \text{°C})</td>
<td>(t = 3600, \text{s})</td>
</tr>
<tr>
<td>6.</td>
<td>(\dot{m}_{qs} = 0.02, \text{kg/s})</td>
<td>(C = 10.6 \times 10^5, n = 3.03) (wheat) [22]</td>
</tr>
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<td></td>
<td>(r = 0.5-0.8)</td>
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Analytical study of water heater

It is necessary to predict the performance of the water heater along with the air collector for Case II as discussed above.

Case I: If \(a_t \ll 1\) in equation (10), i.e. a small time interval for large water mass, then

\[
\frac{1 - e^{-a_t t}}{a_t t} \gg 1,
\]

and equation (10) reduces to \(T_s = T_{sa}\). It is not a required condition due to nonfunctioning of the water heater.

Case II: If \(a_t t > 1\) in equation (10), i.e. a large time interval for small water mass, then

\[
1 > \frac{1 - e^{-a_t t}}{a_t t} > 0
\]

and equation (10) becomes a strong function of the climatic and design parameters. In this case, the proposed system can be used for crop drying as well as water heating. Further, it is important to report that there will not be any stratification for the small water mass which supports assumption (2).

Similar conditions can be obtained for the average crop product temperature \((T_p)\).

NUMERICAL RESULTS AND DISCUSSION

In order to calculate the moisture content \((E)\) in percentage from equation (14), an average drying product temperature \((T_p)\) has been computed by using equation (12) with the help of other equations [(7), (10) and (11)]. The design parameters used for numerical computations have been given in Table 1. Climatic data for a typical day in summer conditions of Delhi have also been used. The design data of Table 1 is based on Case II for the air and water heating system and used for analytical results and discussion.

Fig. 2. Variation in daily air and water temperatures with and without reflector.
The hourly variations of the water and outlet air temperature with and without reflecting mirror have been shown in Fig. 2. It is clear that there is an appreciable increase in the outlet air and water temperature due to the reflecting mirror which enhances the input energy for the drying chamber. During off-sunshine hours, the air cum water system is covered with movable insulation to avoid night losses and properly use the stored energy for air heating purposes only. It has been further noticed that the water and air temperature decreases with an increase of flow rate which is in accordance with Case I of the analytical results and discussion.

Figures 3 and 4 represent the variations of moisture content ($E$) with time of day with and without the reflecting mirror for different masses of the crop product. The results are in our expectation. The moisture content removed decreases with an increase of mass for a given input energy. For a given capacity of crop-product, the present model can be used to design a system described in the paper.

Other parametric studies, namely the effects of water mass, relative humidity inside drying chamber, reflectivity of mirror, length of air heater and absorptivity of absorber, have also been performed. The results are as follows:

(i) The drying time (number of days) increases with an increase of $r$ inside the chamber due to the reduced removal of moisture from the product.

Fig. 3. Variation in daily moisture content with and without reflector.

Fig. 4. Variation in daily moisture content with mass of the crop.
(ii) There is not a significant effect on drying time due to an increase of water mass after 0.10 m depth, except a decrease of its temperature for large flow rates of air.

(iii) The outlet air temperature becomes constant after 3.5 m length of air collector.

(iv) There are about 7.8.1 decrease at the end of day (early next day morning) in removal of moisture from the product when r is decreased from 0.9 to 0.5.

(v) The decrease of absorptivity reduces the outlet air temperature very significantly.

REFERENCES