PARAMETRIC STUDY OF A REVERSE FLAT PLATE ABSORBER CABINET DRYER: A NEW CONCEPT

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Abstract—In this communication, the concept of reverse flat plate collector has been used as a heating medium of air for the drying of agricultural products in a cabinet dryer. The reverse flat plate absorber is a non-concentrating collector which can collect solar heat at high temperature unlike conventional non-concentrating collectors. The thermal performance of the proposed dryer is analyzed by solving the various energy balance equations. An attempt has been made to optimize the vent area of the dryer for speedy flow of humid air from the drying chamber to the atmosphere. In order to have parametric studies, numerical computations have been carried out for a typical day in June for Delhi climatic conditions. The performance of this system is compared with that of conventional cabinet dryers. It is found that the reverse flat plate absorber dryer gives the better performance.

1. INTRODUCTION

The conventional method of crop drying, in most of the developing countries, is to spread the agricultural produce on the ground and allow it to dry in the open sun. Such studies have been conducted by many workers (Sodha et al., 1985; Sodha and Chandra, 1994), and it has been reported that open sun drying has the following disadvantages: (i) it requires both a large amount of space and long drying times; (ii) the crop becomes damaged because of hostile weather conditions; (iii) contamination of crop from foreign materials; (iv) the crop is subject to insect infestation and (v) the crop is susceptible to readsorption of moisture if it is left on the ground during periods of no sun which reduces its quality.

Some of these problems have been solved using a conventional solar dryer (Lawand, 1966; Sharma et al., 1986, 1990; Tiwari et al., 1994). Further, the conventional cabinet dryer also has some disadvantages, such as: (i) it is only suitable for a small capacity, hence cannot be used for commercial purposes; (ii) overheating of the crop may take place as the crop is exposed to solar radiation directly; (iii) the transmissivity of the glass cover is reduced because of moisture condensation on it; (iv) the absorptivity of the absorber plate is reduced as part of the solar radiation is reflected from the uneven crop surface and part of it absorbed by the crop before it reaches the absorber and, lastly, (v) the increase in crop temperature may not be sufficient to allow moisture to evaporate, especially when the sky is not clear.

In order to overcome some of these problems, the reverse flat plate absorber as the air heating medium could be introduced. It is a non-concentrating type air heater which allows high temperature applications (Goel et al., 1987; Madhusudan et al., 1981; Tiwari, 1986).

In the present study, an attempt has been made to develop a model using both a reverse flat plate absorber as the heating medium and a cabinet dryer as the drying chamber. The whole unit is termed a reverse absorber cabinet dryer (RACD) and is shown in Fig. 1. None of the reported studies have considered the effect of vent area, therefore efforts have been made to optimize the vent area to further improve the thermal performance of the system under the natural mode of operation.

Thermal analysis has been carried out by solving the energy balance equations using the finite difference technique. A mathematical expression is proposed for RACD and results are compared with those for a normal cabinet dryer.

Numerical computations have been carried out for optimization of the design and climatic parameters for a typical June day under Delhi climatic conditions.

2. WORKING PRINCIPLE

The principle of a reverse flat plate absorber cabinet dryer is shown in Fig. 1. The absorber plate is horizontal and downward facing. A cylindrical reflector is placed under it to
introduce solar radiation from below. The area of the aperture is the same as that of the absorber plate. The cabinet dryer is mounted on top of the absorber maintaining a gap of 0.03 m for air to flow above the absorber which becomes heated and enters the dryer from the bottom. Unlike a conventional dryer it is not insulated from the bottom but it does not allow insulation from the top. The bottom area of the dryer is equal to that of the absorber plate area. The length-to-width ratio of the dryer is taken as 3:1 to avoid the shading effect (Sodha and Chandra, 1994).

Firstly, hot air heats the crop spread over a wire mesh and then moisture starts moving from the interior of the kernel to the surface and then to the chamber. Secondly, moisture-laden air exits the chamber through the vent because of the vapour pressure difference between the chamber and the outside in the natural mode of operation. An advantage of this assembly is that a selective coating can be used for the absorbing surface which is not possible in the case of conventional cabinet dryers. As the crop is not subjected to direct contact with solar radiation, its quality can be maintained.

3. THERMAL ANALYSIS

The analysis of the configurations as shown in Fig. 1 is carried out by writing the energy balance equations for each component of the reverse flat plate absorber cabinet dryer.

To keep the analysis simple, the energy balance equations at the absorber plate, working fluid (air), crop and chamber are written under the following assumptions: (i) capacity effects of the absorber plate and enclosed air and working fluid (air) have been neglected; (ii) solar radiation is calculated for a 45° tilted surface; (iii) the thin layer drying concept is adopted in which it is assumed that the crop becomes heated uniformly throughout its depth.

The energy balance equations are then given by:

At the absorber plate

\[ \tau x_p \rho \dot{I}(t) = h_{ps}(T_p - T_i) + h_{rps}(T_p - T_c) \]
\[ + U(t)(T_p - T_a). \]  

(1)

At the working fluid (air)

\[ h_{pf}(T_p - T_i) = h_{f0}(T_t - T_a). \]

(2)

At the crop surface

\[ [h_{c0}(T_c - T_c) + h_{rps}(T_p - T_c)] A_p = M_e C_e \frac{dT_c}{dr} \]
\[ + h_{w}(T_c - T_{ch}) A_p + h_{w0}(T_c - T_{ch}) A_p. \]

(3)

At the chamber (Daugherty et al., 1989)

\[ [h_{w0} + h_{w}](T_c - T_{ch}) A_p = C_d A_v \sqrt{2g \Delta H \Delta P} \]
\[ + h_{s} A_s(T_{ch} - T_p) \]

(4)

where

\[ \Delta P = [P(T_{ch}) - \gamma P(T_a)] \]
\[ \Delta H = \frac{\Delta P}{\rho_a g}. \]

(5)

The vapour pressure at temperatures \( T_c, T_{ch} \)
and \( T_a \) has been determined by using a linear regression technique, as discussed in
Appendix A, for the temperature range 15-50°C

$$P(T) = R_1 T + R_2.$$  \hfill (6)$$

With the help of eqns. (1) and (2), eqn (3) is further simplified and written as

$$M_C C_v \frac{dT}{dt} = (a_{sp} \tau \rho l(t)(U_1 + U_2) - (U_p + U_n)$$

$$(T_c - T_a) A_p - (h_{ac} + h_{sc})(T_c - T_{ca}) A_p.$$ \hfill (7)$$
Table 1. Design and climatic parameters used in the study

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_p$, $A_s$</td>
<td>0.75</td>
<td>$s$</td>
<td>0.9</td>
</tr>
<tr>
<td>$C_d$</td>
<td>0.6</td>
<td>$r$</td>
<td>0.9</td>
</tr>
<tr>
<td>$L$</td>
<td>1.5</td>
<td>$\rho$</td>
<td>0.9</td>
</tr>
<tr>
<td>$W$</td>
<td>0.5</td>
<td>$d$</td>
<td>0.03</td>
</tr>
<tr>
<td>$x$</td>
<td>0.1–0.2</td>
<td>$C_s$</td>
<td>4190</td>
</tr>
<tr>
<td>$g$</td>
<td>9.8</td>
<td>$M_e$</td>
<td>19–40</td>
</tr>
<tr>
<td>$R_s$</td>
<td>293.3</td>
<td>$R_s$</td>
<td>-3903.0</td>
</tr>
<tr>
<td>$h_e$</td>
<td>2.4</td>
<td>$h_{ac}$</td>
<td>3.0</td>
</tr>
</tbody>
</table>

Equations (4) and (7) cannot be solved analytically as they are non-linear, therefore eqn (4) is simplified with the help of eqns (5) and (6) and is converted into a quadratic equation as shown by eqn (8) to determine $T_{ch}$ by assuming an initial value of $T_c$

$$A T_{ch}^2 - 2B T_{ch} + C = 0. \quad (8)$$

Further, this value of $T_{ch}$ has been utilized to determine the crop temperature ($T_c$) using the finite difference technique (Fig. 2):

$$T_{ch} = T_{ch-1} + \frac{A_p}{M_e C_c} [x_p \rho (U_1 + U_2) - (U_{pa} + U_{pa}) (T_{ch-1} - T_c) - (h_{ac} + h_{ac}) (T_{ch-1} - T_{ch})]. \quad (9)$$

The overall heat transfer coefficients and constants in the quadratic equation are discussed in Appendix A.

To predict the performance of the system, the thermal loss efficiency has been determined using the expression:

$$\eta_t = \frac{h_{ac} (T_c - T_{ch})}{I} \times 100. \quad \text{(10)}$$

The overall efficiency of the system ($\eta_o$) can be determined using the expression:

$$\eta_o = \frac{Q_o \times 100}{\int I(t) \, dt}. \quad \text{(11)}$$

4. NUMERICAL RESULTS AND DISCUSSION

The different design and climatic parameters used in the study are reported in Table 1 and Fig. 3, respectively.

The hourly variation of various temperatures, namely crop, plate, fluid and chamber, are shown in Fig. 4. The effects of vent area and crop capacity on hourly crop temperature for normal and reverse absorber cabinet dryers are shown in Figs 5 and 6, respectively. Further, the effects of vent area on maximum crop temperature at a mass of 10 kg is shown in Fig. 7. Figs 8 and 9 show the variation of instantaneous thermal loss efficiency and overall drying efficiency, respectively, for both normal and reverse absorber cabinet dryers.

On the basis of numerical computation and Figs 4–9, the following conclusions can be drawn:

![Fig. 3. Hourly variation of solar intensity at an inclination of 45° and ambient temperature for a typical June day for Delhi.](image-url)
(1) The crop and chamber temperatures are less than the plate and fluid temperatures during sunshine hours and the crop temperature remains a little higher than the other temperatures during periods of no sun. This is because of the crop capacity which gives a thermal storage effect (Fig. 4).

(2) The crop temperature in the reverse absorber cabinet dryers is more than for normal cabinet dryers because of the maximum utilization of the available solar energy and minimum heat loss (Figs 5 and 6).

(3) There is significant variation in crop temperature with heat capacity of the crop, as temperature decreases with an increase of crop capacity. Further, it is higher in the case of RACD and remains almost the same in the late hours because of the storage effect (Fig. 6). The variation of the crop temperature with vent area is higher in normal cabinet dryers [Fig. 5(b)] and seems to be
insignificant in RACD [Fig. 5(a)] as its optimum vent width is about 0.14 m [Fig. 7(a)].

(4) The RACD can be operated for longer than normal cabinet dryers because of minimum bottom and top heat losses [Fig. 6(a) and (b)].

(5) The instantaneous thermal loss efficiency factor ($\eta_t$) for RACD is more than for a normal cabinet dryer (Fig. 8). This causes the fast removal of moist air which, in turn, helps in quick drying and therefore the crop temperature in RACD is higher than for normal cabinet dryers.

(6) The effect of mass of crop on the overall system efficiency is significant in both cases; it decreases with increase of crop capacity. Therefore, these systems can be used for small crop capacity (Fig. 9).

**NOMENCLATURE**

- $A_p$ area of product, m$^2$
- $A_e$ area of side walls of drying chamber, m$^2$
- $L_e$ area of seat, m$^2$ = $L_e$
- $C_s$ specific heat of crop, J kg$^{-1}$ °C$^{-1}$
- $C_d$ coefficient of diffusion
- $d$ depth of air duct, m
- $g$ acceleration due to gravity, m s$^{-2}$
- $h_{ec}$ convective heat transfer coefficient from crop to chamber, W m$^{-2}$ °C$^{-1}$
- $h_{ecw}$ evaporative heat transfer coefficient from crop to chamber, W m$^{-2}$ °C$^{-1}$
- $h_{fc}$ convective heat transfer coefficient from fluid to crop, W m$^{-2}$ °C$^{-1}$
- $h_{fa}$ convective heat transfer coefficient from absorber plate to fluid, W m$^{-2}$ °C$^{-1}$
- $h_{rad}$ radiative heat transfer coefficient from plate to crop, W m$^{-2}$ °C$^{-1}$
- $h_{ch}$ convective heat transfer coefficient from chamber to ambient through side walls, W m$^{-2}$ °C$^{-1}$
- $I(t)$ solar intensity at time $t$, W m$^{-2}$
- $L$ length of the absorber plate/cabinet dryer, m
- $M_c$ mass of crop, kg
Fig. 7. Maximum crop temperature as a function of vent width: (a) RACD; (b) normal cabinet dryer.

Fig. 8. Instantaneous thermal efficiency as a function of \((T_{ch} - T_a)/I(t)\). Left axis: RACD. Right axis: normal cabinet dryer.
Fig. 9. Effect of mass of crop on overall efficiency.

\[ \begin{align*}
Q_s &= \text{heat energy of evaporated moisture, } \text{W m}^{-2} \\
T &= \text{temperature, } ^\circ\text{C} \\
P(T) &= \text{partial vapour pressure at temperature } T, \text{Pa} \\
U_i &= \text{overall heat transfer coefficient from absorber plate to ambient through glass cover, } \text{W m}^{-2} ^\circ\text{C}^{-1} \\
W &= \text{width of plate/dryer, m} \\
x &= \text{width of vent, m}
\end{align*} \]

**Greek letters**

\[ \begin{align*}
\alpha &= \text{absorptivity of absorber} \\
\gamma &= \text{relative humidity inside drying chamber, decimal} \\
\rho &= \text{reflectivity of reflector} \\
\rho_a &= \text{density of air, } \text{kg m}^{-3} \\
\tau &= \text{transmissivity of glass cover} \\
\eta &= \text{thermal loss efficiency} \\
\eta_o &= \text{overall drying efficiency}
\end{align*} \]

**Subscripts**

\[ \begin{align*}
a &= \text{ambient} \\
c &= \text{crop} \\
ch &= \text{chamber} \\
f &= \text{fluid} \\
p &= \text{absorber plate}
\end{align*} \]

**REFERENCES**


**APPENDIX A**

The coefficients used in eqn (8) are now described:

\[ A = \left( A_{p} h_{w} \right)^{2} + (h_{c}, A_{c})^{2} + 2 A_{p} h_{w} h_{c} A_{c} \]

\[ B = T_{c} \left( A_{c} h_{w} \right)^{2} + T_{c} (h_{c}, A_{c})^{2} + A_{p} h_{w} h_{c} A_{c} + A_{p} h_{w} h_{c} A_{c} T_{c} \]

\[ + (C_{c} A_{c})^{2} g R_{1} \]

\[ C = 2 g (C_{c} A_{c}) B_{g} \left( R_{1}^{2} T_{c} - R_{2}^{2} (1 - g) \right) + (A_{p} h_{w} h_{c} A_{c})^{2} \]

\[ + A_{p} h_{w} h_{c} A_{c} T_{c} B_{g} + (h_{c}, A_{c})^{2} \]

The different heat transfer coefficients used in the basic energy balance equations (eqns (1) - (4)) are:

\[ h_{sc} = 0.016 h_{sc} \left( \frac{P(T) - \gamma P(T_{w})}{T_{c} - T_{w}} \right) \]

\[ h_{sc} = h_{sc} + h_{sc} \]

\[ U_{sc} = \frac{U_{sc}}{U_{sc} + h_{sc} + U_{t}} \]

\[ U_{sc} = \frac{U_{sc}}{U_{sc} + h_{sc} + U_{t}} \]

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\[ U_{sc} = \frac{h_{sc}}{U_{sc} + h_{sc} + U_{t}} \]

The values of \( R_{1} \) and \( R_{2} \) mentioned in Table I and used in eqn (6) were determined using the linear regression technique. Vapour pressure data for the temperature range 15-50°C were taken from a steam table (Tiwari et al., 1985).