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# Modeling the performance of the reversed absorber with packed bed thermal storage natural convection solar crop dryer

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**Abstract**

A transient analytical model has been presented to study the new concept of a solar crop dryer having reversed absorber plate type collector and thermal storage with natural airflow. The performance of $1 \times 1\ m^2$ area of crop dryer with packed bed and airflow channel was evaluated for drying of onions. The model was solved to compute the air temperatures and various functional components of the drying systems for a day of the month of October for the climatic condition of Delhi (India). The parametric study involved the effect of width of airflow channel and height of packed bed on the crop temperature. The effect of thermal storage is observed on the natural mass flow rate in the drying system. A 30° inclined absorber plate with in-built thermal storage and 0.12 m width of airflow channel induced the mass flow rate in the range of 0.032–0.046 kg $s^{-1}$ during the drying process. The thin layer drying equation was used to study the drying rate and hourly reduction in moisture content in the crop trays. It has been observed that the crop moisture content and drying rate decreases with the drying time of the day. A reversed absorber plate of 1 m length and 1 m breadth with 0.15 m packed bed could dry 95 kg of onion from a moisture content of 6.14–0.27 kg water/kg of dry matter in a 24 h drying period.


*Keywords:* Natural convection; Solar crop drying; Reversed flat plate absorber; Thermal storage; Thermal modeling

## 1. Introduction

Agricultural products have been dried by the sun and wind in the open air for thousands of years. The disadvantage associated with open sun drying are over-drying, insufficient drying, discoloration by the UV radiation and contamination by the dust, foreign materials, insects and microorganisms (Esper & Mühlbauer, 1998). Various cabinet dryers have been developed to overcome the problems of open sun drying. However, sun drying is still common practice in many tropical and subtropical countries (Jain & Tiwari, 2003; Szulmayer, 1971). Low temperature is normally recommended for drying most of the agricultural products. The temperature rise of 15–20 °C of air from ambient is sufficient to meet to the requirement of crop drying (Dubey & Pryor, 1996). Where, the utilization of solar energy is most convenient. Therefore, application of solar energy in agricultural products drying is one of the most important potential and research area for every developing country.

Solar drying refers to the methods of use of sun's energy for drying but excludes open-air sun drying. The justification for solar driers is that they may be more effective than sun drying, but have lower operating costs than mechanized driers.

The durations of solar drying is limited to sunshine hours. For continuous drying, a thermal storage could be provided with the solar air heater (Aboul-Enein, El-Sebaii, Ramadan, & El-Gohary, 2000). A thermal storage unit integrated with the solar air heater may be charged during the peak sunshine hours and utilized (discharged) during off sunshine hours for supplying the hot air to the dryer (Jain, 2005a, 2005b; Jain & Jain, 2004).

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**Nomenclature**

| | |
|---|---|
| $A$ | area, $m^2$ |
| $A_o$ | cross sectional area of outlet air flow channel, $m^2$ |
| $A_s$ | surface area of side wall of dryer, $m^2$ |
| $a_r$ | ratio of cross sectional area of outlet and inlet of air flow channel |
| $a_w$ | water activity, decimal |
| $b$ | breadth of dryer and absorber plate-I, m |
| $C$ | specific heat at constant pressure, $J\,kg^{-1}\,K^{-1}$ |
| $C'$ | coefficient of Eq. (17) |
| $C_d$ | coefficient of diffusivity |
| $c$ | drying constant |
| $D_b$ | diameter of pebbles, m |
| $d$ | duct width of airflow channel, m |
| $G_0$ | mass velocity, $kg\,m^{-3}\,s^{-1}$ |
| $g$ | gravitational acceleration, $m\,s^{-2}$ |
| $H$ | height, m |
| $H_u$ | absolute humidity, kg vapor/kg of dry air |
| $h_c$ | convective heat transfer coefficient, $W\,m^{-2}\,K^{-1}$ |
| $h_r$ | radiative heat transfer coefficient, $W\,m^{-2}\,K^{-1}$ |
| $h_v$ | volumetric heat transfer coefficient, $J\,m^{-3}\,s^{-1}\,K^{-1}$ |
| $h_o$ | convective heat transfer coefficient due to wind, $W\,m^{-2}\,K^{-1}$ |
| $I_{eff}$ | hourly average effective solar radiation on absorber plate-I, $W\,m^{-2}$ |
| $I_t$ | hourly average solar radiation on inclined surface, $W\,m^{-2}$ |
| $K$ | thermal conductivity, $W\,m^{-1}\,K^{-1}$ |
| $k$ | coefficient of Eq. (17) |
| $k_d$ | drying constant, $s^{-1}$ |
| $L$ | length of dryer and absorber plate-I, m |
| $l$ | thickness, m |
| $M$ | moisture content of grain, kg water/kg of dry matter |
| $M_c$ | mass of the crop tray, kg |
| $M_e$ | equivalent moisture content, kg water/kg of dry matter |
| $M_0$ | initial moisture content of crop, kg water/kg of dry matter |
| $M_s$ | mass of storage material, kg |
| $\dot{m}_a$ | mass flow rate, $kg\,s^{-1}$ |
| $Nu$ | Nusselt number |
| $R$ | universal gas constant, $J\,K^{-1}\,mol^{-1}$ |
| $Ra$ | Rayleigh number |
| $T$ | temperature, K |
| $\Delta T$ | temperature difference, K |
| $t$ | time, s |
| $U$ | overall heat loss coefficient from sides of dryer, $W\,m^{-2}\,K^{-1}$ |
| $U_{p1}$ | overall heat loss coefficient from bottom of absorber plate-I, $W\,m^{-2}\,K^{-1}$ |
| $v$ | wind velocity, $m\,s^{-1}$ |
| $v_f$ | velocity of air, $m\,s^{-1}$ |
| $x$ | length of coordinate in direction of flow, m |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | absorptivity |
| $\alpha_f$ | diffusivity of air, $m^2\,s^{-1}$ |
| $\beta'$ | expansion factor, $K^{-1}$ |
| $\beta$ | tilt angle of absorber plate, degree |
| $\varepsilon$ | emissivity |
| $\varepsilon_b$ | porosity of packed bed |
| $\nu_f$ | kinematic viscosity of air, $m^2\,s^{-1}$ |
| $\rho'$ | reflectivity of reflector |
| $\rho$ | density, $kg\,m^{-3}$ |
| $\sigma$ | Stefan–Boltzmann constant, $W\,m^{-2}\,K^{-4}$ |
| $\tau$ | transmitivity |

*Subscripts*

| | |
|---|---|
| a | ambient air |
| b | packed bed |
| c | crop |
| c1, c2 | crop in trays-I and II |
| c1c2 | crop tray-I to crop tray-II |
| c2ch | crop tray-II to chamber |
| ch | chamber |
| f | air in packed bed |
| f1 | air stream-I |
| f2 | air stream-II |
| f1b | air stream-I to packed bed |
| g | glass cover |
| ga | glass cover to ambient air |
| gf2 | glass cover to air stream-II |
| gsky | glass cover to sky |
| $i$ | number of side wall of dryer or number of reflector, integers, 1–4 |
| p1 | absorber plate-I |
| p1b | absorber plate-I to packed bed |
| p1f1 | absorber plate-I to air stream-I |
| p2 | absorber plate-II |
| p2f2 | absorber plate-II to air stream-II |
| p2g | absorber plate-II to glass cover |
| p2s | absorber plate-II to storage material |
| r | reflector |
| s | storage material |
| sa | storage material to air |
| sch | storage material to air in chamber |
| sky | sky |
| w | dryer wall |

Several designs of solar air collector and crop dryer have been studied over the years (Ekechukwu & Norton, 1999a, 1999b). Flat plate solar collectors are mostly used. Air may be allowed to flow above, below or on both sides of the absorber plate. Airflow under the absorber plate reduces heat losses through the glazing. Major losses from the collector occur at the front cover because the front face must be exposed to the ambient. A number of designs have been proven technically (Ekechukwu & Norton, 1999b; Pangavhane & Sawhney, 2002), but while none is yet in widespread use, there is still optimism about their potential (Khiari, Mihoubi, Mabrouk, & Sassi, 2004). The efficiency of the solar collector depends on its type and model as well as on the rate of heat loss during operation (Timoumi, Mihoubi, & Zagrouba, 2004).

Among above all, few natural convection solar dryers have been studied. In some recent studies, the natural convection dryers have been developed for drying the fruits (Bala, Mondol, Biswas, Chowdury, & Janjai, 2003; El-Sebaii, Aboul-Enein, Ramadan, & El-Gohary, 2002; Pangavhane, Sawhney, & Sarsavadia, 2002) and could be very well used during sunshine hours. There is a need for the design of natural convection solar dryer, which could be used for throughout the day and night. Mathematical modeling and computer simulation are important tools for determining the energy efficient design as well as predicting the overall performance of drying system (Dubey & Pryor, 1996; Jain, 2005b). Therefore, the present simulation is carried out with a new concept of solar air heating by reversed plate absorber, where the major thermal losses from the collector are avoided. The packed bed thermal storage is proposed in-built with dryer for continuous use of the dryer. An inclined flat plate absorber is provided for natural flow of air in the system and another thermal storage is in-built with it to maintain thermal buoyancy for the continuity of airflow during off sunshine hours.

## 2. Description and working principle of drying system

A schematic diagram of a proposed reversed flat plate absorber type solar crop dryer is shown in Fig. 1. Various heat transfer coefficients are also shown in Fig. 1, which explains the working principle of solar crop dryer. The system is assumed to face towards the midday sun. The absorber plate-I is horizontal and downward facing. A fraction of polygonal shape reflector (segment of five equal flat pieces) is placed under the flat plate absorber-I to introduce solar radiation from below. The radius (of polygon) of opening of reflector is same as that of the absorber plate. A drying setup is placed above the absorber plate-I at the gap of 0.04 m for entering ambient air. A packed bed (pebble) is provided before the crop tray for storage of thermal energy. Two drying trays are proposed for keeping the crop. An inclined flat absorber plate-II with single glazing at 30° tilt (Jain & Jain, 2004) is provided above the crop tray to induce the draft in the crop dryer. A triangular cavity below the inclined absorber plate-II is filled with the storage material of granite grits (Aboul-Enein et al., 2000) to store the thermal energy during sunshine hours and supply (release) during off sunshine hours.

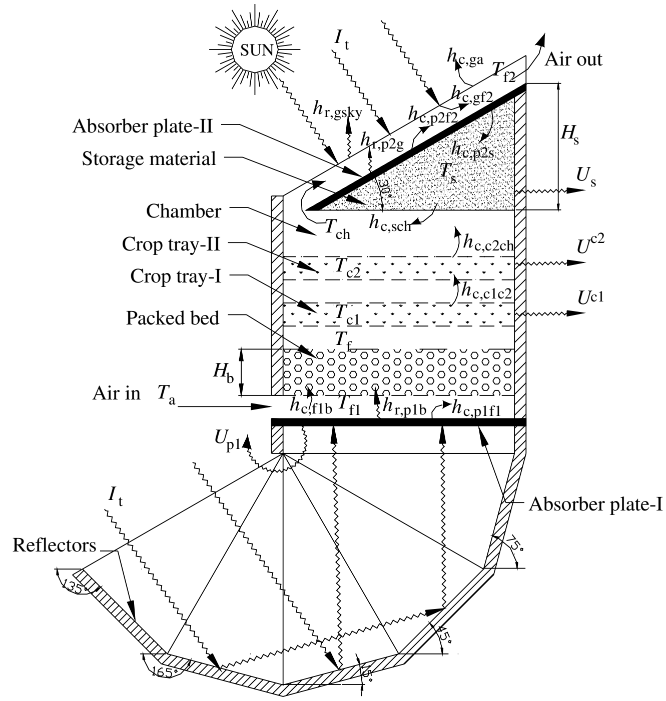


Fig. 1. Schematic view of reversed absorber with thermal storage natural convective solar crop dryer.

An inclined absorber plate-II absorbs the solar radiation and is being heated up. The part of this heat is transferred by convection and radiation to the air flowing stream-II, creates the draft due to thermal buoyancy and air starts flowing in the system. The rest of energy is transferred to storage material and it is being utilized during off sunshine hours. Thus, a nominal airflow is maintained throughout the drying process. On the other side, at the absorber plate-I receives the reflected solar radiation and is being heated. The heat is transferred by convection and radiation to air in stream-I, which is coming into the system at ambient condition. Thus, the hot air starts flowing into the packed bed. The hot air heats up the pebbles of packed bed, resulting in a drop in the air temperature, which flows towards the crop already placed in trays for drying. During sunshine hours, the packed bed is being charged by the hot air comes from absorber plate-I and supplies air at moderate temperature for crop drying. Outside of these sunshine hours, the ambient air at the lower temperature is being heated in the packed bed and passed through the crop dryer. Thus, the packed bed charged during sunshine hours due to the high temperature of air stream-I is discharged (release heat) during off-sunshine hours.

## 3. Thermal analysis

The thermal model was developed based on the energy balance equation on the various components of the solar crop dryer. The energy balance equations for the various functional components of the reversed absorber, packed bed, inclined absorber and dryer as shown in Fig. 1 are written on the following assumption:

(i) absorptivity of air is negligible,
(ii) heat capacities of the air, glass cover, absorber plate and insulation are negligible,
(iii) there is no temperature gradient along the thickness of glass cover,
(iv) storage material has an average temperature ($T_s$) at a time ($t$),
(v) there is no stratification exists perpendicular to the air flow in channel,
(vi) the system is perfectly insulated and there is no air leakage,
(vii) heat flow is one dimensional and in a quasi-steady state condition,
(viii) dryer is facing the midday sun.

### 3.1. Energy balance equation on solar energy collector components

(a) Reversed absorber plate-I

$$\tau\alpha_{p1}I_{eff}A_{p1} = h_{c,p1f1}(T_{p1} - T_{f1})A_{p1} + h_{r,p1b}(T_{p1} - T_b)A_{p1} + U_{p1}(T_{p1} - T_a)A_{p1} \tag{1}$$

where; $I_{eff} = \sum(\rho_i' I_i A_{ri})/A_{p1}$, effective solar radiation per unit area on the collector plate-I.

(b) Air stream of absorber plate-I (air stream-I)

$$h_{c,p1f1}(T_{p1} - T_{f1})A_{p1} = h_{c,f1b}(T_{f1} - T_b)A_{p1} \tag{2}$$

### 3.2. Energy balance equation on packed bed (Schumann, 1929)

(a) Pebbles in packed bed

$$A_b\rho_b C_b(1 - \varepsilon_b)\frac{\partial T_b}{\partial t} = h_v(T_f - T_b)A_b + h_{r,p1b}(T_{p1} - T_b)A_{p1} \tag{3}$$

(b) Air in packed bed

$$A_b\rho_f C_f\varepsilon_b\frac{\partial T_f}{\partial t} = -\dot{m}_a C_f\frac{\partial T_f}{\partial(x_b/H_b)} + h_v(T_b - T_f)A_b \tag{4}$$

For an air based packed bed system, the first term of Eq. (4) can be neglected and equation can be written as

$$\dot{m}_a C_f\frac{\partial T_f}{\partial(x/H_b)} = h_v(T_b - T_f)A_b \tag{5}$$

### 3.3. Energy balance equation on drying trays

(a) Crop tray-I

$$\dot{m}_a C_f(T_{f2} - T_{c1}) = M_{c1}C_c\frac{dT_{c1}}{dt} + h_{c,c1c2}(T_{c1} - T_{c2})A_c + \sum U_{c1,i}A_{s,c1,i}(T_{c1} - T_a) \tag{6}$$

(b) Crop tray-II

$$h_{c,c1c2}(T_{c1} - T_{c2})A_c = M_{c2}C_c\frac{dT_{c2}}{dt} + h_{c,c2ch}(T_{c2} - T_{ch})A_c + \sum U_{c2,i}A_{s,c2,i}(T_{c2} - T_a) \tag{7}$$

(c) Chamber (Bansal & Mathur, 1993)

$$h_{c,c2ch}(T_{c2} - T_{ch})A_c + h_{c,sch}(T_s - T_{ch})A_{sch} = \dot{m}_a C_f(T_{f2} - T_{ch}) \tag{8}$$

Bansal and Mathur (1993) and Anderson (1995) had given the air mass flow in the chimney with natural convection as

$$\dot{m}_a = \frac{C_d\rho_f A_o}{\sqrt{1 + a_r}}\sqrt{\frac{2gH_s(T_{f2} - T_{ch})}{T_{ch}}} \tag{9}$$

#### 3.3.1. Energy balance on the components of natural convection collector and storage

(a) Glass cover

$$\alpha_g I_t A_g + h_{r,p2g}(T_{p2} - T_g)A_g = h_{c,gf2}(T_g - T_{f2})A_g + h_{c,ga}(T_g - T_a)A_g + h_{r,gsky}(T_g - T_{sky})A_g \tag{10}$$

where; $T_{sky} = T_a - 6$ (Whiller, 1967).

(b) Inclined absorber plate-II

$$\tau_g\alpha_{p2}I_t A_{p2} = h_{r,p2g}(T_{p2} - T_g)A_{p2} + h_{c,p2f2}(T_{p2} - T_{f2})A_{p2} + h_{c,p2s}(T_{p2} - T_s)A_{p2} \tag{11}$$

(c) Air stream in channel (air stream-II)

$$h_{c,gf2}(T_g - T_{f2})b\,dx + h_{c,p2f2}(T_{p2} - T_{f2})b\,dx = \dot{m}_a C_a\frac{dT_{f2}}{dx}dx \tag{12}$$

(d) Storage material below the absorber plate-II

$$h_{c,p2s}(T_{p2} - T_s)A_{p2} = M_s C_s\frac{dT_s}{dt} + h_{c,sch}(T_s - T_{ch})A_{sch} + \sum U_{s,i}A_{s,sa,i}(T_s - T_a) \tag{13}$$

#### 3.3.2. Modeling of thin layer drying

The theory of drying is described by Lewis (1921) based on the analogous of Newton's law of cooling in heat transfer and is often used to mass transfer in thin layer drying and is as follows:

$$\frac{dM}{dt} = -k_d(M - M_e) \tag{14}$$

On integration of Eq. (14) yields

$$\frac{M - M_e}{M_0 - M_e} = c\exp(-k_d t) \tag{15}$$

where the drying coefficient $k_d$ for onion slices is evaluated as the function of air velocity, absolute humidity and crop temperature by Sarsavadia, Sawhney, Pangavhane, and Singh (1999).

$$k_d = 47.57 \times v_f^{0.31} H_u^{-0.2} \exp\left[\frac{-3034}{T}\right] \tag{16}$$

and $c = 1.01$.

The expression of equilibrium moisture for onion slices with the help of well-known GAB (Guggenheim–Anderson–de Boer) equation by Kiranoudis, Maroulis, Tsami, and Marinos-Kouris (1993) as

$$M_e = \frac{0.202 \times C'ka_w}{(1 - ka_w)[1 - (1 - C')ka_w]} \tag{17}$$

where

$$C' = 2.30 \times 10^5 \exp\left[\frac{32.5}{RT}\right] \tag{18}$$

$$k = 5.79 \times 10^2 \exp\left[\frac{6.43}{RT}\right] \tag{19}$$

## 4. Input parameters

The mathematical model is solved for Delhi (latitude 28°35′ N, longitude 77°17′ E and altitude 216 m from mean sea level) climatic conditions during October (day of year, $n = 288$), since this is a most suitable time for drying most of the crops in India. The onion is taken for present simulation study. Hourly average solar intensity and ambient air temperature used in solving the model are given in Fig. 2. On the abscissa of Fig. 2, hourly average time is given, which started from the average of 6 and 7 h as 1 h (corresponding to 0 of $y$-axis). Similarly, every hour on $x$-axis represents an average value of time. Therefore, the maximum solar radiation lies between 12 to 13 hours is represented at 7 h from the starting point. The maximum ambient temperature lags by 2 h of maximum solar intensity and lies between 15–16 hours (10 h on $x$-axis). Similarly, from Figs. 3–8 on $x$-axis data are the response of the average of two consecutive hourly data.

Solar intensities on the different inclined surfaces of reflectors and 30° tilted absorber were computed by using the method given by Lui and Jordan (1962). The effective solar intensity ($I_{eff}$) available for the reversed absorber plate-I and solar intensity on inclined ($\beta = 30°$) absorber plate-II are also shown in Fig. 2 and are equal throughout the day. The various input parameters are given in Table 1. The various design parameters are given in Table 2. The Matlab-6.1 software has been used to solve the mathematical model.

## 5. Results and discussion

### 5.1. Performance of solar crop drying system

A computer program was prepared to solve the energy balance Eqs. (1)–(13) on different components of solar drying system and to find out the temperatures of the air in stream-I ($T_{f1}$), stream-II ($T_{f2}$) packed bed ($T_f$), and drying chamber ($T_{ch}$) and temperature of absorber plate-I ($T_{p1}$), absorber plate-II ($T_{p2}$), storage material ($T_s$), crop in tray-I ($T_{c1}$) and tray-II ($T_{c2}$). The results are obtained for the solar intensity and ambient air temperature of the

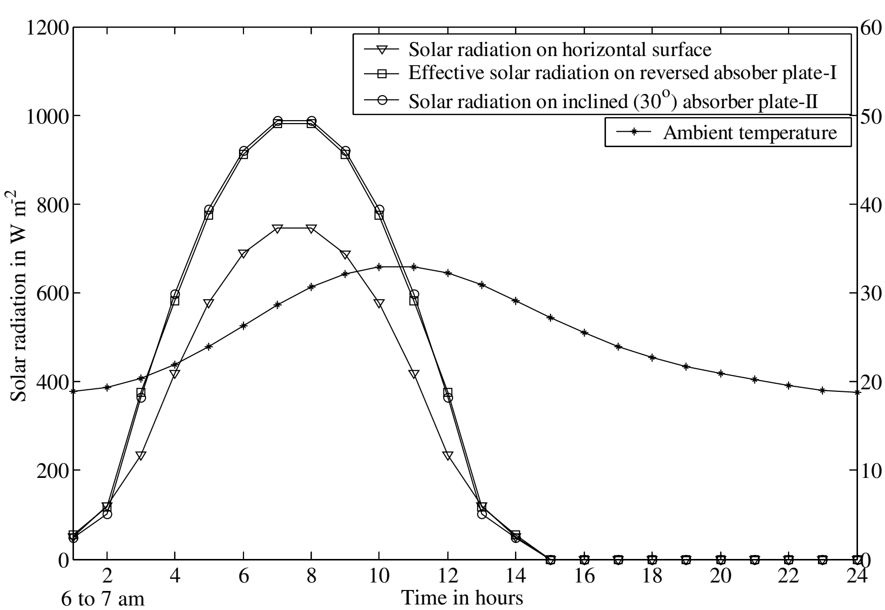


Fig. 2. Diurnal variation of average hourly ambient temperature and solar intensity on different surfaces during the month of October.

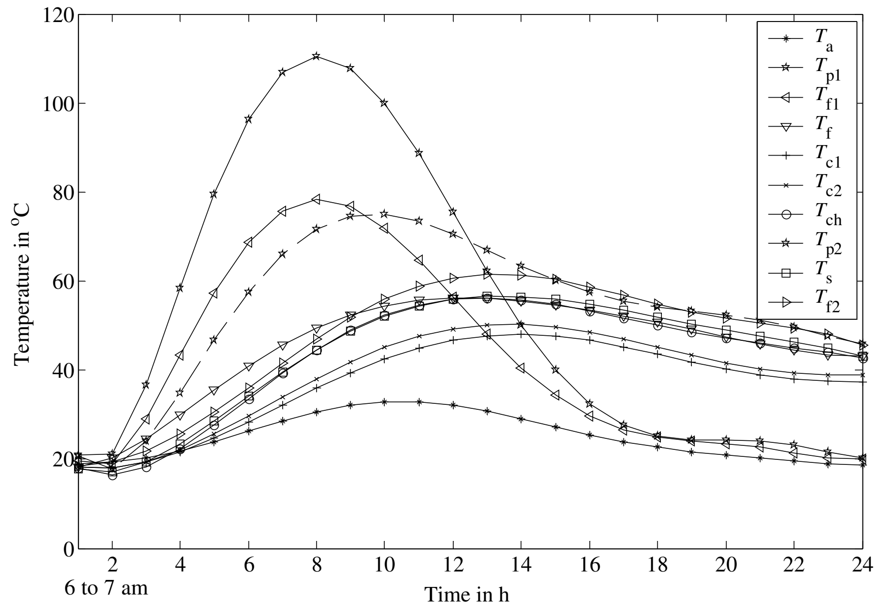


Fig. 3. Hourly variation in temperatures of different component of solar drying systems for $d = 0.12$ m and $H_b = 0.15$ m.

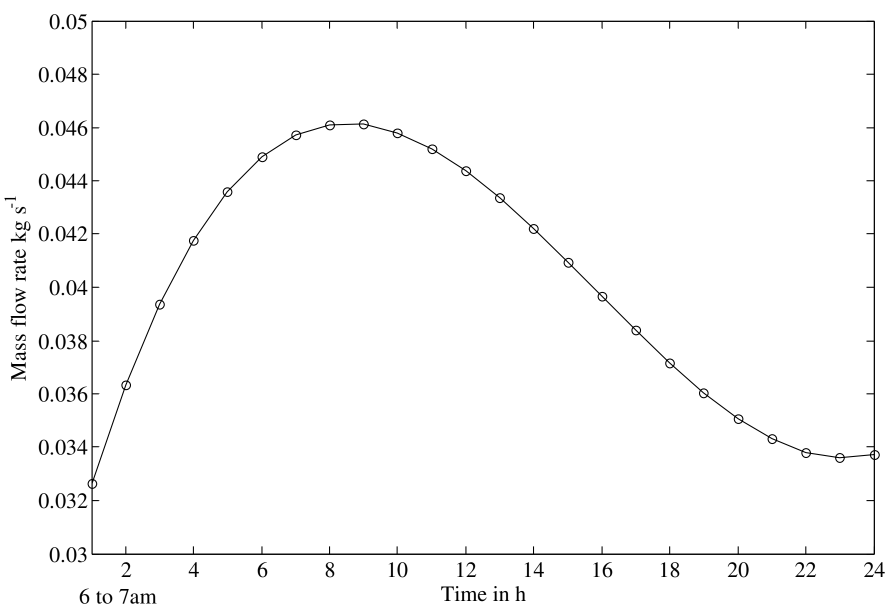


Fig. 4. Hourly variation in mass flow rate in solar crop drying system.

month of October for the climatic condition of Delhi. The design parameters of the drying system are; $L = 1$ m, $b = 1$ m, $d = 0.12$ m and $H_b = 0.15$ m.

Fig. 3 shows the hourly temperatures of the air in different streams, absorber plates, packed bed, storage material and crop. The ambient air temperature is also shown in Fig. 3 to observe in the difference of temperature from each component. The working principle of the solar crop drying system can be explained from the results obtained in Fig. 3.

During sunshine hours, the absorber plates being heated by the absorption of solar radiation. Though, the effective solar intensities on the reversed absorber plate-I and solar intensities on the inclined absorber plate-II are equal (Fig. 2), but the temperature of reversed absorber plate-I ($T_{p1}$) is higher than the absorber plate-II ($T_{p2}$) during day hours, since there is very little convective and radiative from absorber plate-I. However, $T_{p1}$ gets reducing rapidly after sunshine hours and becomes less than $T_{p2}$. On the other side, a slow reduction in $T_{p2}$ is observed. It is due to the fact that absorber plate-II is attached with the storage material, which release heat during off sunshine hours. The temperature of air in stream-I ($T_{f1}$) and stream-II ($T_{f2}$) follows the similar pattern of $T_{p1}$and $T_{p2}$, respectively; since they are directly depend on their respective absorber plates of airflow channels.

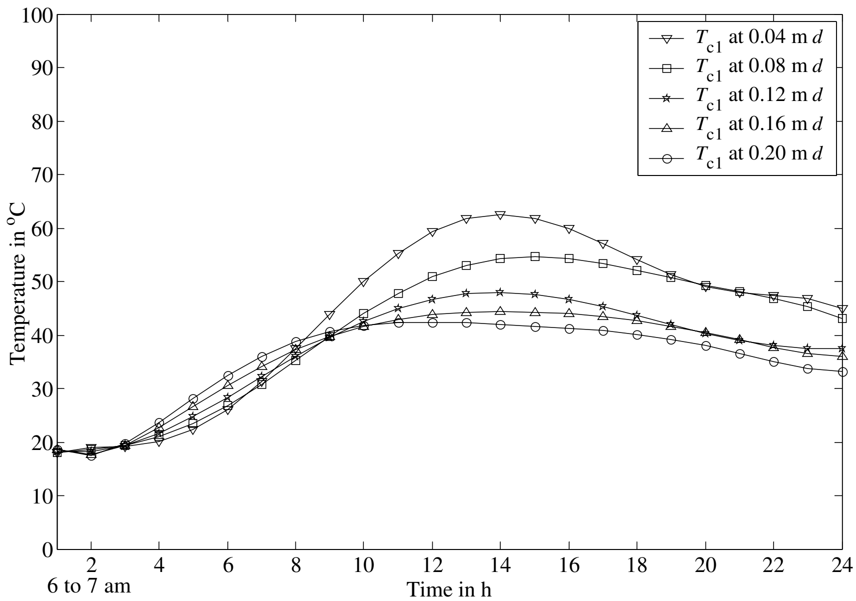


Fig. 5. Effect of width of airflow channel on crop temperature in tray-I in solar crop drying system.

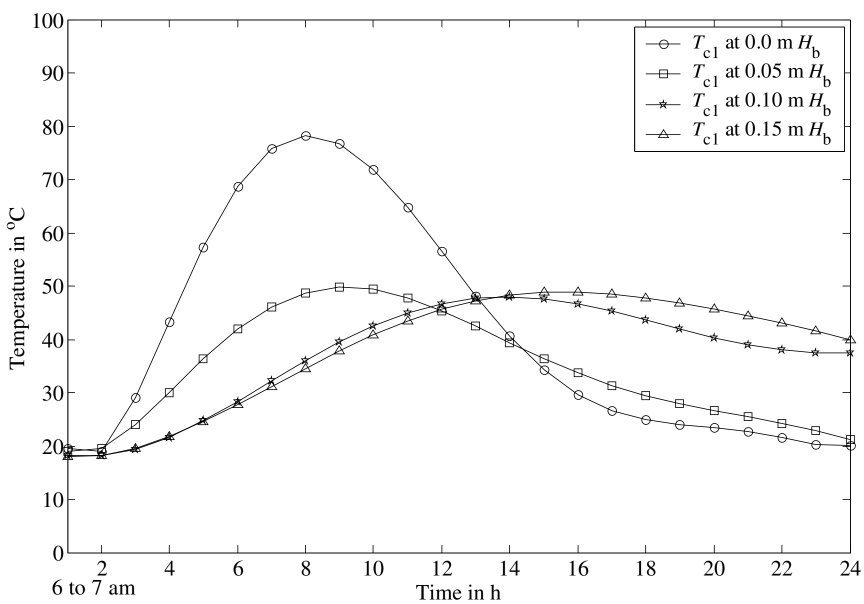


Fig. 6. Effect of height of packed bed on crop temperature in tray-I in solar crop drying system.

During sunshine hours, the ambient air at temperature $T_a$ entering into the stream-I is being heated due to the convective and radiative heat transfers from absorber plate-I and attain the temperature $T_{f1}$. Air of stream-I, then entered into the packed bed and reduced to $T_f$ and utilize for crop drying. During off sunshine hours, air of stream-I ($T_{f1}$), which is close to ambient temperature ($T_a$), while passing through the packed bed is being heated and increased to $T_f$. Thus, the air temperature ($T_f$) is supplied to crop dryer is at optimum and less fluctuating.

The storage material (granite grits) below the inclined absorber plate-II is being heated throughout the sunshine hours, which results in reduction of $T_{f2}$, $T_{p2}$, and $T_{ch}$ and provide the stable temperatures (~50–40 °C) out of sunshine hours. The crop temperature in tray-II ($T_{c2}$) is slightly higher than crop in tray-I ($T_{c1}$). It is due to fact that crop in tray-II received heat from two sides i.e. the residual heat of tray-I and from chamber ($T_{ch}$), which is being receiving heat from thermal storage.

Effect of thermal buoyancy on mass flow rate is presented in Fig. 4. The mass flow rate varies from 0.0322 to 0.0463 kg s$^{-1}$. The mass flow rate is the resultant of temperature difference between the air in stream-II ($T_{f2}$) and drying chamber ($T_{ch}$) and evaluated from the Eq. (9). The mass flow rate increase in the first 8 h of process till 13 h of time, and gets reducing after 14 h. The minimum

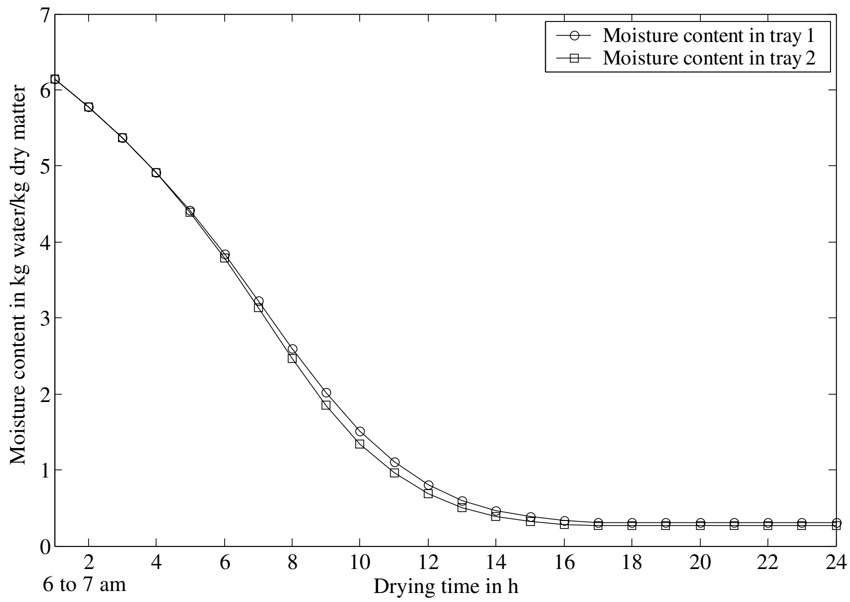


Fig. 7. Hourly variation of moisture content with drying time.

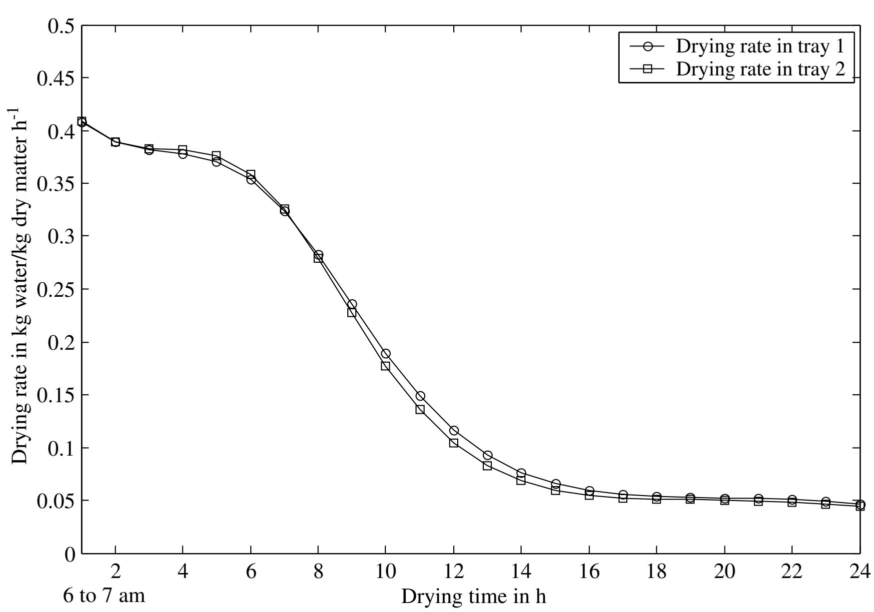


Fig. 8. Hourly variation in drying rate with drying time.

mass flow rate is assured after the sunshine hour due to effect of thermal storage.

### 5.2. Effect of width of air flow channel on crop temperature

Variation in crop temperatures ($T_{c1}$) with the change in the channel width ($d$) of air stream-II from 0.04–0.20 m has been evaluated. The effect of change in channel width ($d$) of air stream-II on the crop temperatures ($T_{c1}$) is shown in Fig. 5. A diverse behavior in crop temperature is observed with the change in width of air channel. For the minimum channel width as 0.04 m of air stream-II, the crop temperature ($T_{c1}$) is initially lower for 6–7 h of operation; thereafter it increases and remains higher than the response of crop temperatures from the larger width of channel after 9 h of running. This trend becomes reverse while increment in the $d$ as 0.08, 0.12, 0.16, and 0.20 m. This may be explained as the mass flow rate is reduced at the smaller opening of air stream-II, which gives smaller effect of $T_{p1}$ and $T_{f2}$ on the $T_f$ due to low convective and radiative heat transfer and results in low $T_{c1}$. When the time proceeds, the $T_{p1}$ increases rapidly and $T_{f1}$ gets increased at slow mass flow rate due to more residence time is available. This results in increase of $T_f$ and $T_{c1}$ also. Whereas, this trend gets reversed for higher $d$. Therefore, at the larger $d$, $T_{c1}$ is higher for the first 9 h and lower in subsequent

Table 1
Input operating parameters used for numerical computation

| Parameters | Values |
|---|---|
| $a_w$ | 0.4 |
| $C_a$, $C_c$, $C_b$, $C_s$ | 1004.8, 3800, 700, 794 J kg$^{-1}$ K$^{-1}$ |
| $g$ | 9.81 m s$^{-2}$ |
| $H_u$ | 0.01 kg vapor/kg dry air |
| $K_{p2}$, $K_w$ | 135, 0.84 W m$^{-1}$ K$^{-1}$ |
| $M_0$ | 6.14 kg water/kg dry matter |
| $R$ | 8314.3 J K$^{-1}$ mol$^{-1}$ |
| $v$ | 0.5 m s$^{-1}$ |
| $\alpha_{p1}$, $\alpha_{p1}$, $\alpha_g$ | 0.8, 0.8, 0.05 |
| $\varepsilon_b$ | 0.4 |
| $\tau$ | 0.9 |
| $\rho_c$, $\rho_b$, $\rho_s$ | 950, 1900, 2700 kg m$^{-3}$ |
| $\rho'$ | 0.9 |
| $\sigma$ | $5.67 \times 10^{-8}$ W m$^{-2}$ K$^{-4}$ |

Table 2
Design parameters of greenhouse and dyer used for optimization

| Parameters | Values |
|---|---|
| $A_{p1}$ | $L \times b$ m$^2$ |
| $A_{p2}$ | $b(L - d)/\cos(30)$ |
| $a_r$ | 1 |
| $b$ | 1 m |
| $C_d$ | 0.6 |
| $D_b$ | 0.025 m |
| $d$ | 0.04, 0.08, 0.12, 0.16, 0.20 m |
| $H_b$ | 0.0, 0.05, 0.10, 0.15 m |
| $H_s$ | $A_{p2}$.sin(30) |
| $L$ | 1 m |
| $l_w$, $l_{p2}$ | 0.025, 0.002 m |

operation. However, 0.12 m $d$ has been considered appropriate for this design (Aboul-Enein et al., 2000), which effects less fluctuation in crop temperature.

### 5.3. Effect of height of packed bed on crop temperature

Effect of height of packed bed range from 0.0 to 0.15 m has been studied on the crop temperature and the results are presented in Fig. 6. The crop temperature is maximum as 78 °C during sunshine hours without packed bed ($H_d = 0.0$ m) and comes to minimum ($\sim T_a$) out of sunshine hours, since it gives the direct effect of $T_{p1}$ and $T_{f1}$ to the $T_{c1}$. When the packed bed is provided of height of 0.05 m (thin layer), which reduces the $T_{c1}$ (maximum to 50 °C) during the sunshine hours and this temperature is more suitable for crop drying. However, the small height of packed bed provides very small thermal storage effect, therefore $T_{c1}$ reduces gradually out of sunshine hours. The increasing of $H_b$ to 0.10 and 0.15 m provides the more thermal storage effect and low fluctuation in $T_{c1}$. A packed bed with the height of 0.15 m could be taken as appropriate.

### 5.4. Performance of crop dryer

The performance of the reversed absorber natural convective tray dryer was evaluated in terms of the hourly reduction in the moisture content of crop and the rate of drying with the design parameters as $L = 1$ m, $b = 1$ m, $d = 0.12$ m and $H_b = 0.15$ m. Hourly variation of the moisture content in both the drying trays is shown in Fig. 7. The 47.5 kg of onion slices in each tray was considered for drying. The onion slices were dried from an initial moisture content of 6.14 kg water/kg dry matter. The final moisture content that could be obtained was 0.30 and 0.27 kg water/kg dry matter in crop tray-I and II, respectively. The reduction in moisture content of crop in both the tray was at the same rate in the first 6 h of process, afterward a little faster moisture reduction is observed in tray-II for rest of the

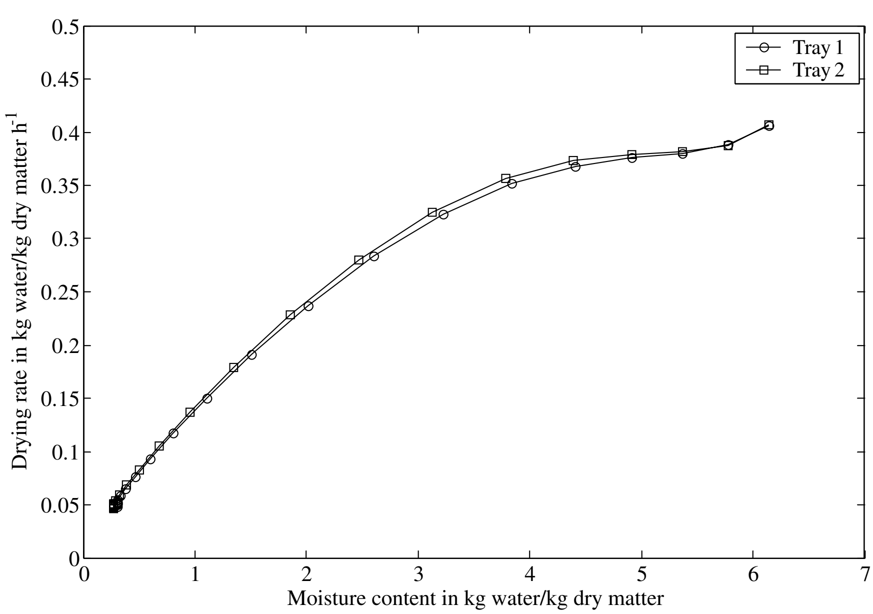


Fig. 9. Variation of drying rate with change in moisture content in crop.

drying time. This is due to slightly higher temperature of crop in tray-II. The hourly variation in drying rate is shown in Fig. 8. There was slightly linear reduction in drying rate for first 6 h of drying than a fast linear reduction up to 16th hour and thereafter the drying rate was steady for the rest of the drying time. The effect of moisture content on the rate of drying is presented in Fig. 9. The linear reduction in drying rate can be observed with reduction in moisture content of the onion crop. These results also concur with the work presented by Sarsavadia et al. (1999).

## 6. Conclusions

The performance of a reversed absorber natural convective crop dryer was carried out for drying onions in trays. The crop temperature depends on the width of the air flowing channel and height of packed bed. The thermal energy storage affects drying during the non-sunshine hours and is very pertinent in reducing the fluctuation in temperature for drying. The proposed mathematical model is useful for evaluating the performance of reversed absorber type collector and thermal storage with natural convective solar crop dryer. It is also useful for predicting the crop temperature, moisture content and drying rate of the crop.

## Appendix

The convective heat transfer coefficients from the plate to glass, parallel to each other and inclined at an angle $\beta$ to the horizontal has been expressed as

$$h_c = \frac{NuK_f}{d} \tag{A.1}$$

where Nusselt number ($Nu$) can be obtained by using expression given by Hollands, Unny, and Konicek (1976) for air as medium between the plate and cover;

$$Nu = 1 + 1.44\left[1 - \frac{1708}{Ra\cos\beta}\right]^{+}\left[1 - \frac{1708(\sin 1.8\beta)^{1.6}}{Ra\cos\beta}\right] + \left[\left(\frac{Ra\cos\beta}{5830}\right)^{1/3} - 1\right]^{+}$$
$$\text{for } 0 < Ra \leqslant 10^5 \text{ and } 0 \leqslant \beta = 0^\circ \tag{A.2}$$

The notation $[\,]^{+}$ is used to denote that only positive value to the term is to be considered else it is zero for negative value, where $Ra = \frac{g\beta' \Delta T d^3}{\upsilon_f \alpha_f}$.

The wind heat transfer coefficient from the cover to ambient (Watmuff, Charters, & Proctor, 1977)

$$h_o \& h_{c,ga} = 2.8 + 3.0v \quad (\text{for } 0 \leqslant v \leqslant 7 \text{ m s}^{-1}) \tag{A.3}$$

The radiative heat transfer coefficients are calculated as (Duffie & Beckman, 1991)

$$h_{r,p2g} = \varepsilon_{eff}\sigma(T_{p2}^2 + T_g^2)(T_{p2} + T_g) \tag{A.4}$$

where $\varepsilon_{eff} = \left[\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_{g2}} - 1\right]^{-1}$.

The expression of surface heat transfer given by Bradshaw and Myers (1963), which can be used as the convective heat transfer coefficient from crop to air in forced convection

$$h_c = 172.5\dot{m}_a^{0.5} \tag{A.5}$$

The volumetric heat transfer coefficient between the bed and air is given by the relation (Tiwari, 2002)

$$h_v = 824\left[\frac{G_0}{D_b}\right]^{0.92} \tag{A.6}$$

Side loss coefficient from dryer

$$U = \left[\frac{l_w}{K_w} + \frac{1}{h_o}\right]^{-1} \tag{A.7}$$

Thermal properties of moist air (Tiwari, 2002)

$$C_f = 999.2 + 0.1434T_f + 1.101 \times 10^{-4}T_f^2 - 6.7581 \times 10^{-8}T_f^3 \tag{A.8}$$

$$K_f = 0.0244 + 0.6773 \times 10^{-4}T_f \tag{A.9}$$

$$\alpha_f = 7.7255 \times 10^{-10}T_f^{1.83} \tag{A.10}$$

$$\upsilon_f = (0.1284 + 0.00105T_f) \times 10^{-4} \tag{A.11}$$

$$\rho_f = \frac{353.44}{T_f + 273.15} \tag{A.12}$$

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