Thermal Modeling of Indirect Solar Drying System: An Experimental Validation

A.K. Srivastava and S.K. Shukla

ABSTRACT

The expressions for crop and moist air temperatures, drying rate and efficiency of indirect solar drying with phase change material (PCM) storage systems in quasi-steady state conditions have been derived. The analysis is based on the basic energy balance for the system. A computer model has been developed to predict the performance of the solar dryers. Experimental validation of the thermal model has been carried out by using modified heat transfer coefficients. Internal heat and mass transfer coefficients have been evaluated with PCM for March 24, 2014 in Varanasi, India. A fair agreement has been observed between theoretical and experimental results by using the modified internal heat and mass transfer coefficients.

Keywords: Direct solar drying; indirect solar drying; PCM drying; Thermal efficiency

INTRODUCTION

Solar energy is an ideal alternative source of energy as it is abundant and inexhaustible. There are promising strategies for harnessing solar energy which are environmentally friendly and have less damaging impact on public health and security. Modern method of food preservation such as canning and refrigeration are expensive due to soaring prices of energy. In developing countries the traditional direct sun drying method is very slow. The shortcoming of this method has been overcome by the use of solar dryer.

Nomenclature

A_p	area of absorber plate (m^2)
A_{c1}	area of crop surface in tray 1 (m^2)
A_{c2}	area of crop surface in tray 2 (m^2)
A_{c3}	area of crop surface in tray 3 (m^2)
A_{ch}	area of drying chamber (m^2)
A_t	area of tray (m^2)
A_{pc}	area of PCM chamber (m^2)
m_f	mass flow rate of air (kg/ m^3)
m_{pcm}	mass of PCM (kg)
mc1	mass of crop in tray 1 (kg)
mc2	mass of crop in tray 2 (kg)
mc3	mass of crop in tray 3 (kg)
x	thickness of bottom insulation (mm)
k_{gw}	thermal conductivity of glass wool (w/ m°C)
L	length of collector plate (m)
W	width of absorber plate (m)
I	intensity of solar radiation (w/ m^2)
C_f	specific heat of air (J/ kg°C)
C_{spcm}	specific heat of PCM (J/ kg°C)
C_c	specific heat of crop (J/ kg°C)
Q_u	useful energy available from solar collector (w)
T_a	ambient temperature (k)
T_{pm}	mean plate temperature (k)
T_{bm}	mean bottom plate temperature (K)
T_{fi}	temperature of air at the inlet of collector (k)
T_{fo}	temperature of air at the outlet of collector (k)
T_{dco}	temperature of air at the outlet of dryer (k)
T_{c1}	temperature of crop surface in tray1 (k)
T_{c2}	temperature of crop surface in tray2 (k)
T_{c3}	temperature of crop surface in tray3 (k)
T_{ch}	temperature in drying chamber (k)
T_e	temperature just above crop surface (K)
U_b	bottom loss coefficient (w/ m^2)
Ut	top loss coefficient (w/ m^2)
h_{bf}	the convective heat transfer coefficient between bottom andfluid (w/ m^2 °C)
hpf	the convective heat transfer coefficient between plate and fluid (w/m^2°C)
h_{rpb}	the radiative heat transfer coefficient between bottom and plate (w/m^2 °C)

h_{ch}	heat transfer coefficient in drying chamber (w/ m^2 °C)
h_{c1}	heat transfer coefficient for tray1(w/m^2C
$h_{c1}2$	heat transfer coefficient for tray 2 (w/ m^2 °C)
h_c3	heat transfer coefficient for tray3 (w/m^2°C)
hpg	convective heat transfer coefficient between plate and glas
hga	the convective heat transfer coefficient between ambient air and glass cover (w/ m^2°C)
h_{rc}	radiative and convective heat transfer coefficient from crop surface to environment (W/m^2° C)
h_i	overall bottom loss coefficient for drying chamberW/ m^2° C)
h_{2c}	wind heat transfer coefficient (W/ m^2° C)
v	wind velocity (m / sec)

Greek letters:

τ	transmitivity of glass cover (0.9)
α_p	absorptivity of absorber plate (0.85)
λ	latent heat of evaporation
σ	Stephen boltzman, constant
γ	relative humidity of moist air
η	efficiency
ε_p	absorber plate emittance
ε_g	glass cover emittance

Solar drying technology offers an alternative which can process the vegetables and fruits in clean hygienic and sanitary conditions to national and international standards with zero energy costs. It saves energy, time, occupies less area, improves product quality, makes the process more efficient and protects the environment. Solar drying can be used for complete drying process or as a supplement to artificial drying system in the later case reducing the fuel energy required.

Literature Survey

As a result of extensive literature survey we come to the finding that drying is perhaps the oldest and most diverse of chemical emerging unit operations. Over 400 types of dryers have been reported whereas 100 distinct type are commonly available [1]. In the past, several drying methods have been examined to produce high quality of dehydrated food. Studies on open sun drying of various fruits and vegetables have been reported, Sachin V. Jangam.

C.L. Law and A.S. Majumdar [2]. Torgul and Pehlivan, 2004 [3]; Torgul and Torgul 2007 [4]and Shukla et al. 2008 [5]. It is also to be noted that the method employed for drying process has a significant influence on the quality of dehydrated food products in terms of color, texture and sorption properties characteristics, Krokida & Maroulis 1997 [6], Krokida et al. 1998 [7], Yaung and Atallah 1985 [8]; Ehsan Mohseni Languri et al. [9] manufactured and tested a flat plate solar air heater in the north of Iran and connected to a room as the model to study the possibility of using such solar heating system in the north of Iran. The experimental data obtained through accurate measurements were analyzed using Second Law approach to find the optimum flow rate which leads to maximum energy efficiency. It was pointed out that the maximum irreversibility occurs at noon when radiation reaches maximum and it decreases as solar energy decreases. T. Kiatsiroat et al. 2007 [10] performed an experimental study on the performance of a flat plate solar air heater in presence of electric field. In this work a model was developed to predict the heat transfer data and the predicted results agree very well with the experimental results.

Ekechukwu and Norton et al. 1999 [11] presented a comprehensive review on design, construction and operation of different types of solar dryers. However all these dryers can be broadly grouped into three major types as direct, indirect and mixed mode, depending on arrangement of system components and modes of solar heat utilization. In the same year, a comprehensive review of the fundamental principles and theories governing the drying agricultural products has been presented by O.V. Ekechukwu et al. 1999 [12]. A chronological development of the purely theoretical, semi theoretical and empirical models has been outlined. Theoretical models have fallen short of predicting accurately the exact processes involved in drying, due to over simplification of assumptions. Empirical models for specific products and conditions offer better predictions.

Since drying commonly describes the process of thermally removing volatile substances (moisture) to yield a solid product. According to Sodha et al. 1985 [13], drying depends upon the rate at which the moisture within the product moves to surface by diffusion process depending upon the type of product. There are three modes of drying open Sun, Direct and Indirect in presence of solar energy. In open sun drying, the crop is spread in a thin layer on ground and exposed directly to solar radiation, wind and other ambient conditions. It is well known fact that the task of modeling heat transfer between the crop and surrounding air in the presence of solar energy is a complex phenomenon. An attempt has been made by Anwar and Tiwari et al. 2001 [14] for O.S.D for different crops using linear regression analysis. It has been observed that convective heat transfer vary significantly from crop to crop due to presence of different levels of moisture content. Their results are within 16% internal uncertainty (one standard deviation of the measurement error). Garg and Kumar et al. 2000(15) developed an analytical model to determine the drying characteristics of any product under O.S.D.

An analytical model based on the principal of heat and mass transfer has been presented by Sodha et al. 1985 [16] in which he has included the effect of wind speed, relative humidity, product thickness and heat conducted to the ground along with heat and mass transfer. The analytical models are helpful in predicting the hourly variation of product temperature and moisture content under constant rate and falling period of drying. Sodha M.S., Dang A, Bansal P.K., Sharma S.B. et al. 1985 [17] developed a theoretical and experimental study of solar cabinet dryers. The experimental results has shown that cabinet dryers were very useful for domestic applications for drying fruits and vegetables of high moisture content. The overall efficiency in open sun drying is much less than that of the cabinet dryer and quality of product is maintained.

Arata and Sharma et al. 1999 [18] developed a simple design using simple tools and relatively cheap and locally available materials by small scale industries using solar heater with simple design.

Pawar et al 1995 [19] designed and fabricated a large scale forced convection solar drying system comprising an array of forty solar collectors and three drying cabinets with a blower. It was shown that use of this type system was feasible and had an ability to save large amounts of fuel. It keeps the product clean and it was dried in shorter period than in open sun drying.

Kiatsiroat et al. 2007 [20] performed an experimental study on the performance of a flat plate solar air heater in presence of electric field. In this work a model was developed to predict the heat transfer data and the predicted results agree very well with the experimental results.

Akinola, A.O. and Fapeta et al. [21] performed exergetic analysis of a mixed mode solar dryer. Exergetic analysis of the dryer revealed that drying in a cabinet other than Sun drying made drying more attractive and as well conserve energy. It has an overall average exergetic efficiency of 50% and thermal efficiency of 66.95%. Onder Ozgener, Arif Hepbasil et al. 2006[22] presented a review on the energy and exergy analysis of solar assisted heat pump systems. M.K. Gupta et al. [23] performed a comparative study of various types of artificial roughness geometries in the absorber plate of solar air heater and their characteristics. Donation Njome et al. [24] developed a mathematical model to analyze the thermal performance of four different types of solar air collector in a typical climatic conditions.

Thermal Energy Storage (TES) and Phase Change Material (PCM).

The idea of using solar energy during rainy season or when energy availability is inadequate was always the point of attraction among researchers. Among various energy storage techniques of interest, latent heat storage is particularly attractive because of its ability to provide a high energy storage density and its ability to store energy at a constant temp to the phase transition temp of the energy storage substance. Since nature of solar energy is highly intermittent, unpredictable, and available only during the daytime hence its application requires efficient thermal energy storage so that the surplus heat collected during sunshine hours may be stored for later use during night. In thermal energy storage, the useful energy from the collector is transformed into internal energy. This may occur in the form of latent heat, sensible heat, or both.

Latent heat storage is a new area of research and pioneered by Dr. Telkes in the 1940s (Lane, 1983). It did not receive much attention, however, until the energy crisis of the late 1970s and early 1980s. PCM was first used in British trains against coolness. The first application of PCM described in the literature was their use for heating and cooling in the buildings, by Telkes (1975) [25], and Lane (1983)[26]. Telkes at al. (1978)[27] published the idea of using PCMs in walls, better known as Trombe walls. Although research into latent heat storage for solar heating system continues. Literature reveals that Sakamon Devahastin, Saovakhon Pitaksuriyarat et al. 2005 [28] investigated the feasibility of using [L.H.S] with paraffin wax as a phase change material (P.C.M.) to store excess solar energy and release it when the availability is inadequate or not available. S.D. Sharma, Sagara Kazunobu et al. 2005[29] made an effort to find a suitable P.C.M. for various purposes, a suitable heat exchanger with ways to enhance heat transfer, to provide a variety of designs to store the heat using, PCM for different applications.

Moh. M. Farid, Amar M. Khudhair, Said Al-Hallaj et al. 2004.[30] has reviewed the work on latent heat storage and provides an insight to recent efforts to develop new classes of PCMs for use in energy storage. Three aspects have been the focus of this review, PCM materials encapsulation and application. The problems associated with the application of PCM with regards to the materials and the methods used to contain them are also addressed. Hardorn, J.C. et al. 2004 [31] has suggested many storage solutions for solar thermal energy.

Amar M. Khudhair, Moh M. Farid, et al. 2004 [32] has reviewed the work on energy conservation in building applications with thermal energy storage by latent heat using PCM materials. Energy storage in the walls, ceiling, and floor of a building may be enhanced by encapsulating suitable PCM materials within these surfaces to capture solar energy directly and increases human comfort by decreasing the frequency of internal temp swings and so temp is maintained closer to the desired temp for a longer period of time.

M.M. Alkilani, K. Sopian, Sohif Mat, M.A. Alghoul et al. 2009[33] has presented a thermal investigation of output air temperature due to thermal energy discharge process from a PCM unit consisting of inline single row of cylinders containing a compound of paraffin wax with aluminum powder.

V.V. Tyagi, A.K. Pandey, S.C. Kaushik, S.K. Tyagi et al. 2011[34] has performed experimental study for evaluating thermal performance of a solar air heater with and without thermal energy storage. Three different arrangements viz without PCM, with PCM and with hytherm oil were used to study the comparative performance of the experimental setup. The measured parameters were inlet, outlet temperature and radiation with respect to time have been recorded and that the outlet temperature in case with thermal energy storage is higher than that of without TES, besides the outlet temperature with paraffin wax is slightly greater than that of with hytherm oil. Also then is no energy gain in the evening in the case of with TES but in case of with TES then is a heat gain for around 4 hrs in the evening which gives the backup for hot air for around more h0urs which is the main advantage of this system with TES. Based on this data, the efficiency of the system has been calculated and it is noted that the efficiency in the case of heat storage is higher than that of without TES, besides the efficiency in the case of paraffin wax is slightly higher than that of the hytherm oil case.

Shukla S.K., Saraswat D.C., and T. Raj et al. 2008[35] has performed a study on open house and green house drying and evaluated the convective heat and mass transfer coefficients as a function of climatic parameters. This was found that the value of heat and mass transfer is more in OSD than in Green House Drying under natural convection mode. However, its value increases in forced mode of Green house drying when compared with natural mode in the initial stage of drying. The objective of the investigation is to predict and validate the thermal models of solar drying system with PCM as energy storage and alternate energy backup system.

Research Objective and Scope

The present study deals with the experimental analysis of thermal performance of a solar drying system for a clear day in a composite climate. Experimental data have been recorded for two days for a period of seven hours in a day. To study the variation of air mass flow rate on thermal performance, data have been taken during the day time when the solar radiation does not vary much (10:30 a.m. to 11:30). The experimental data obtained through measurements for the most typical day have been analyzed using first law and second law efficiencies to determine variation of collector efficiency and dryer efficiency, energy and exergy efficiency and mass flow rate of air. In this article, an attempt has been made to evaluate the convective heat transfer for indirect solar dryers by using the expression for Nusselt number as used by Tewari and Suneja et al. (1997) [36].

EXPERIMENTAL SETUP

The drying system is shown in Figure 1. The system consists of four flat plate solar air hot collectors, with 1.93 m2each cabinet. Each drying chamber accommodates 08 trays, each tray is 760 × 555 mm. The PCM material used as thermal energy storage was kept in an air chamber of the cabinet. A blower of 0.37 kW and 2785 rpm is used to force the air through the solar collector. Solar energy collected by solar collector is taken away by atmospheric air which is forced through collector and finally which enters into drying chamber via air chamber which causes the dehydration of fruit or vegetable kept on trays.

Thermal charging of phase change materials during the day time takes place and the energy is stored in the form of sensible and latent heat. The energy stored may be utilized when the solar radiation is not sufficient due to bad weather conditions or intensity of incident solar radiation is very weak and fluctuating. Lauric acid was used as PCM material for storing thermal energy in latent heat form and it is converted into heat whenever required.

A schematic diagram of simple cabinet dryer, PCM chamber and indirect solar drying system are shown in Figures 2 and 3, respectively. Indirect solar drying or conventional solar drying incorporates a separate unit termed as solar air heater which is used for the collection of solar energy for heating of entering air into this unit. The air heater is connected to a separate drying chamber where the crop is kept. Here, the heat from moisture evaporation is provided by convective heat transfer between the hpt air and the wet crop. The drying is basically achieved by the difference in moisture concentration between the drying air and the air in the vicinity of the crop surface.

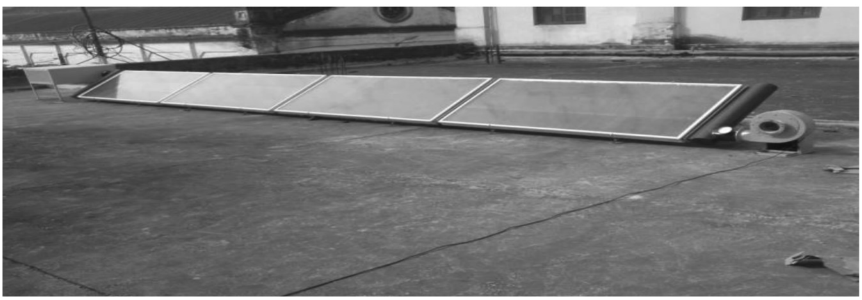

Figure 1. Solar Collector

Figure 2. Solar Dryer

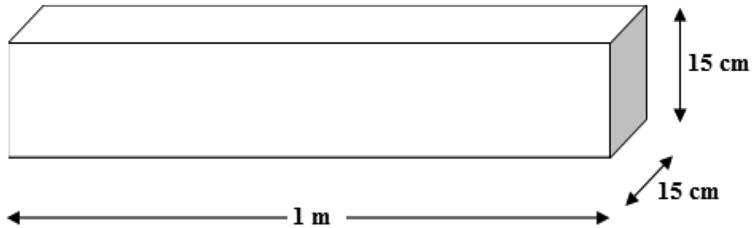

Figure 3. PCM chamber

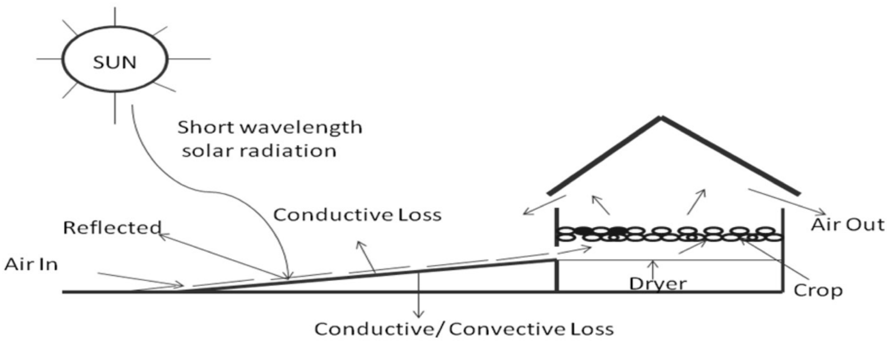

Figure 4. Indirect Solar Cabinet Drying

The experiments were conducted on the roof of Renewable Energy Lab, Department of Mechanical Engineering, Indian Institute of Technology, Banaras Hindu University, Varanasi, India for several days during the month of June 2013 and March 2014. The latitude, longitude and altitude of Varanasi are, 25.2°N and 83.00°E, 80.71 m above the sea level. The data presented in the table 1-4 correspond to two typical days namely 14 March 2014 and 24 March 2014. The analog type solarimeter instrument (type SM201) has been used to measure the solar radiation of intensity in the range of 1-100 mW/cm^2 with an accuracy of 2 mW/cm^2. The temperatures at required places were measured by thermocouples, which were connected to digital temperature indicator. The indicator used to measure the temperature in the of 0-200°C range had an accuracy of 0.1°C. Wind velocity has been measured by two channel hot wire anemometer. The sunny period, one hour more has been taken for summer in comparison to winter. Experimental data are used to calculate the convective heat transfer coefficient for indirect solar dryers. These are represented in Table [1].

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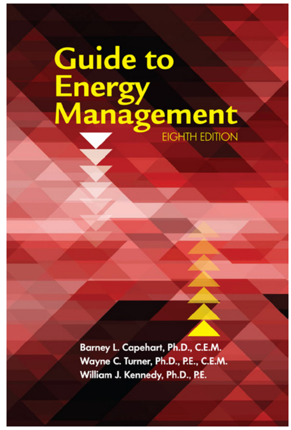

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Experimental Uncertainty

The experimental method used for estimating the convective heat transfer coefficient based on mass of moisture evaporated is an indirect approach. Therefore the result will have a considerable degree of experimental uncertainty. An estimation of uncertainty has been carried out for indirect type of drying systems. Data of a particular measurement for a number of days have been taken and an estimate of individual uncertainties of the sample values has been calculated. An estimate of internal uncertainty U_i can be then found by using

$$U_i = \sqrt{\frac{\Sigma\sigma i2}{N2}}$$

where σi is the standard deviation of the ith and N is the total number of samples.

Mathematical Models

The following assumptions have been made in order to write the energy balance equations for the various components of drying system.

i) Air heater and drying chamber have been assumed as individual system.

ii) The temperature gradient through PCM is negligible. It means Tpcm represents average temperature through the PCM chamber.

iii) The mode of heat transfer between PCM and the air during the discharging period of stored energy is purely convection.

iv) Heat conduction through PCM and drying chamber insulation is unidirectional.

v) Thermal, physical and transport properties of plate and air are uniform over the entire collector area and do not change with temperature.

vi) Radiation, and convection losses takes place only from the collector' top surface.

vii) Heat conduction and air flow are one dimensional and are perpendicular to the bounding surfaces of matrix.

Thermal Modeling

In proposed methodology, the energy balance equation of indirect dryer is written by considering air heater and drying chamber as the individual systems. Thus the governing equation for the determination of drying rate and T_{pcm} under steady state condition, the rate of thermal energy supplied to air collector is utilized for meeting thermal losses from various individual components of air collector and drying chamber through conduction, convection and radiation.

Solar Air Heater

$$\tau \eta_p I. Wdx = U_t(T_p - T_a). Wdx + h_{pf} (T_p - T_f). Wdx \tag{1}$$

Working Fluid

$$h_{pf} (T_p - T_f). W dx = m_f.C_f \frac{dTf}{dx} dx + U_b (T_f - T_a)w.dx \tag{2}$$

From Equation (1), we get

$$T_p = (\tau \eta_p I + U_t. T_a + h_{pf}. T_f)/(U_t + h_{pf}) \quad (3)$$

Substituting the value of T_P in equation (2), and arranging the terms we get—

$$h. \tau \, ap I + Ufa (T_a - T_f) - U_b (T_f - T_a) = \frac{m.cf}{w} . \frac{dTf}{dx} \tag{4}$$

Where $h = \frac{hpf}{Ut+hpf}$, and $U_{fa} = \frac{Ut.hpf}{Ut+hpf}$

On further rearranging the terms, we get

$$\frac{dTf}{dx} + a. T_f = f(t) \tag{5}$$

Where $f(t) = \frac{W.(ftapl+(Ufa+Ub).Ta}{mf.Cf}$ and $a = \frac{(Ub+Ufa).W}{mf.Cf}$

Solution of differential equation (5) under boundary condition at x=0, $T_f = T_{fi}$ becomes—

$$T_f = T_{fi} . e^{-ax} + (1- e^{-ax}). \frac{f(t)}{a} \tag{6}$$

The rate of useful energy available to drying chamber—

$$Q_u = m_f. C_f [T_f / \text{at } x = L - T_{fi}] \; Q_u = m_f C_f (1 - e^{-aL}). \{ \frac{f(t)}{a} - T_{fi}\} \tag{7a}$$

$$\text{collector Efficiency } \eta_c = \frac{Q_u}{I_t.A_c} \tag{7b}$$

Drying Chamber

Energy balance equation for drying chamber between crop and drying chamber area may be given as—

$$m_f C_f (T_{fo} - T_{fi}) = M_c C_C \frac{dTc}{dT} + h (T_c - T_{ch}). A_{ch} \tag{8}$$

$$h. (T_c - T_{ch}) = \dot{m} .C_f (T_{ch} - T_a) + h_s A_s. (T_{ch} - T_a) \tag{9}$$

Energy balance equation on crop surface for moisture evaporation can be written as—

$$Q_u - h_{rc} (T_c - T_e) - 0.016 h_c [P(T_c) - P(T_e)] - h_i (T_c - T_a) = \frac{Mc.Ce}{At} \frac{dTc}{dt} \tag{10}$$

Energy balance equation of moist air above the crop surface may be written as—

$$h_{rc} (T_c - T_e) + 0.016 h_c [P(T_c) - \gamma P(T_e)] = h_2 (T_e - T_a) \tag{11}$$

Where h_2 = convective and radiative heat transfer from environment to ambient air

$$= 5.7 + 3.8 V \tag{12}$$

From equation No (11) we have—

$$T_e = \frac{hrcTc + h2Ta + 0.16\ hc\ [P(Tc)–P(Tc)]}{h2 + hrc} \tag{13}$$

Putting the value of Te in equation No.(10) and rearranging the terms we get—

$$\frac{dTc}{dt} + a.\ T_c = f(t) \tag{14}$$

$$\text{where } a = \frac{(hi+u1).Ad}{mc\ Cc}$$

$$\text{and } f(t) = \frac{Ad}{mc.Cc}\ [m_{f.}\ c_f\ (1 - e^{-aL})\ \{\frac{f(t)}{a} - T_{fi}\} - \frac{h2}{h2+hrc}\ .\ 0.16\{P(T_c) - \eta\ P(T_e)\}-(h_i+u_1)\ T_a]$$

$$\text{and } u_1 = \frac{h2.hrc}{h2+hrc}$$

Solution of equation No.(14) with boundary condition at t = 0, $T_c = T_{co}$

$$T_c = T_{co}\ .\ + (1-\ .\ e^{-at})\ .\ \frac{f(t)}{a} \tag{15}$$

$$\text{Moisture evaporated } m_{ev} = \frac{hew}{\lambda}\ .(T_c - T_e) \tag{16}$$

$$\text{Where } h_{ew} = 0.016\ h_c\ \frac{[P(Tc)–\lambda\ P(Te)]}{(Tc–Te)}$$

$$P(T) = \exp\ [25.317 - \frac{5144}{273+T}]$$

Energy balance equation for PCM

$$m_f\, C_f\, (T_{fo} - T_{fi}) = m_{pcm} C_{spcm} \frac{dTpcm}{dt} + h_i\, (T_{pcm} - T_a) A_{pc} \quad (17)$$

From equation No (14) after rearranging the terms we get—

$$\frac{dTpcm}{dt} + a_p = f(t) \quad (18)$$

where $a_p = h_i\, A_{pc} / m_{pcm} C_{spcm}$ and $f(t) = .\{m_f\, C_f\, (T_{fo} - T_{fi}) + h_i\, T_a A_{pc}\} / m_{pcm} C_{spcm}$

Solution of above differential equation with boundary condition at t= 0, $T_{pcm} = T_{pcm0}$, may be given as—

$$T_{pcm} = T_{pcm0}\, e^{-ap.t} + \frac{f(t)}{ap} \,.\, [1- e^{-ap.t}] \quad (19)$$

Energy balance equation during the discharge of PCM—

$$\frac{Meq}{Ad} \frac{d\mathrm{Tpcm}}{d\mathrm{T}} = 0.16\, h_w\, [P(T_c) - \eta\, P(T_e)] + h_i (T_{pcm} - T_a) \quad (20)$$

Convective heat transfer coefficients

According to Tiwari and Suneja, 1997[36] the convective heat transfer coefficient for forced convection drying system can be computed by the following equations—

$$N_u = \frac{heX}{Kv} = C\, (R_e\, P_r)^n \text{ (for forced convection drying)} \quad (21)$$

$$h_c = \frac{Kv}{X}\, C\, (R_e\, P_r)^n \quad (22)$$

The heat required to evaporate moisture from the crop is given by the equation—

$$Q_e = 0.016\, h_c\, [P(T_c) - \eta\, P(T_e)] \tag{23}$$

From equation No. (21) and (23), we get-

$$Q_e = 0.016\, \frac{Kv}{X}\, C\, (R_e\, P_r)^n\, [P(T_c) - \eta\, P(T_e)] \tag{24}$$

Moisture evaporated $m_{ev} = \dfrac{Qe}{\lambda}\, t\, A_t = 0.016\, \dfrac{Kv}{X\lambda}\, C\, (R_e\, P_r)^n$

$$[P(T_c) - \eta\, P(T_e)]\, t.\, A_t \tag{25}$$

Let us assume that $.016\, \dfrac{Kv}{Z}\, [P(T_c) - \eta\, P(T_e)]\, t.\, A_t = Z$

$$\text{Therefore, we get } \frac{mev}{Z} = C\, (R_e.\, P_r)^{\,n} \tag{26}$$

Taking logarithms on both sides of equation (26) we get—

$$l_n\, [\frac{mev}{Z}] = l_nC + n.\, l_n(R_e.P_r) \tag{27}$$

The above equation may be expressed as- $Y = m.X_0 + C_0$

Where m= n, $X_0 = \ln(R_{e.}\, P_r)$ and $C_0 = l_nC$

$$\text{Thus we have the solution, } C = e^{c0} \tag{28}$$

Therefore by using the data of Tables 1 and 2, the values of Y and X0 were calculated for different time interval and constants C and n were evaluated from above equations using method of linear regression for solar drying under direct and indirect drying mode.

According to Anwar and Tiwari 2001a:[37] following expressions have been taken for the calculation of physical properties of humid air—

Specific heat $C_v = 999.2 + 0.143\, T_i + 1.101\times 10^{-4}\, T_i^2 - 6.7581\times 10^{-8}\, T_i^3$

Thermal conductivity $K_v = 0.024 + 0.7673 \times 10^{-4}\, Ti$

$$\text{Density } \eta_v = \frac{353.44}{Ti+273.15}$$

$$\text{Viscosity } \mu_v = 1.718 \times 10^{-5} + 4.620 \times 10^{-8}\, T_i$$

$$\text{Partial pressure of vapor, } P(T) = \exp[25.317 - \frac{5144}{T+273.15}]$$

$$\text{and mean temp } T_i = \frac{Tc^{-} + Te^{-}}{2}$$

Table 1. Experimental observation of march 24, 2014 for forced convection in-direct solar drying(with PCM)

Time (hr)	Incident Radiation (It) (watt/m^2)	Ambient air Temp (K)	Collector Inlet Temp (K)	Collector Outlet Temp (K)	Dryer Outlet Temp (K)	Temp of PCM (K)	Crop temp in tray1 (Tc1)	Crop temp in tray2 (Tc2)	Crop temp in tray3 (Tc3)	Relative Humidity in drying chamber γ(%)
9:00	510	306	300.2							12.8
10:00	888	307	303	327	321	324	326	324	322	11.8
11:00	905	307	305	329	324	326	328	325	324.5	11.7
12:00	965	304.5	307	332	323.5	327	331	327	325.5	10.5
13:00	880	304	308	333	328	331	332	330	329	10.4
14:00	803	303	309	328	325	331	327	326	325	10.6
15:00	521	300	310	326	321	324	325	324	322	10.4
16:00	299	298	309	320	316	319	319	318	317	10
17:00	27	296	309	317	314	317	316	315	315	10

Table 2. Experimental observation of march 14, 2014 for forced convection in direct solar drying(without PCM)

Time (hr)	Incident Radiation (It) (watt/m^2)	Ambient air Temp (K)	Collector Inlet Temp (K)	Collector Outlet Temp (K)	Dryer Outlet Temp (K)	Relative Humidity in drying chamber γ(%)
9:00	640	300	301			15
10:00	665	301	303	337	320.6	13
11:00	733	302.5	305	332	324.3	12
12:00	780	302.7	305	332	324.5	15
13:00	685	303.2	305	317	315.5	14
14:00	459	303.7	305	313	310.2	15
15:00	344	303.5	304	310	307.5	16
16:00	140	301.5	302	308	306.1	16
17:00	100	301	301	306	304	16

EXPERIMENTAL RESULTS AND DISCUSSION

Referring to data of Tables 1 and 2 for forced convection indirect solar drying with and without P.C.M. material (Lauric acid) on 24/03/14 and 14/03/14, the hourly variation of incident solar radiation on both days has been represented in Figures 1 and 2, respectively. Intensity of radiation is maximum at 12 noon and all other temperatures namely collector inlet and outlet temperature, dryer outlet temperature, and crop temperatures in all three trays are maximum at the same timing.

A computer program has been developed in MATLAB 7.9 to predict the performance of solar collector. Top loss coefficient, bottom loss coefficient, side loss coefficient and useful energy available from solar collector for drying purpose has been computed. A fair agreement between experimental and predicted value of collector efficiency has been achieved and is shown in Figure 7.

Dryer efficiency as shown in Figure 8, varies from 7% to 14% on 24/03/14 with P.C.M and it is maximum between 12.00 Noon to 1.00 Noon but it decreases to 4% around 3.00 P.M. After 3.00 P.M a steep increment in dryer efficiency is observed probably due to back up energy source provided by P.C.M. Figure 6 shows the resemblance of experimental value of hourly P.C.M. temperature with predicted value.

Performance Studies- Experimental Validation:

Three parameters have been used in the modeling for the computation of c and n in the linear regression analysis. See Table 3.

Climatic Parameters

The hourly variation of temperature variation of ambient temperature and solar intensities falling on the collector has been given in Tables 1 and 2.

The value of observations for forced convection drying mode were recorded in Table 1 for direct indirect drying (with PCM) and Table 2 for direct drying (without PCM), for potatoes of 2.5 mm thickness kept in three trays. The average of crop surface temperature (T_c) and exit air temperature from dryer (T_{dco}) and exit air relative humidity were used to determine the physical properties of humid air which were further used to calculate the value of Reynold number and Prandtl number for the crops kept in trays. The values of c and n for tray 1, tray 2, and tray 3 in Equation 22 were obtained by simple linear regression analysis as

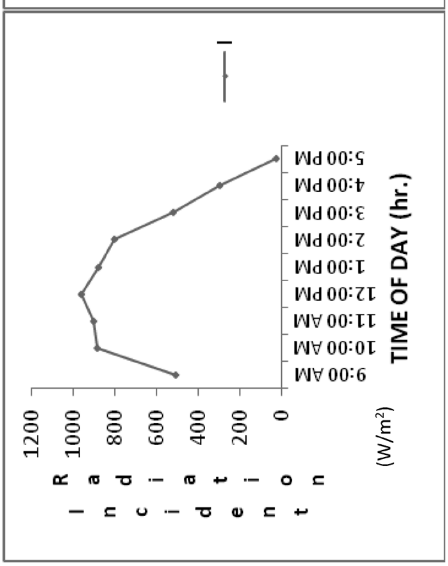

Figure 5. Hourly data on 24/03/14

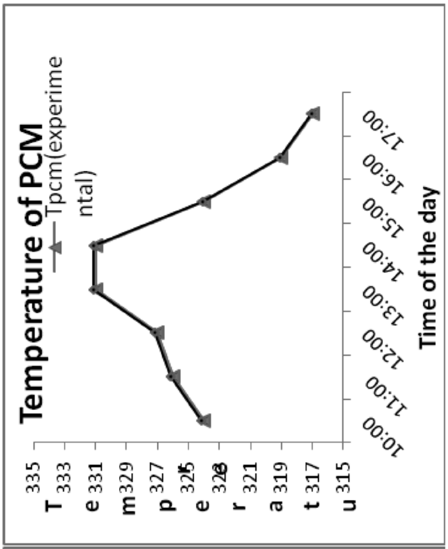

Figure 6. Hourly variation of PCM temp

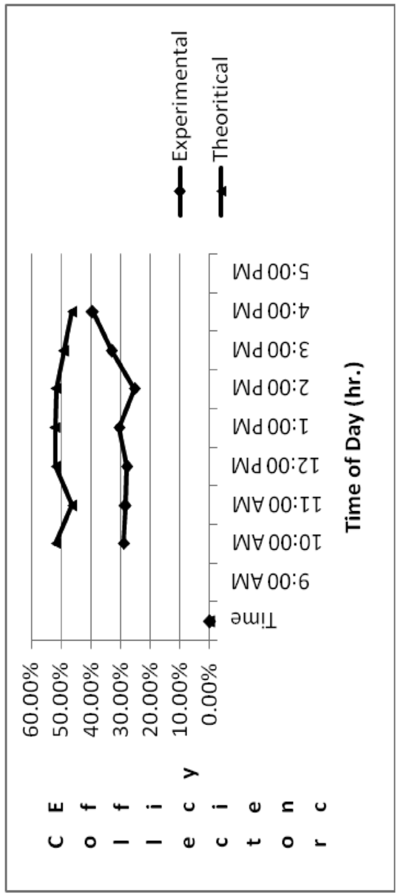

Figure 7. Hourly variation of theoretical and experimental collector efficiency on 24/03.14

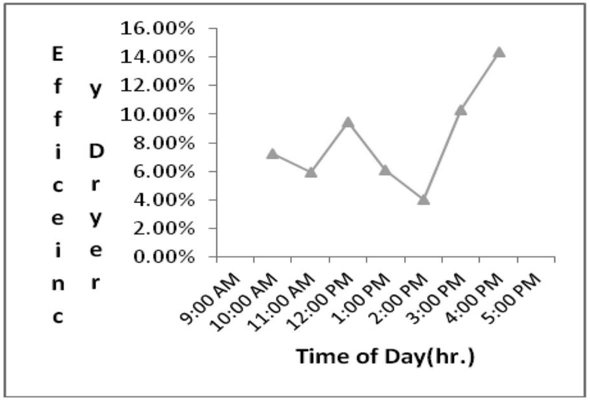

Figure 8. Hourly variation of ηd n 24/03/14

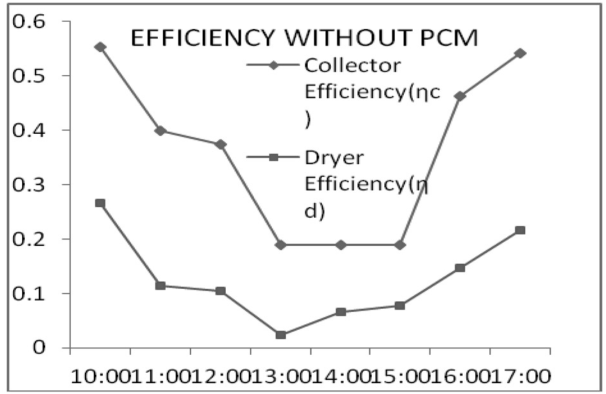

Figure 9. Hourly variation of ηc & ηd on14/03/14

Table 3. Design Parameters

A_t	0.76 x 0.555 m^2	λ	2.26exp (6) j/ kg^{o}C	h_i	5.7 (w/m^2)
g	9.81 m/ sec^2	σ	5.67 exp (-8)	C_c	3.67 (KJ/ Kg^{o}C)
X	(L+B) / 2 = 0.6575 m	t	60x60 sec	C_{spcm}	2.15(KJ/ Kg^{o}C)

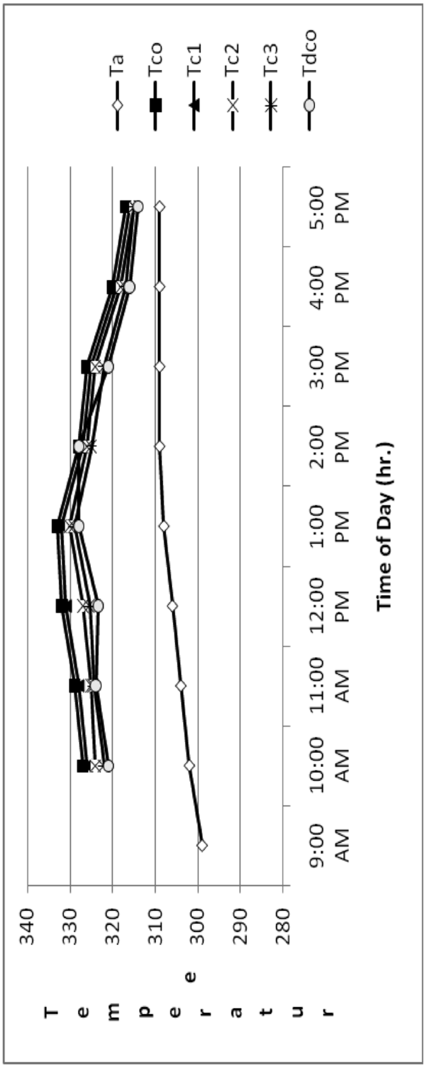

Figure 10. Hourly variation of various temperatures on 24/03/14

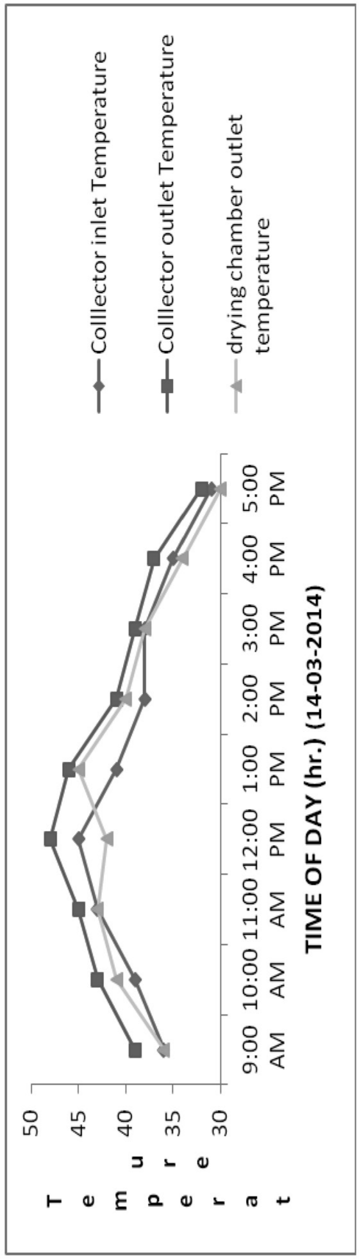

Figure 11. Hourly variation of temperature on 14/03/14

shown in Table 3 and Table 4. The value of heat transfer coefficients for trays has been computed using equation 22b and the result has been represented in Table 5. Forced convection heat transfer were also calculated using correlation equation by Kays and Graw Ford 1980 as

$$N_u = h_c\, X/K = 0.0158\, Re^{0.8} \text{ for } R_e > 2100\ \&\ L/D_n \text{ is large.}$$

The value of c and n after regression analysis were again used and modified heat transfer coefficient were computed using the formula

$$N_u = h_c\, X/K = C.\, (Re\ Pr)^n.\, K/X$$

Average value of heat transfer coefficient (experimental) and average value of heat transfer coefficient (modified) for tray 1, tray 2 and tray 3 has been represented in Table 5. The results show that average value of modified heat transfer coefficient is more than the experimental value but the trend of variation of graph is almost same. This is because the used experimental method was an indirect approach for determining the convective heat transfer coefficient based on the mass of moisture evaporated from potato surface. Thus, a considerable degree of experimental uncertainty in the estimation of convective heat transfer coefficient already exists in the approach.

The uncertainty in collector and dryer efficiency has been computed as 0.0859 and 0.0507 respectively. Results tabulated in Table 6 are within the range of errors and uncertainty.

DRYING KINETICS OF POTATO CROP

Drying Rate Calculation

According to Itodo, et al. (2002), the drying rate is the quantity of moisture removed from the food item ie potato in a given time.

$$\frac{dM}{dt} = \{(M_i - M_f)/t\}/100 \tag{30}$$

Where M_i = Initial moisture content (% dry basis, M_f = Final moisture content (% dry basis)

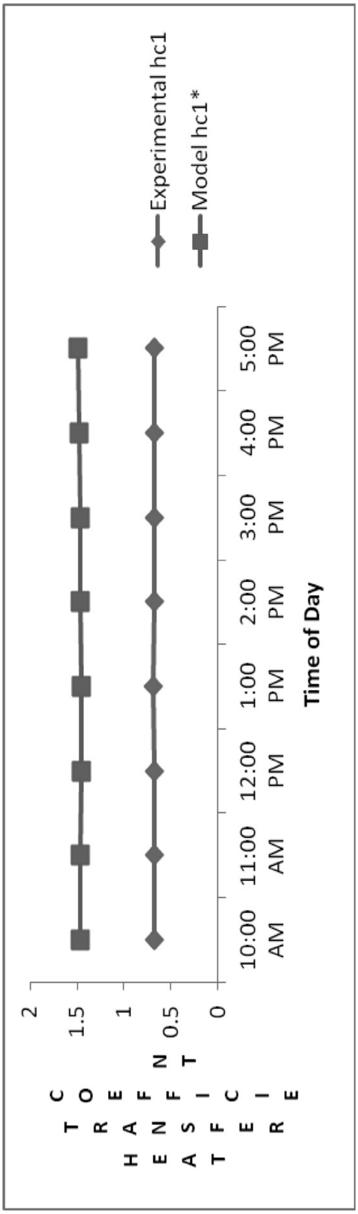

Figure 12. Hourly variation of heat transfer coefficient for tray1 on 24/03/14

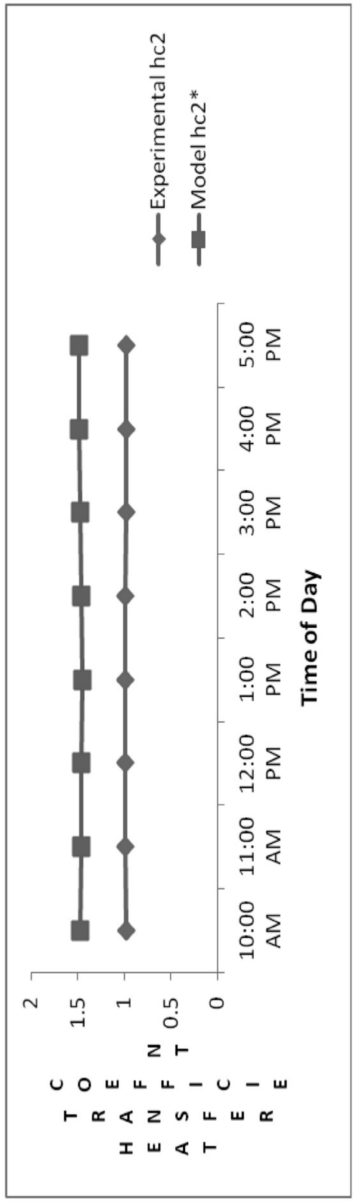

Figure 13. Hourly variation of heat transfer coefficient for tray1 on 24/03/14

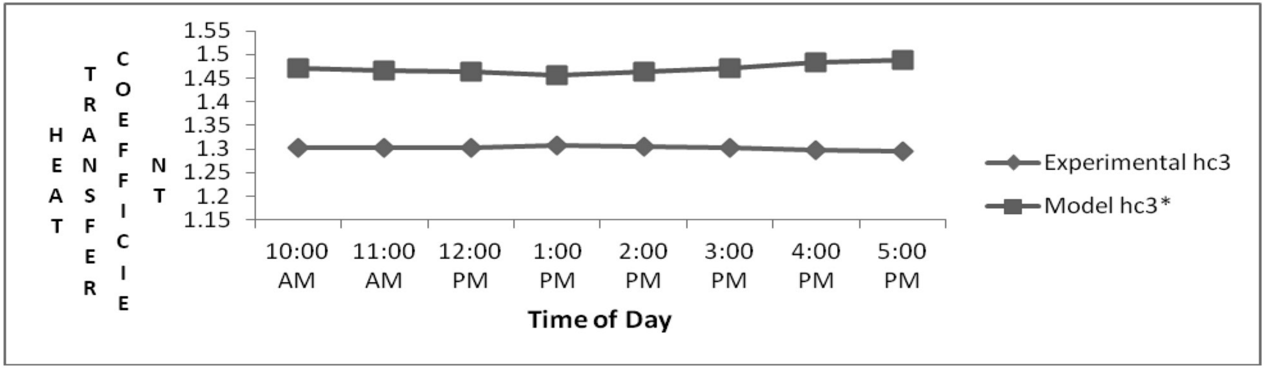

Figure 14. Hourly variation of heat transfer coefficient for tray1 on 24/03/14

Table 4. Values of C, n and hc under forced convection in direct drying modes

Sno.		C	n	hc(W/m2^{o}C) average	hc(W/m2^{o}C) model avg
1	Tray1	0.999337	0.299193	0.680602815 4	1.467139425
2	Tray2	0.996385	0.33969	0.985517409	1.469403035
3	Tray3	0.997236	0.369778	1.301949165	1.470768959

The mass of water evaporated is calculated as—

$$Mw = \frac{mi(Mi-Mf)}{100-Me} \tag{31}$$

Where m_i = initial mass of the food item, M_e = Equilibrium moisture content (% dry basis)
M_i = Initial moisture content (% dry basis).

Drying rate was experimentally obtained for all the three trays as 93.125 gm/hr,129.25 gm/hr and 171.25 gm/hr. The trend of variation is almost identical for all the trays. The drying rate v/s time relationship has been represented from Figures 16, 17 and 18. The best fit curves which define the drying behavior of potato crop kept in trays are given below.

Drying rate for tray1 = $f(x) = 2.06e^{-3.006x} + 0.02535e^{-0.2969x}$, R2=0.9996

Drying rate for tray2 = $f(x) = 0.0187e^{-0.2133x} + 1.038e^{-2.339x}$, R2= 0.9994

Drying rate for tray3 = $f(x) = 0.01644e^{-0.1858x} + 0.7433e^{-2.034x}$, R2=0.9997

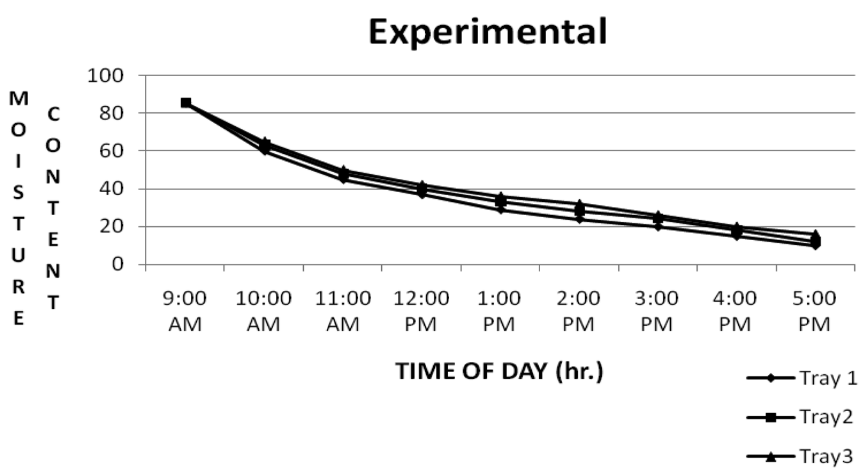

Figure 15. Hourly variation of moisture content in potato slices kept on tray1, tray2 and tray3 on 24/03/14

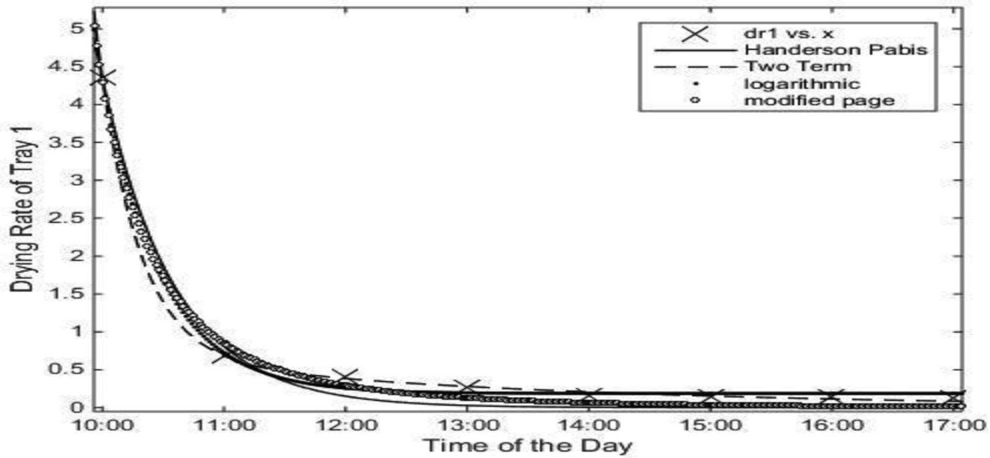

Figure 16. Hourly variation of drying rate with respect to time in hours for tray1 on 24/03/14

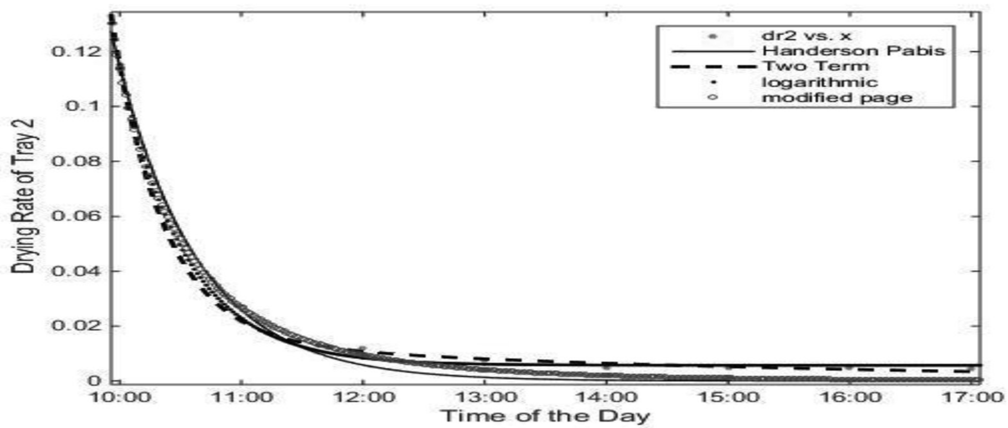

Figure 17. Hourly variation of drying rate with respect to time in hours for tray 2 on 24/03/14

Comparison of the drying rate of Tray1, Tray2 and Tray3 on the basis of four types of fitted curve models are listed on Table 7. It is obvious from the above results that the drying behavior of crops kept in all the three trays vary in the same manner. Thus, a two-term mathematical may be selected as the best fit curve to describe the drying behavior of crops very________________

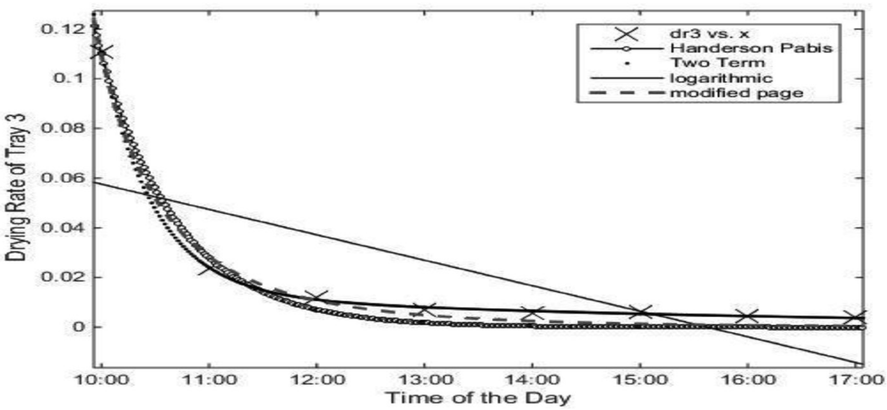

Figure 18. Hourly variation of drying rate with respect to time in hours for tray 3 on 24/03/14

CONCLUSION AND RECOMMENDATIONS

It is evident that over the last few decades there has been much demonstration for drying of agricultural food products using solar dryer having solar thermal energy storage in the form of sensible heat and mass transfer analysis has been done with heat Storage drying systems to investigate the performance and proved better than drying system without heat storage. Heat storage using 'phase change materials is a wise alternative. The main applications for phase change materials or PCMs are when space restrictions limit larger thermal storage units in solar drying systems. Solar energy holds the key to future's non-exhaustive energy source. Because of lack of time synchronicity between the energy supply and demand in solar heating applications, thermal energy storage (TES) device has to be used for the most effective utilization of the energy source. This concept of 'solar thermal energy storage using PCM in the solar dryer' reduces the time between energy supply and energy demand, thereby playing a vital role in energy conservation and improves the solar drying energy systems by smoothening the output and thus increasing the reliability for continuous drying of agricultural food products. This system can be considered as a reliable alternative to similar systems with conventional backup for medium scale drying of vegetables and fruits, which require a drying temperature of 50-60°C. It is also inferred that the solar air heater is the most important component of the indirect solar drying system. Improvement of the solar air heater would lead to better performance of the drying system. Therefore; more studies to investigate and improve the thermal performance of double pass flat, v-corrugated and finned plate solar air heaters is still of considerable interest. The latent storage media is preferable compared to the sensible store media to achieve nearly constant drying air temperature during the drying process. Furthermore, before using the drying systems on large scale, computer simulation models must be performed to simulate the short and long terms performance of the drying systems with and without the storage media to estimate the solar drying curves of the dried products and investigate the cost benefits of the solar drying of agricultural products.

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Table 5. Comparison of Drying rate of Tray1, Tray2 and Tray3 according to four types of functions or curve models

Sno.	Type of model	Function used	Values of coefficients				Goodness of fit:			
							SSE	R-squar	ted R-s	RMSE
1	henderson and pabis	f(x) = a*exp(b*x)	a =0.642	b = -1.674			0.0001696	0.9851	0.9827	0.00532
2	two term	f(x) = a*exp(b*x) + c*exp(d*x)	a =2.06	b = -3.006	c = 0.02535	d =-0.296	4.19E-06	0.9996	0.9994	0.00102
3	logaritmic	f(x) = a*exp(-b*x)+c	a =0.8933	b = 2.045	c=0.005192		3.13E-05	0.9973	0.9962	0.0025
4	modified page	f(x) = a*exp(-b*(x^n)	a =6.468	b =3.987	n=0.4878		0.0001182	0.9896	0.9855	0.00486

Table for comparison of Drying Rate for tray2

Sno.	Type of model	Function used	Values of coefficients				Goodness of fit:			
							SSE	R-squar	ted R-s	RMSE
1	henderson and pabis	f(x) = a*exp(b*x)	a =0.5092	b =-1.491			0.0001826	0.9821	0.9791	0.00552
2	two term	f(x) = a*exp(b*x) + c*exp(d*x)	a =0.0187	b =-0.213	c =1.038	d =-2.339	5.60E-06	0.9994	0.999	0.00118
3	logaritmic	f(x) = a*exp(-b*x)+c	a = 0.6972	b =1.852	c =0.005715		2.09E-05	0.9979	0.9971	0.00204
4	modified page	f(x) = a*exp(-b*(x^n)	a =8.145	b = 4.269	n =0.4215		0.0001104	0.9892	0.9848	0.0047

Table for comparison of Drying Rate for tray3

Sno.	Type of model	Function used	Values of coefficients				Goodness of fit:			
							SSE	R-squar	ted R-s	RMSE
1	henderson and pabis	f(x) = a*exp(b*x)	a =0.4344	b =-1.371			0.0001707	0.9818	0.9788	0.00533
2	two term	f(x) = a*exp(b*x) + c*exp(d*x)	a =0.0164	b =-0.1858	c =0.7433	d =-2.034	3.19E-06	0.9997	0.9994	0.00089
3	logaritmic	f(x) = a*exp(-b*x)+c	a =0.5733	b = 1.697	c =0.005692		1.55E-05	0.9984	0.9977	0.00176
4	modified page	f(x) = a*exp(-b*(x^n)	a =5.838	b =3.976	n =0.4216		9.40E-05	0.99	0.986	0.00434

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ABOUT THE AUTHORS

S.K. Shukla, corresponding author, is a professor in Mechanical Engineering Department, Indian Institute of Technology (BHU), Varanasi, India. He completed his Ph.D. from IIT Delhi. His areas of interest are thermal engineering, heat and mass transfer analysis in solar thermal systems and design of renewable energy systems, modeling etc. E-mail: shuskla@gmail.com.

A.K Srivastava is a faculty of SRMGPC Lucknow. He is pursuing a Ph.D. from Dr. A.P.J.A.K.T.U Lucknow formerly known as U.P.T.U Lucknow India. He has more than 20 years of teaching experience in the area of mechanical engineering with thermal engineering as specialization.