Analysis of an indirect air heater solar dryer with multiple PCM

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Abstract: This paper deals with designing an experimental model of an indirect solar dryer which uses phase change materials (PCMs) for thermal energy storage. Corrugated aluminium sheet is used as the absorber plate. Aluminium pipes of 3/4" OD are welded under the corrugated sheet to store the PCM. Here, multiple PCMs are used, one with a high melting point and the other one with a lower melting point for the purpose of improving efficiency. A single air pass model in which air moves over the absorber plate is fixed for the study. Air is heated in air heater section which also contains thermal energy storage. Part of the energy obtained in the air heater section is first used to heat and melt the PCM. This heat energy is stored in the PCM and remaining energy in the heated air moves to drying chamber in which the drying process takes place. When the sun's insolation reduces, discharging from the PCM take place. Thus, it reduces the fluctuation in the energy and thus provides a more continuous energy to the system. Glass wool is used as the insulation material. Different parameters for this air heater-dryer have been calculated.

Keywords: phase change material; PCM; solar energy; renewable energy; air heater.

Reference to this paper should be made as follows: Sreerag, T.S. and Jithish, K.S. (2016) 'Analysis of an indirect air heater solar dryer with multiple PCM', *Int. J. Sustainable Design*, Vol. 3, No. 1, pp.38–59.

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1 Introduction

Agriculture is a major source of employment and income in India. Agriculture offers great opportunities to stimulate economic growth. To capture these opportunities modifications of agricultural processing systems and application of sustainable energy technologies need to be incorporated. Solar drying is an excellent way to preserve food and contribute to a sustainable world. If solar drying of produce is widely implemented, significant savings to farmers would be achieved and these savings could help strengthen the economic situation as well as change the nutritional condition of numerous developing countries. As a result, considerable research and development activities have taken place to identify reliable and economically feasible alternative clean energy sources. The choices for the alternate energy sources are: energy from the sun, waves, wind and geothermal etc. but solar energy being the most promising for food drying.

Direct sun drying, one of the oldest techniques employed for processing agricultural and food products, has been traditionally practiced in India for drying agricultural products. Considerable savings can be made with the open sun drying since the source of energy is free and sustainable. However, this method of drying is extremely weather dependent and has to face the problems of contamination, infestation, microbial attacks, etc., thus affecting the product quality. Additionally, the drying time required for a given commodity is quite long and results in post-harvest losses.

Drying is an important post handling process of agricultural produce which can extend shelf life, improve quality, minimise losses during storage and lower transportation costs since most of the water is taken out from the product during this process. Drying under controlled conditions of temperature and humidity helps the agricultural products to dry reasonably rapidly to safe moisture content and to ensure superior quality of the product. Controlled drying is practiced mostly in industrial drying processes which use large quantities of fossil fuels. The potential of using solar energy in the agricultural sector has increased due to fluctuation in the price of fossil fuel, environmental concerns and expected depletion of conventional fossil fuels.

Sodha and Chandra (1994) have observed that India receives an enormous amount of solar energy: on average, in order of 5 kWh/m^2day for over 300 days/year. Bala and Woods (1995) have observed that the natural convection solar dryer suffers from the limitations due to extremely low buoyancy induced airflow inside the dryers. Forced convection solar dryers (or active solar dryers) are suitable for larger amounts of material. They use either a direct absorption system through transparent covers or a system connected to solar collectors using indirect solar heat.

Harringshaw (1997) has mentioned that the food scientist have found that by reducing the moisture content of food by a process of heat induced drying to between 10% and 20%, bacteria, yeast, moulds and enzymes are all prevented from spoiling it, since microorganisms are effectively killed when the internal temperature of food reaches 140°F.

Scalin (1997) has mentioned that the flavour and most of the food nutritional value of dried food is preserved and concentrated.

Thermal drying, which is most commonly used for drying agricultural products, involves vaporisation of moisture within the product by heat and subsequent diffusion of vapour out of the product. Thus, thermal drying involves simultaneous heat and mass transfer (Ekechukwu, 1999).

Goswami et al. (2000) observed that the thermal performance of the corrugated solar air heaters have a significantly superior thermal performance to that of the flat-plate one, with the achievable efficiencies up to 60.3%. The use of selective coatings on the absorbing plates of all the solar heaters considered can substantially enhance their thermal performances and therefore its use is strongly recommended in practical applications, whereas such a selected coating is not recommended for the underside of the plates nor the glass covers.

According to Oguntola et al. (2010), solar dried food are quality products that can be stored for extended periods, easily transported at less cost while still providing excellent nutritive values. Thus solar drying in a weather proof enclosive is a better alternative to natural sun drying and artificial mechanical drying. Also solar dryers generate higher temperatures and lower relative humidity than natural solar drying.

Belessiotis and Delyannis (2011) have given descriptions on various direct and indirect solar dryers.

Although several researchers have conducted lots of research on solar driers, the use of an indirect solar dryer which uses two or more phase change materials (PCMs) for thermal energy storage has not been addressed. The proposed work addresses this issue for agricultural applications.

2 Description of the solar dryer

Drying is one of the oldest methods of food preservation. In India, agriculture is a major source of employment, income and foreign exchange. It offers great opportunities to stimulate economic growth. Capitalising on these opportunities requires modification of agricultural processing systems and application of sustainable energy technologies. Solar radiation, in the form of solar thermal energy, is an alternative source of energy for drying especially to dry fruits, vegetables, agricultural grains and other kinds of material, as wood, etc. This procedure is especially applicable in the so-called ‘sunny belt’ worldwide, i.e., in regions where the intensity of solar radiation is high and sunshine duration long. It is estimated that in developing countries there exist significant post-harvest losses of agricultural products, due to lack of other preservation means that can be saved by using solar dryers.

Open-air solar drying uses solar radiation to heat directly the material. It is a process used for millennia to preserve food, a natural convection drying procedure, as the air movement is due to density differences. Direct solar drying has some disadvantages concerning both quality and quantity due to losses, attacks by insects, etc., thus in recent years direct sun drying is replaced by mechanical dryers heated indirectly by solar energy.

Indirect solar drying is a rather new technique, not yet standardised or widely commercialised, that involves some thermal energy collecting devices and dryers of using special techniques. There exist several types of dryers size, the construction technique of which fulfil the special drying requirements of food products, many of which still operate rather based on experience than on scientific basis.

3 Problem formulation and underlying concepts

The intermittent nature of the solar energy, which is the main source of energy in solar drying, is indeed one of the major shortcomings of the solar drying system. This can be alleviated by storing excess energy during the peak insulation time and use it in off sun hours or when the energy availability is inadequate. Today different types of energy storage systems have been designed. Sensible heat storage systems are designed and used. But it is found that latent heat storage system is more efficient in energy storage density as there is a phase change taking place in the system involving significantly more energy than that associated with temperature change alone.

When the temperature reaches the melting point of the PCM, it starts to melt and energy is stored. And on discharging when the temperature goes below the melting point of PCM, it starts to solidify and hence emitting energy for a reasonable time. It will act as a thermal flywheel which reduces the fluctuation in heat energy.

In this paper, in order to improve the energy release time during discharge cycles, a multiple PCM system is proposed. By using a high temperature PCM at inlet side and a low temperature PCM at outlet end during charging energy supply time can be improved. The mount of PCM to be used is calculated. It was decided as the amount of material which can be melted in one hour with the available useful energy. Different PCMs used are Paraffin wax and Glauber's salt. The scope of the current work includes developing an experimental model of solar air heater-dryer with multiple PCM storage system. Corrugated aluminium sheet is used as the absorber plate in this model. PCMs are kept inside aluminium tubes which were welded under the absorber plate. A glass covering was fitted on the top of absorber plate in order to reduce the losses. An air gap of 2 cm is maintained between absorber plate and glass cover. To avoid losses insulation was provided by glass wool for both the air heater section and the drying chamber.

4 Atmospheric condition of the test region

This solar dryer is to be tested for the climatic conditions of Kochi, Kerala State, India. Kochi is located on the southwest coast of India at 9°58'N, 76°17'E. Kochi features a tropical monsoon climate. Kochi's proximity to the equator along with its coastal location results in little seasonal temperature variation, with moderate to high levels of humidity. Annual average temperatures range between 25 and 37°C with the record high being 38°C (100°F). From June to September, the south-west monsoon brings in heavy rains as Kochi lies on the windward side of the Western Ghats. From October to December, Kochi receives lighter (yet significant) rain from the northeast monsoon, as it lies on the leeward side. Average annual rainfall is 3,228.3 mm (127.10 in), with an annual average of 132 rainy days.

The average monthly solar insolation received in Kerala is shown in Table 1. It can be observed that state receives adequate amount of solar insolation for the thermal applications.

Table 1 Average insolation for different months in Kerala

Month	*Average insolation (kWh/sq.m/d)*
January	5.80
February	6.46
March	6.83
April	6.24
May	5.57
June	4.83
July	4.91
August	5.26
September	5.74
October	5.24
November	4.94
December	5.30

Source: Hegde and Ramachandra (2012)

5 Design of experimental model

5.1 Tilt angle of air heater

The optimum tilt of solar collector is an important subject from application of thermal/electrical energy point of view. By utilising maximum solar energy through the optimum tilt, the energy needed without polluting our environment can be harnessed.

5.1.1 Angle of tilt of the solar collector (β)

$$\beta = 10 + lat\ ø \tag{1}$$

where *lat ø* is the latitude of the place that the drier is designed.

$$\text{Latitude of Kochi} = 9,058\text{'N}$$
$$\therefore \beta = 10 + 9,058\text{'} = 19.58°$$

Angle of tilt of the solar collector is taken as 20°.

5.2 Mass flow rate and air velocity

There are three major factors affecting food drying: temperature, humidity and air flow. They are interactive. Increasing the vent area by opening vent covers will decrease the temperature and increase the air flow, without having a great effect on the relative humidity of the inlet air. In general more air flow is desired in the early stages of drying to remove free water or water around the cells and on the surface. Reducing the vent area by partially closing the vent covers will increase the temperature and decrease the relative humidity of the inlet air and the air flow.

5.2.1 Air velocity (v)

Volumetric flow rate is given by,

$$V_a = v \cdot H \cdot W \tag{2}$$

where

v velocity of air flow (m/s)

H height of the vent (m) = 2 cm = 0.02 m

W width of air vent (m) = 60 cm= 0.6 m

V_a volumetric flow rate of air (m^3/s).

Also, mass flow rate,

$$m_a = V_a \cdot \rho_a \tag{3}$$

where

ρ_a density of air(kg/m^3) = 1.28 kg/m^3.

Making the simplifying assumption that all the radiation that is absorbed by the absorber plate is then transported away by the air gives:

$$\tau I_c \alpha A_c = m_a c_{pa} \Delta T \tag{4}$$

where

τ Transmittance of glass = 0.92.

α Absorptivity of absorber plate = 0.81.

C_{pa} Specific heat capacity of air = 1,005 J/kg.K.

ΔT Temperature difference between inlet and outlet. Assume ΔT as 35°C.

A_c Area of the collector (m^2). Collector area is fixed as 1 m × 0.6 m.

I_c Total solar insulation received on the glass cover

m_a Mass flow of air over the absorber plate.

5.2.2 Solar insolation

Solar insolation is given by Olaloye (2008) as

$$Ic = HI \cdot R \tag{5}$$

where

HI Insolation received on horizontal surface.

R average effective ratio of solar energy on tilt surface to that on the horizontal surface = 1.0035.

Taking average insolation to be 750 W/m^2

$$Ic = 750 \times 1.0035 = 752.62 \text{ W/m}^2$$

and mass flow rate

$$m_a = A_c I_c \tau \alpha c_{pa} \Delta T$$

$$\therefore \text{ mass flow rate} = (0.6 \times 752.62 \times 0.7452) / (1{,}005 \times 35) = 9.56 \times 10^{-3} \text{ kg/s}$$

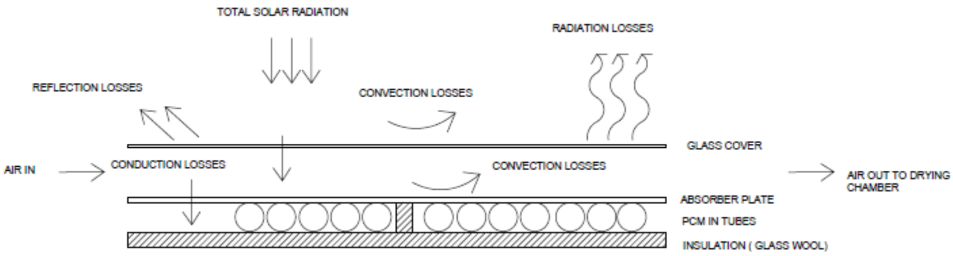
$$\text{Volumetric air flow rate} = 7.47 \times 10^{-3} \text{ m}^3/\text{s}$$

$\therefore$ velocity of air needed for this air flow rate = 0.62 m/s.

5.3 Energy balance of the absorber

The Figure 1 shows different losses from the air heater section from the glass plate like convection losses, reflection losses, radiation losses and conduction losses.

Figure 1 Energy balance of the air heater assuming no heat flow into the storage



The total heat energy gained by the collector's absorber is equal to the heat lost by heat absorber of the collector (Bukola and Ayoola, 2008).

Writing energy balance of the glass cover,

$$I_c A_c = Qu + Qcond + Qn.conv + Qrad + Qp \tag{6}$$

where

I_c	rate of total radiation incident on the absorber's surface W/m^2
A_c	area of the collector (m^2)
Qu	rate of useful energy collected by air (W)
$Qcond$	rate of conduction losses through the glass (W)
$Qn.conv$	rate of natural convection losses from the glass surface (W)
$Qrad$	rate of long wave re-radiation (W)
Qp	rate of reflection losses from the absorber (W).

If τ is the transmittance of the top glazing, total solar radiation incident on the absorber is given as

$$
\begin{aligned}
I_c \cdot A_c &= \tau \cdot I_c \cdot A_c \\
&= 0.88 \times 752.62 \times 0.6 \\
&= 397.38 \text{ W}
\end{aligned}
\tag{7}
$$

The reflected energy from the absorber is given by

$$Qp = \rho \cdot \tau \cdot I_c \cdot A_c$$

where ρ = reflection coefficient of the absorber. It is 4% for the glass.

$$Qp = 0.04 \times 0.88 \times 752.62 \times 0.6 = 15.89 \text{ W}$$

In order to calculate the convective and radiation losses the temperature of glass plate and ambient air are measured on an existing system. It is found that for glass temperature (Tg) of 64°C the ambient temperature (Ta) is 30.4°C.

$$\text{Convection losses from glass cover} = Ac \cdot hc \cdot \Delta T \tag{8}$$

where hc = heat transfer coefficient.

To find the value of hc, air properties are to be taken at mean temperature.

$$
\begin{aligned}
Tmean &= (64 + 30.4)/2 \\
&= 47.2°C
\end{aligned}
$$

- kinematic viscosity v =18.22 × 10^{-6} m^2/s
- air density at Kochi = 1.14 kg/m^3
- C_p = 1.0063 kJ/kg°C
- temperature difference ΔT = 33.6°C.

$$\text{Grashof Number} = L3g\beta\Delta T/v^2 \tag{9}$$

where

l length of the absorber plate = 1 m

g constant of gravity

$$\beta = \frac{1}{T} \tag{10}$$

v kinematic viscosity.

Substituting values,

$$Gr = 2.07 \times 10^{10}$$

Also,

$$\text{Prandtl number } \Pr = (\mu Cp)/k \tag{11}$$

where

μ is the dynamic viscosity. It is given by ρv

where

ρ density of air kg/m^3

v kinematic viscosity of air m^2/s

C_p specific heat of air at constant pressure kJ/kg°C

k conductivity coefficient for atmospheric air = 0.027 W/m°C

Substituting values,

$$\text{Prandtl number} = 9.5\times10^{-4}$$

Therefore,

$$\begin{aligned} Gr\cdot Pr &= 2.075\times10^{10}\times9.5\times10^{-4} \\ &= 1.9\times10^{6} \end{aligned}$$

For the range of Gr · Pr; $10^5 < \text{Gr}\cdot\text{Pr} < 2\times10^7$

$$h = 1.32\times(\Delta T/L)^{1/4} \tag{12}$$

Substituting values

$$h = 3.17 \text{ W/m}^2\text{K}$$

Thus heat lost through convection from the top surface of glass plate in contact with atmospheric air is thus given by

$$Qconv = h\cdot Ac\cdot(Tg - Ta) \tag{13}$$

Substituting values,

Therefore, convective losses from top of glass plate $Qconv$ = 64.06 W.

Radiative losses is given equation

$$Qrad = Ac\cdot\varepsilon\cdot\sigma\left(Tg^4 - Ta^4\right) \tag{14}$$

where

ε emissivity of glass = 0.9

σ Stefan Boltzmann constant = 5.61×10^{-8} W/m^{2}K^4.

For Tg = 64°C and Ta = 30.4°C; Qrad = 135.467 W.

Conduction losses,

$$Qcond = -k\cdot Ac\cdot dT/dx \tag{15}$$

Conduction losses from the glass are neglected. Convection and radiative losses will be dominating in the glass surface.

Useful energy Qu, for an average value of Ic =752.62 W, Tg = 64°C, Ta = 30.4°C. So,

$$\begin{aligned} &397.38 = Qu + 64.06 + 135.46 + 15.89 \\ &\Rightarrow Qu = 181.97 \text{ W} \end{aligned}$$

This is the energy available through the glass. At bottom of the glass surface there is forced convection heat transfer due to the flowing air from the air blower.

Heat transfer coefficient *h* for forced convection is given as

$$h = 0.665(k/L)Re^{1/4}Pr^{1/3} \tag{16}$$

where

Re Reynold's number is given by the relation

$$Re = \frac{Lv}{v} \tag{17}$$

where

L length of surface = 1 m

v velocity of air flow = 0.6 m/s

v viscosity of air = 18.22 × 10^{-6} m^2/s.

$\therefore$ Reynold's number *Re* = 32,930.84.

Also,

$$Pr = 9.5 \times 10^{-4}$$

$\therefore$ heat transfer coefficient *h* = 0.508.

$\therefore$ heat lost through forced convection.

$$Qf.conv = Ac \cdot h \cdot \Delta T \tag{18}$$

Substituting values,

$\therefore$ *Qf.conv* = 10.241 W

Now.

$$\text{Available energy after theforced convective losses} = 181.97 - 10.241 = 171.73 \text{ W}$$

Now as the air from blower passes over the absorber sheet, air will absorb energy = *Qu*. But there will be some losses before this energy is transferred to energy storage material (PCM). There will be conduction losses, radiative losses and some reflective losses from the corrugated aluminium absorber plate. For design purpose all these losses are assumed to be together 10% of the available energy through the glass.

$$10\% \text{ of the available energy through the glass} \Rightarrow 0.10 \times 171.73 = 17.17 \text{ W}$$

$$\text{Actual useful energy } Qu = 171.3 - 17.17$$

- *Qu* = 154.56 W, so average useful energy can be taken as *Qu* = 154.56 W.

5.4 Volume calculation of PCMs

In current work, two PCMs are needed to be placed. One is with higher melting point than the first one. Assume that the useful energy received after considering all the losses will be equally divided to the PCMs kept under the absorber plate. Therefore each PCM will receive 0.0772 kW-hr energy. When the PCM melts there will be both sensible heat storage and latent heat storage taking place. To calculate the heat storage per unit volume, assume that the temperature is raised in a range of 10°C. That is from a temperature of 5 degree lower than the melting point to a temperature of 5 degree above the melting point of PCM.

Sensible heat stored per unit volume

$$\frac{Q}{V} = \rho_s c_{ps} \left(T_{sat} - T_1 \right) + \rho_s C_{pl} \left(T_{sat} - T_2 \right) \tag{19}$$

The latent heat storage is given as

$$\frac{Q}{V} = \rho_s L \tag{20}$$

where

T_1 temperature of heat transfer fluid entering the absorber (K)

T_2 temperature of heat transfer fluid leaving the absorber (K)

V volume of the material

ρ_s density in solid state

L latent heat of fusion

C_p specific heat.

Among the two PCMs, one is selected from the paraffin and other was selected from salt hydrates. Paraffin wax consists of a mixture of mostly straight chain-alkanes CH3-(CH2)-CH3. The crystallisation of the (CH3)-chain release a large amount of latent heat. Both the melting point and latent heat of fusion increase with chain length. Paraffin qualifies as heat of fusion storage materials due to their availability in a large temperature range. Due to cost consideration, however, only technical grade paraffins may be used as PCMs in latent heat storage systems. Paraffin is safe, reliable, predictable, less expensive and non-corrosive. They are chemically inert and stable below 5,008°C, show little volume changes on melting and have low vapour pressure in the melt form. For these properties of the paraffins, system-using paraffins usually have very long freeze-melt cycle. Table 2 lists thermal properties of some technical grade paraffins, which are essentially, paraffin mixtures and are not completely refined oil. The melting point of alkane increases with the increasing number of carbon atoms. Apart from some several favourable characteristic of paraffins, such as congruent melting and good nucleating properties. They show some undesirable properties such as:

1 low thermal conductivity

2 non-compatible with the plastic container

3 moderately flammable.

All these undesirable effects can be partly eliminated by slightly modifying the wax and the storage unit. Some selected paraffins are shown in Table 3 along-with their melting point, latent heat of fusion and groups. PCMs are categorised as:

1 group 1, most promising
2 group 2, promising
3 group 3, less promising.

Table 2 Melting point and latent heat of fusion of paraffins

No. of carbon atoms	*Melting point (°C)*	*latent heat of fusion(Kj/Kg)*	*Group*
14	5.5	228	1
15	10	205	2
16	16.7	237.1	1
17	21.7	213	2
18	28	244	1
19	32	222	2
20	36.7	246	1
21	40.2	200	2
22	44	249	2
23	47.5	232	2
24	50.6	255	2
25	49.4	238	2
26	56.3	256	2
27	58.8	236	2
28	61.6	253	2
29	63.4	240	2
30	65.4	251	2
31	68	242	2
32	69.5	170	2
33	73.9	268	2
34	75.9	269	2

Source: Sharma et al. (2009)

Table 3 Melting point and latent heat of fusion of salt hydrates

Material	*Melting point (°C)*	*Latent heat (Kj/Kg)*	*Group*
$K_2HPO_46H_2O$	14.0	109	II
$FeBr_36H_2O$	21.0	105	II
$Mn(NO_3)_26H_2O$	25.5	148	II
$CaCl_212H_2O$	29.8	174	I
$LiNO_32H_2O$	30.0	29.6	I
$LiNO_33H_2O$	30	189	I
$Na_2CO_310H_2O$	32.0	267	II
$Na_2SO_410H_2O$	32.4	241	II
$KFe(SO_4)_212H_2O$	33	173	I
$CaBr_26H_2O$	34	138	II

Source: Sharma et al. (2009)

The Tables 2 and 3 show properties of some commonly used PCMs. In current design, there are two PCMs. One is with higher melting point than the first one. So accordingly latent heat of fusion of both varies. Paraffin wax with a melting temperature of 36.7°C and L = 246 kJ/kg is chosen as high temperature PCM. $Na_2SO_4.10H_2O$ which is commonly known as Glauber's salt with melting temperature 32.4°C and L = 241 kJ/kg is taken as low temperature PCM.

Consider the temperature range of Glauber's salt as 27°C to 37°C. Its properties are given as

- melting temperature = 32.4°C
- latent heat of fusion = 241 kJ/kg
- specific heat = 1,926 J/kg°C (solid) and 2,846 J/kg°C (liquid)
- density = 1,600 kg/m^3
- sensible heat stored per unit volume = 10.4 kW-hr/m^3
- latent heat stored per unit volume = 107.11 kW-hr/m^3
- the total storage = 117.5 kW-hr/m^3.

Volume required to store 0.0772 kW-hr energy is

$$V = 0.0772/117.5$$
$$= 6.5 \times 10^{-4}\ \text{m}^3$$

$$\text{Density} = \text{Mass (kg)}/\text{Volume (m}^3\text{)}$$

$$\text{Density of Glauber's salt} = 1,600\ \text{kg/m}^3$$

$$\text{Mass of Glauber's salt} = 6.5 \times 10^{-4}\ \text{m}^3 \times 1,600\ \text{kg/m}^3$$
$$= 1.05\ \text{kg.}$$

Consider the temperature range of Paraffin wax as 31°C to 41°C. Its properties are given as:

- melting temperature = 36.7°C
- latent heat of fusion = L = 246 kJ/kg
- specific heat = 2.0 kJ/kg°C (solid) and 2.15 kJ/kg°C (liquid)
- density = 860 kg/m^3
- sensible heat stored per unit volume = 4.93 kW-hr/m^3
- latent heat stored per unit volume = 58.76 kW-hr/m^3
- the total storage = 63.69 kW-hr/m^3.

Volume required to store 0.0772 kW-hr energy is

$$V = 0.0772/63.69$$
$$= 1.2 \times 10^{-3}\ \text{m}^3$$

Therefore mass of paraffin wax = 1.04 kg

Let us take aluminium tubes of diameter 3/4” and length = 0.6 m to hold PCM.

$$\text{The volume of tube} = \pi \cdot r^2 \cdot l \tag{21}$$

where

l length of tube

r radius of the tube.

$$\text{volume of one tube} = 1.65 \times 10^{-4}$$

Number of tubes needed for Paraffin wax = $1.2 \times 10^{-3}/1.65 \times 10^{-4}$ = 7.20 tubes, i.e., approximately 7.5 tubes (4.5 m in length).

Similarly, number of tubes for Glauber’s salt = 3.93 tubes, i.e., approximately 4 tubes (2.4 m in length).

5.5 Drying chamber

The optimum size of drying chamber is very important for the proper drying rate. Approximately 3 cubic feet of drying volume or 3 square feet of glazing for every 1 cubic feet of drying chamber is necessary for the optimum drying rate.

$$\text{Area of glazing} = 1\text{ m} \times 0.6\text{ m} = 0.6\text{ m}^2$$

Therefore volume of drying chamber = 0.6/3 = 0.2 m^3

Therefore dimensions are taken as 0.5 m × 0.5 m × 0.5 m.

6 Fabricated model of solar dryer

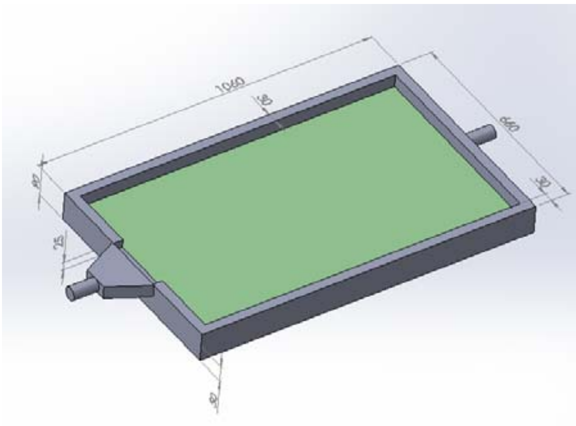
Experimental model of solar air heater-dryer is fabricated after the completion of design as shown in Figure 2. Body of the air heater and dryer are constructed using mild steel sheet. Full body is constructed as a double frame structure.

Figure 2 Fabrication of experimental setup (see online version for colours)

Figure 3 shows the fabricated drying chamber. The dimensions of air heater section are 1 m × 0.6 m × 0.09 m. The gap between two sheets is taken as 0.03 m is shown in Figure 4. This gap is filled with insulation material. Insulation is given on all sides with glass wool. Horizontally corrugated sheet coated with black paint is taken as absorber plate. 0.05 m thick glass wool is kept under the absorber plate in order to prevent the heat loss from the absorber plate. Aluminium tubes, sealed at both ends are used to place PCMs.

Figure 3 Drying chamber (see online version for colours)

Figure 4 Drawing of air heater section (see online version for colours)



A blower of 0.373 kW is used to circulate air. A regulator is used to vary the speed to attain desired air velocity. Glass covering for the air heater section is given with a clear glass of thickness 4 mm. K type (Chromel-Alumel) thermocouple wires are chosen owing to its high sensitivity at a reasonable cost as shown in Figure 5. Voltage is measured with a multi-meter and it is converted to corresponding temperatures using a chart for K-type thermocouple.

Figure 5 K-type thermocouple wires and tubes for PCM under the corrugated absorber sheet (see online version for colours)

7 Results and discussions

The experimental model of solar air heater-dryer is tested with green banana. After peeling of the skin from the banana, it is cut into small pieces of 3 mm thickness. Each day, approximately about one kg of banana is tested. Experiment is conducted from 8 am to 8 pm. Measurements are taken from 9 am onwards. This is done in order to make the model into steady state conditions. Experiments are done during the months of February and March 2014. Experiments are conducted at Govt. Model Engineering College, Kochi. Latitude of the place is 9°58'N.The developed experimental model of dryer is shown in Figure 6.

Figure 6 Completed experimental setup (see online version for colours)

During the experiment temperatures at inlet of air heater, outlet of air heater, three points at absorber plate, glass cover, drying chamber inlet, drying chamber outlet and tray temperatures were measured. Also relative humidity at inlet of air heater, outlet of air heater, outlet of drying chamber is also measured. Relative humidity is calculated by measuring dry bulb and wet bulb temperatures. Velocity of drying air is measured with a probe anemometer. A pyranometer is used to measure the solar radiation.

7.1 Tabulation and analysis

a Efficiency of collector η_c,

$$\eta_c = \frac{m_a c_{pa} \Delta T}{A_c I_c} \tag{22}$$

where

m_a mass flow rate of air

$m_a = V \times \rho_a$

where

ρ_a density of air(kg/m^3) = 1.28 kg/m^3

V volumetric flow rate (m^3/s).

Speed of circulated air in the collector = 0.5 m/s.

ΔT rise in temperature as air passes from inlet to outlet of air heater section

C_{pa} specific heat capacity of air at constant pressure

A_c effective area of the collector

I_c solar radiation.

b Moisture content (Senger, 2009) on percentage wet basis is expressed as follows:

$Mc_{wb} = (W_1 - W_2)/W_1$

where

W_1 weight of sample before drying in kg

W_2 weight of sample after drying inn kg.

c Moisture content on percentage dry basis is expressed as follows:

$Mc_{wb} = (W_1 - W_2)/W_2$

d Drying rate of sample during drying

$$Rd = (Mi - Md)/t \tag{23}$$

where

Mi initial mass of the sample in kg

Md final mass of the dried product in kg.

t drying time in hours.

e Dryer energy utilisation ratio is given as

$$EUR = (m_a \cdot h_{di} - m_a \cdot h_{do})/mC_p(T_o - T_i) \tag{24}$$

where

Enthalpy $h = C_p \cdot t + w \cdot h_{sat}$ (25)

where

w specific humidity,

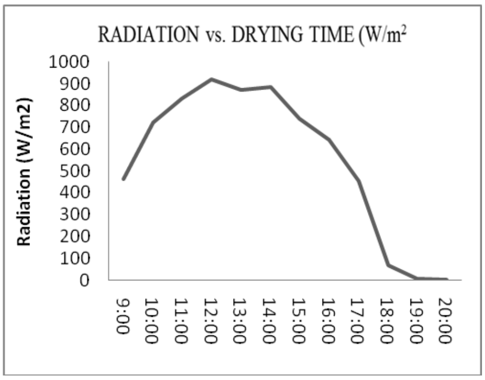
h_{sat} enthalpy of saturated vapour.

PCMs are incorporated into the system. It is kept in aluminium tubes which after sealing are placed under the absorber plate. Thus heat from the absorber plate is transferred to these tubes and thus into PCMs. The glass wool insulation which is kept under the absorber plate prevents the heat loss from these tubes. Total of 12 tubes are kept under absorber plate. Glauber's salt and Paraffin wax are taken as latent heat thermal energy storage systems. The readings noted for 20 March 2014 is shown in Table 4.

Table 4 Hourly temperature variation for green banana on 20 March 2014

Sl no.	*Time*	*Radiation (W/m^2)*	*Air heater inlet (°C)*	*Air heater outlet (°C)*	*Absorber plate (°C)*	*Tray temp (°C)*	*Drying chamber (°C)*	*PCM 1 (°C)*	*PCM 2 (°C)*
1	09:00	462	33.4	44.3	60.9	32.4	32.4	49.9	58.4
2	10:00	720	32.4	48.2	64.9	44	44.1	52.4	62.4
3	11:00	832	35.7	54	72.9	45.6	44.7	60.7	69.7
4	12:00	920	34	51.6	78.5	46.2	44.3	64	74.4
5	13:00	873	34.6	51.4	86.6	44.3	41.5	67.1	78.2
6	14:00	884	37	53	77	43.7	43.6	64.2	69.8
7	15:00	738	33.7	46.3	70.9	42.1	42.4	50.7	58.6
8	16:00	644	34.2	44.4	65.45	40.3	40	45	46.7
9	17:00	455	34.4	44.4	48.15	38.8	38.7	36	38
10	18:00	68	32.3	41.2	43.55	34.2	32.9	34.8	35
11	19:00	4	31.9	35.5	39.5	32.9	29.8	31.5	31.9
12	20:00	0	29.3	32	37	31.6	27.8	31	31.1

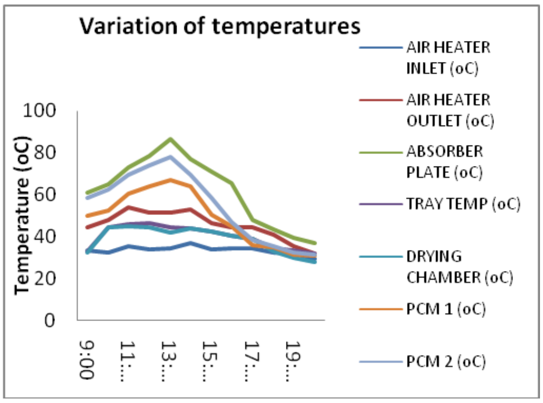
Figure 7 Variation of radiation with drying time



Variation of radiation with drying time is shown in Figure 7. The solar radiation varies from above 462 W/m^2 in the morning to above 920 W/m^2 at noon and then it decreases down to less than 4 W/m^2 at 7:00 PM and to 0 W/m^2 at the end of experiment.

The changes in radiation with drying time, temperature variation at different points with drying time is shown in Figure 8.

Figure 8 Variation of temperatures (see online version for colours)



The air heater inlet temperature, i.e., ambient temperature is the minimum. Absorber plate is having maximum temperatures. Heat from the absorber plate is transferred to incoming air and the air heater outlet temperature rises. The tray temperature shows the air temperatures inside the drying chamber. It is shown in Figure 8.

Variation in relative humidity is measured by noticing dry bulb and wet bulb temperatures at corresponding points. RH is measured at inlet of collector, outlet of collector and outlet of drying chamber. The humidity at collector's outlet is taken equal to the humidity at the inlet of drying chamber. Variation of humidity with drying time is shown in Table 5.

Table 5 Relative Humidity variation with drying time

Sl no.	*Time*	*Inlet relative humidity*	*Outlet relative humidity*	*Chamber outlet relative humidity*
1	09:00	75.4	48.5	59.2
2	10:00	76.8	28.4	51.3
3	11:00	58.5	46.3	56.5
4	12:00	69.7	37.8	65.4
5	13:00	64.1	43.6	54.6
6	14:00	59.9	21.8	40.5
7	15:00	66.3	25.9	40
8	16:00	63.3	30.6	41.5
9	17:00	57.1	37.1	40.4
10	18:00	66.1	49.2	57.8
11	19:00	68.8	61.8	65.5
12	20:00	69.4	67.4	68.4

Figure 9 shows the variation of relative humidity with drying time. From the figure it is clear how the humidity is varying at different points. Also, the inlet or ambient relative humidity is the maximum relative humidity present. As the air passes over the heated absorber plate, it absorbs sensible energy and thus relative humidity decreases. This high temperature air then heat the bananas raising the vapour pressure in the food such that it is higher than in the air and hence the vapour moves from the bananas to the air. From the graph it is very clear that the outlet relative humidity from the collector is the minimum. Now as this hot air passes over the specimen, it absorbs moisture from it and thus the moisture content increases. Thus the chamber outlet RH will be greater than the collector outlet relative humidity.

Figure 9 Variation of relative humidity with drying time (see online version for colours)

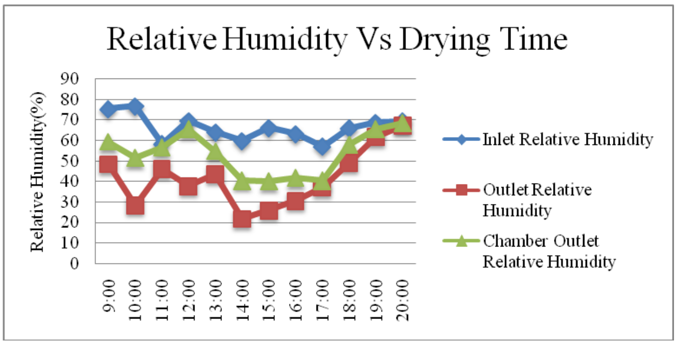


Table 6 Results of ten days experiments with PCM

Sl no.	*Day*	*Initial mass (kg)*	*Final mass (kg)*	*Temp rise (°C)*	*Drying rate (kg/hr)*	*Drying time (hrs)*	*Dryer energy utilisation ratio (%)*	*Collector efficiency (%)*
1	14-3-14	1.0	0.395	14.5	0.055	11	51.6	39.02
2	17-3-14	1.0	0.398	14.2	0.054	11	49.7	38.17
3	18-3-14	1.03	0.400	14.2	0.057	11	49.5	37.76
4	19-3-14	1.02	0.390	14.6	0.057	11	51.6	39.24
5	20-3-14	1.0	0.385	16.8	0.055	11	52.1	44.10
6	21-3-14	1.0	0.380	17.4	0.056	11	53.2	45.89
7	22-3-14	1.05	0.405	13.5	0.058	11	48.4	36.93
8	25-3-14	1.0	0.402	13.6	0.054	11	48.8	36.34
9	26-3-14	1.06	0.380	16.6	0.061	11	52.1	44.25
10	27-3-14	1.02	0.388	14.6	0.057	11	51.7	39.01

The experiments are conducted for a period of ten days and specimen is taken as green banana. The approximate weight of specimen to be dried for a day is taken as 1 kg. It is weighed after a drying period of 11 hours.

Drying rate, collector efficiency, etc. are calculated and tabulated. It is found that drying rate has improved. Also the final mass of the specimen have come below that of system without PCM. It shows an improvement is achieved with the new system.

8 Conclusions and scope of future work

An experimental model of indirect solar air heater-dryer with thermal energy storage was designed and fabricated. The model was fabricated as a double frame structure with mild steel sheets. It is insulated from all the sides by using glass wool.

The following are the main observations. It is found that in system without PCMs, the temperature in the evening hours falls rapidly. The rate of drying hence is low compared to the designed system. In the system with PCMs installed, the air temperatures at elevated levels can be maintained for the evening hours. This will keep the process of drying continuous. It is observed that the drying rate for the advanced system is an improved one. Thus the energy absorbed into the PCM during the early stages is given back in the later stages. Also in the process of drying, the amount of water removed in early stages will be low as the transportation of moisture from inside to outside of crop take place in different steps. But as soon the moisture removal start to take place in large quantity, if continuous heat is supplied the drying will take place in a steady and fast rate. Thus upon using system with PCM, more constant rate of heat energy can be supplied at evening hours, which have helped in making the specimen dry faster.

PCMs used in the experimental model are not of commercial grade. So after a number of cycles of experiment it will lose the quality. This can be reduced by using commercial grade PCMs. Wide range of PCMs are available with a large range of melting temperatures. System can be improved by experimenting with different types of PCMs. This will make system performance optimised.

Also from literature survey it is clear that increasing number of PCMs will improve system performance. So increasing collector area and experimenting with more number of PCMs may improve overall efficiency.

Heat transfer from PCMs to air is always low. So implementing a system to improve heat transfer to air can be considered. This experimental model is a single pass system. If number of passes is increased, system is expected to improve its performance.

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