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Design of Novel Indirect Solar Dryer with PCM Heat Storage Unit

Yash Hareshbhai Beladiya

Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad, Gujarat, India

Email: yashbeladiya@gmail.com

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Abstract:

Drying is the most effective and widely used preservation technique. In this paper, freely available solar energy is used to dry perishable foods by designing a solar dryer. An evacuated tube type solar heat collector is used to collect heat energy. It gives clean drying, a high temperature, and efficiency compared to a flat plate collector. The system and its components are designed by considering grapes as a drying product. The initial moisture content in grapes varies between 77-82%, and the final moisture content in grapes should be between 15-19%. To use it in off-sunshine hours, a latent heat storage unit is designed. Acetamide is used as a phase change material, which can store approximately 10500 kJ of heat energy in 28 kg of material, this unit can store thermal energy during the daytime and would be used during the night hours. Water is used as heat transfer fluid, which transfers heat energy from collector to heat storage unit cum heat exchanger, and forced air is passed through that unit, which heats the air which is transferred to the drying chamber, which is made up of plywood and can dry 5 to 6 kg of grapes in a lot. All calculations are carried out by considering the climatic condition of Ahmedabad, a city in India. This calculation can be used to fabricate an efficient solar dryer for drying grapes.

Keywords — **Phase Change Material, Solar Energy, Solar Dryer, ETC, Heat Storage.**

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I. INTRODUCTION

Drying is the most efficient and oldest approach to food processing. Improper processing of perishable foods and inadequate storage facilities are responsible for almost one-third of the produced food waste throughout the world. Drying is the process of removal of moisture due to simultaneous heat and mass transfer. It is a classic method of food preservation that provides longer shelf life, small space for required storage, and lighter weight for transportation. The quality of dried food is affected by the method of preparation, the state of fresh food, and drying conditions[1].

The open sun drying is most common used traditional drying techniques. In this process substance is kept in open area where direct sun rays are coming. This Drying process has advantages like simplicity and less initial investment. Draw backs of this drying process are large area requirement, high labour cost, large time required for the drying process, poor quality due to contamination by birds, dust and insects, uncertainty in weather conditions and impact on natural colour of food being dry[2].

The solar drying process is a very good alternative to open sun drying. Farmers have developed tent and box dryers using locally available materials for drying crops, fruits, and vegetables. They have used the transparent cover to reduce heat loss and give protection from dust and rain. These types of solar dryers are usually referred to as Direct solar dryers as they directly utilize solar radiation. The direct solar dryer has advantages like being comparatively inexpensive, it gives protection to the substance from dust and rain. A direct Solar dryer is mostly used in on-farm drying. Direct Solar dryers have certain disadvantages like changing colour, vitamins, and nutrients and loss of dried material because of direct exposure to solar radiation. Due to moisture compression in the glass cover transmissivity may be reduced. A direct solar dryer required a long duration for drying[3].

In an Indirect type Solar dryer, first heat is collected in the solar collector, then it is transferred to the drying chamber via heat transfer fluid. The drying process occurs in a separate drying chamber. The heated air removes the moisture from the product and goes away via the chimney. The product is kept in the drying chamber in certain numbers of perforated trays. There are two types of solar collectors used in the case of indirect type solar dryers. 1) Evacuated tube collector. 2) flat plate collector. Both collectors are commonly used. An evacuated tube solar collector has higher efficiency than a flat-plate collector. Indirect type solar dryer has certain advantages like no colour loss, no vitamins and nutrients loss, and lesser drying time required. an indirect solar dryer can provide a better quality of drying compared to other dryers as drying happens in the controlled environment of the chamber. Indirect solar collector has required more initial investment than other types of the solar dryer[3].

Ubale et al. [4] have designed, fabricated and experimentally tested an evacuated tube solar collector in the forced convection mode of heat transfer. The grapes they have taken had initial moisture content of 77% wet basis to final moisture content of 19% wet basis in 36 hours with a moisture removal rate of 0.21 kg/h. They obtained a maximum collector outlet temperature of 82.3 °C with a mass flow rate of air of 0.00817 kg/s and an average collector outlet temperature of 65.1°C. They have obtained 24.3% overall thermal efficiency. Umayal Sundari et al. [5] have designed and developed a solar dryer assisted with the evacuated tube collector to study and examine the drying kinetics of muscat grapes in Thanjavur, Tamil Nādu. Their designed dryer took 14 hours to reduce the moisture content of muscat grapes from 78% to 9.5% (wet basis). The maximum efficiency of the dryer was obtained at 29.92% during the drying period.

Rabha and Muthukumar [6] carried out performance studies on a forced convection solar dryer which is integrated with a paraffin-wax based shell and tube latent heat thermal storage unit. The solar dryer has been tested with 20 kg of red chilli in the drying air temperature range of around 36–60 °C. There was 73.5% initial moisture content present and it dried up to 9.7% final moisture content in 4 consecutive days. The overall efficiency of the drying system was 10.8% and the specific energy consumption of the dried chilli was 6.8 kWh/kg of moisture. Jain and Tewari [7] have developed a solar crop dryer with a thermal energy storage medium. The solar crop dryer consists of a flat plate solar collector, a drying plenum with crop trays, packed bed phase change energy storage, and a natural ventilation system. The dryer is attached to thermal energy storage having a capacity of 50 kg PCM. The highest temperature in FPC was observed at about 95°C. They obtained an efficiency of the dryer of about 28.2%.

Shalaby and Bek [8] have experimentally investigated a novel indirect solar dryer with phase change material as an energy storage medium. Their system consists of a drying chamber, two identical solar air heaters, a PCM storage unit and a blower. They have performed an experiment on both with and without Phase change material at a wide range of mass flow rates (0.0066-0.2182 kg/s). Two plastic cylinders filled with PCM (paraffin wax) were used as heat storage. Subramanian et al. [9] have designed and fabricated a solar dryer based on an evacuated tube solar collector with and without heat storage material (gravel is used as heat storage material). Designed dryers have reduced the moisture content of chilly from 87.36% to 3.4% in 10 hours in case of heat storage material used.

Shringi et al. [10] have developed a solar dryer system and evaluated the performance of the thin layer drying characteristic of the garlic clove. They have dried garlic cloves from initial moisture content of 55.5% (wet basis) to final moisture content of 6.5% (wet basis) in 8 hours. The energy efficiency with and without recirculation of the air exiting the drying chamber during the study varied from 3.98 to 14.95% and 43.06 to 83.73%, respectively.

II. SYSTEM COMPONENTS OF THE NOVEL SOLAR DRYER

Components of PCM based solar dryer incorporated with ETC given in figure 1.

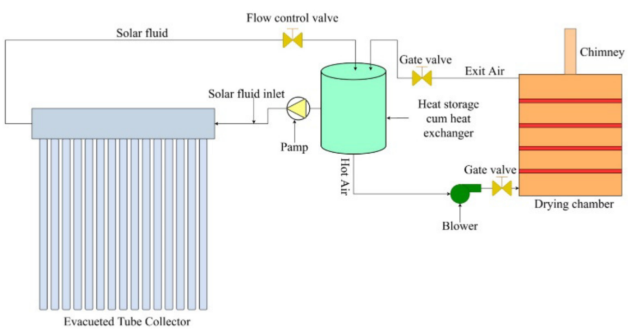

Fig. 1 Setup of Novel ETC based Solar Dryer with Heat Storage

A. *Evacuated Tube Solar Collector (ETC)*

Evacuated tube collector is selected to collect the heat energy from solar radiation and convert it into the thermal energy. Performances are affected by various parameters like tilt angle, climate conditions, collector dimension, etc. [11] Specification of the selected ETC for the design of the novel dryer given in the table.

Table 1 Specifications of evacuated tube collector

Parameters	Specifications
No of tubes	15 unit
Tube outer diameter	58 mm
Length of the tube	1800mm
Thickness of glass	1.6 mm
Gross area	2.34 m^2
Opening area	1.566 m^2
Type of heat pipe	R8(14/24) *1800
Collector content	0.638 L
Distance between pipes	75 mm
Direction of tubes	vertical
Connection diameter	22 mm
Maximum working pressure	8 bars
Glass tube Material	Borosilicate
Maximum stagnation temperature	245 °C
Heat pipe material	copper

B. *Phase Change Material (PCM)*

Phase Change Material (PCM) is used for thermal management solution because it is reserve and liberate thermal energy during melting and freezing[12]. We have selected 'Acetamide' as PCM for this dryer based on its various characteristics (given in the table) which suits requirement of the system.

Table 2 Characteristics of Acetamide [13]

Name of PCM	Acetamide
Melting Temperature	79-81 °C
Latent heat of Fusion	243-263 kJ/kg
Density (solid)	1159 kg/m^3
Density (liquid)	998 kg/m^3
Specific Heat	2.173 kJ/kg °C
Thermal conductivity	0.5 W/ m K

C. *PCM Heat Storage Unit cum Heat Exchanger*

It is Latent heat storage unit which also work as heat exchanger between heat source and air. Charging and discharging of PCM depend on heat transfer rate. Latent heat storage gives heat energy at constant temperature which is required for drying. PCM heat storage is useful for energy conservation [14].

PCM heat storage cum heat exchanger contains acetamide as phase change material, water as heat transfer fluid and air which is used for drying purpose. Water and air are passing through two different pipes in heat exchanger. Heated water comes from ETC to heat exchanger. Water gives its heat energy to PCM and air. When temperature of PCM reaches to the melting point, PCM is converted from solid to liquid. Now PCM stores latent heat which can be use in off sunshine hour and heated air directly used for drying during day. PCM releases its heat at night and this heat energy used for heating the air.

D. *Drying Chamber*

Drying chamber is used to remove moisture content. It is one type of dehumidifier. Heated air comes from blower to chamber. At initial drying stage moisture removal rate is high so exit air which has high moisture from drying chamber allowed dumping in atmosphere via chimney. At later drying stage moisture removal rate is decrease so exit air which has low moisture. This air is transferred to heat exchanger from chamber.

E. *Pump and Blower*

Pump is provided at outlet of heat exchanger. Pump is required to accommodate the pressure drop water in heat exchanger. Pump also regulates the flow of water.

Blower is provided at inlet of drying chamber. When air is passing through heat exchanger, pressure of air at outlet of heat exchanger is reduced due to pressure drop so that blower is required to accommodate the pressure drop.

F. *Gate Valve and Flow Control Valve*

Flow rate control valve is provided at the inlet of the heat exchanger to regulate of mass flow rate of water.

One gate valve is provided at the inlet of the drying chamber. This gate valve regulates the mass flow rate of air. Second gate valve is provided at the outlet of the drying chamber. This gate valve is closed at initial stages of drying so that air is passing through chimney. At later stages this gate valve is opened so that air can pass through heat exchanger.

III. DESIGN CALCULATION OF DRYER SYSTEM

A. *Heat Required for Drying Grapes*

We have selected grapes for our experiment as drying product. Initial moisture available in grapes is 77% and final moisture needed in dried grapes is around 19% [15].

Amount of moisture to be removed from the product, m_o:

$$m_o = m_w \frac{m_i - m_f}{100 - m_f}$$

For 1kg of initial mass of grapes:

$$m_o = 1 \cdot \frac{77 - 19}{100 - 19} = 0.716\ kg$$

The quantity of heat required to evaporate water was determined based on below equation

$$Q = m_o \times h_{fg}$$

Latent Heat of Vaporization,

$$h_{fg} = 4.186(597 - 0.46 \times T_p)$$

Where T_p = Temperature of the product in °C

Q = 1739 kJ (If Product temperature is assumed to ambient temperature)

B. *Solar Irradiance and ETC*

The experimental setup is located at Nirma University, Ahmedabad, Gujarat, India (23.1284° N, 72.5449° E). Average solar irradiance for particular month in Ahmedabad shown in figure 2:

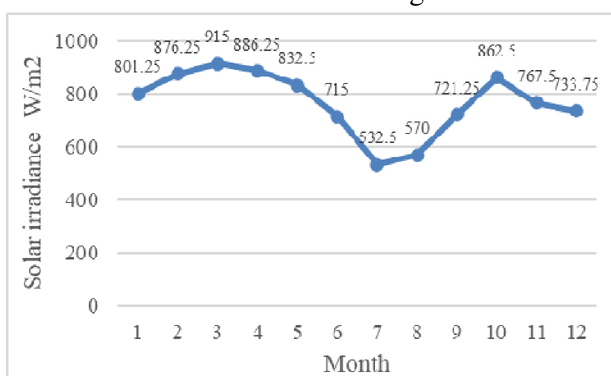

Fig. 2 Solar Irradiance at Ahmedabad

Total Solar Energy Absorbed during sunshine is given by equation (1):

Total Solar Energy Absorbed during sunshine

$$G = \int_0^t (\alpha\tau) G_s A_s \tag{1}$$

G = 35862 kJ/day

Efficiency of Solar collector is assumed is 67%

Efficiency of collector, η = 67%

So that energy transfers to HTF

Q = 0.67xG

Q = 24027.4 kJ/day

Florida Solar energy centre of USA (FSEC Solar collector test report no. 97005, May 1998) has conducted test with the performance test reports by Technikon Rappers will [16].

Linear fit:

$$\eta = 0.82 - 2.192\left(\frac{T_m - T_a}{G_s}\right)$$

Second order fit:

$$\eta = 0.82 - 1.23\left(\frac{T_m - T_a}{G_s}\right) - 0.0122G_s\left(\frac{T_m - T_a}{G_s}\right)^2$$

Solar Collector efficiency depends on tilt angle, Climate condition of location, absorber area, mass flow rate of HTF, temperature gradient of HTF and Solar insolation, etc. efficiency mainly depends on mass flow rate of HTF and $\left(\frac{T_m - T_a}{G_s}\right)$.

Accounting average solar insolation for all month and average ambient temperature we have plotted graph of average temperature of HTF get from ETC. Mean temperature of HTF as shown in figure 3 is different for different month but overall, it comes above 85° C.

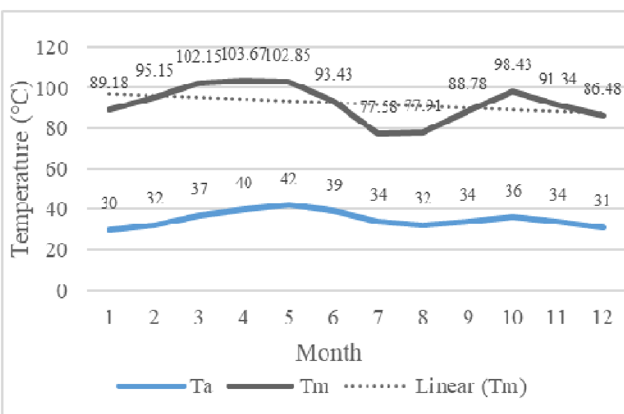

Fig. 3 Mean temperature of HTF at outlet of ETC

C. PCM Storage cum Heat Exchanger Design

Latent heat storage unit is tube in tank type heat exchanger. Method selected for the designing the heat exchanger is effectiveness – NTU method[17].

The mathematical illustration of the heat transfer is a one-dimensional formulation between the HTF and the PCM at the phase change profile. The NTU is determined from the thermal resistance to heat transfer in the HTF, the tube wall, and the segment of the PCM which has undergone phase change. This assumption is based on experiment of internal temperature measurement in the PCM [18]

Assumption:

- The inlet velocity and inlet temperature of HTF are constant.
- Outer wall of tank is remaining adiabatic.
- The initial temperature of latent heat storage unit is uniform.
- Initially PCM is in solid phase for melting or in liquid phase for freezing.
- The model is axisymmetric.
- The thermophysical properties of HTF, tube wall and PCM are constant and taken at constant mean film temperature which is 45° C.
- The natural convention in liquid phase of PCM has been ignored.

NTU at any point of time represented by:

$$NTU = \frac{(UA)}{(\dot{m}C_p)} = \frac{1}{(R_T \dot{m}\, C_p)} \qquad (2)$$

For formulate the effectiveness the total thermal resistance (R_T) needs to be determine. The thermal resistance is function of overall heat transfer coefficient between HTF and PCM and the heat transfer area.

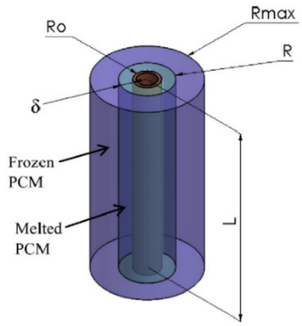

Fig. 4 Simplified model of the round shape factor [19]

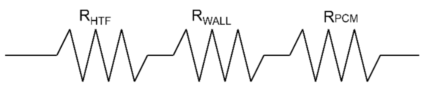

Fig. 5 Thermal circuit of HTF to PCM

1) *HTF side analysis*

As shown in above figure 4 the long tube in which HTF from ETC is passed is surrounded by volume of PCM, the total thermal resistance (R_T) is given by equation (3).

$$R_T = R_{HTF} + R_{WALL} + R_{PCM} \quad (3)$$

Mass flow rate of HTF ($\dot{m}$) is 0.035 m^3/s which is constant.

Specific heat of water (C_p) is 4197 J/kg K
From TEMA standard diameter of HTF tube
Outer diameter of tube is 12.7 mm
Inner diameter of tube is 11.28 mm

Effectiveness varies over the phase change process but for safer design we assumed minimum Effectiveness (ε_{HTF}) at HTF side analysis is 27% at maximum phase change of 88%.
Effectiveness (ε_{HTF}) is given by equation (4):

$$\varepsilon_{HTF} = 1 - exp\left(\frac{-1}{\dot{m}C_p R_T}\right) \quad (4)$$

Putting the value of mass flow rate and effectiveness in equation (4), we get the value of total thermal resistance which is: R_T = 0.02165 K/W

$$R_T = R_{HTF} + R_{WALL} + R_{PCM}$$

$$R_T = \left(\frac{1}{2R_i h_f L_f}\right) + \left(\frac{\ln\left(\frac{Ro}{Ri}\right)}{2\pi k_w L}\right) + \left(\frac{\ln\left(\frac{R}{Ro}\right)}{2\pi L_f k_{PCM}}\right) \quad (5)$$

L_f is length of HTF tube, the arrangement is assumed to be single length tube. The effect of bends is ignored, as it represents 20-25% of tube length, but it was included in total tube length [17].

Calculation of Heat transfer coefficient (h_f):

Dynamic viscosity of HTF (μ_f) is 3.55 Pa·s

$$\text{Prandtl number}, P_r = \left(\frac{\mu_f C_p}{k_f}\right) = 5.22$$

Where k_f is the Thermal conductivity of HTF = 0.67 W/ m K

Reynolds number;

$$R_e = \frac{\dot{m}D_i}{A_c \mu_f} = 1.11 \times 10^4$$

Turbulent flow for, Re ≥ 10000;

$$0.7 \leq P_r \leq 160$$

Dittus–Boelter equation for turbulent flow

$$Nu = 0.023Re^{0.8}Pr^{0.3}$$

$$Nu = 50.5$$

Heat transfer coefficient,

$$h_f = \frac{Nuk_f}{2R_i} = 3 \times 10^3 \ W/m^2K$$

Length of HTF Pipe, L_f = 19.15 m

From equation (5),

$$R_{HTF} = \left(\frac{1}{2R_i h_f L_f}\right) = 4.92 \times 10^{-4} \ K/W$$

$$R_{WALL} = \left(\frac{\ln\left(\frac{R_o}{R_i}\right)}{2\pi k_w L}\right) = 2.32 \times 10^{-6} \ K/W$$

Tube is made of copper, Thermal conductivity of copper, k_w = 396 W/m K

From equation (3),

$R_{PCM} = 2.11 \times 10^{-2}$ K/W

Now,

$$R_{PCM} = \left(\frac{\ln\left(\frac{R}{R_o}\right)}{2\pi L_f k_{PCM}}\right) = 2.11 \times 10^{-2} \ K/W$$

From solving above equation, we get changing radius of the PCM during phase change (R)

$$R = 2.22 \times 10^{-2}\ m$$

R_{max} is the shape factor calculated based on compactness factor of thermal storage system. The shape factor can be very with time, as time is independent variable shape factor can be calculated from the phase change fraction δ. The phase change fraction is proportion of PCM that has yet to change its phase. Phase change fraction (δ) directly relates to solid to liquid line within the PCM when phase change occurs [17].

The Phase change fraction δ for circular tube by cylindrical volume of PCM is given by,

$$\delta = \frac{R^2 - R_o^2}{R_{max}^2 - R_o^2} \quad (6)$$

We can assume that maximum phase change occurs 88%, so we get maximus radius of phase change (R_{max})

$$R_{max} = 2.38 \times 10^{-2}\ m$$

The compactness factor (CF) is the ratio of the volume of PCM to the volume of the tank which given by following equation (7).

$$CF = \frac{R_{max}^2 - R_o^2}{R_{max}^2} = 93\% \quad (7)$$

2) *Tube side Analysis of Air*

Dimensions of air tube

Length of air tube (L_{air}) is 12.45 m

From TEMA standard diameter of tube:

Outer diameter of tube (D_o) is 22.2 mm

Inner diameter of tube (D_i) is 18.93 mm

Tube material is copper

Mass flow rate of air is 0.004 kg/s (within permissible limit)

Specific heat of air (C_p) is 1007kJ/kg K

Thermal conductivity of air (k_{air}) is 0.027 W/m K

Dynamic viscosity of air (μ_{air}) is 1.941×10^{-5} Pa·s

We got changing radius of PCM phase change (R) from tube side analysis of HTF, that radius is used here for getting the effectiveness of air tube heat exchange from PCM to air so here we do same calculation but in reverse. Thermal circuit for same shown below:

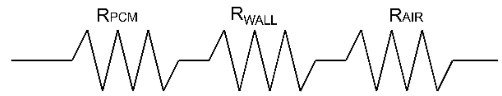

Fig. 6 Thermal circuit of heat transfer PCM to Air

Resistance due to PCM is R_{PCM}

$$R_{PCM} = \left(\frac{\ln\left(\frac{R}{R_o}\right)}{2\pi L_{air} k_{PCM}} \right) = 3.43 \times 10^{-4}\ K/W$$

Resistance due to wall of air tube (R_{air}) is given by:

$$R_{WALL} - \left(\frac{\ln\left(\frac{R_o}{R_i}\right)}{2\pi k_w L} \right) - 5.20 \times 10^{-6}\ K/W$$

Reynold number of air,

$$R_e = \left(\frac{\dot{m} D_i}{A_c \mu_{air}} \right) = 1.44 \times 10^4$$

$$\text{Prandtl number}, P_r = \left(\frac{\mu_f C_p}{k_f} \right) = 0.7241$$

Turbulent flow for, Re ≥ 10000; 0.7≤ Pr ≤ 160

Dittus–Boelter equation for turbulent flow

$$Nu = 0.023 Re^{0.8} Pr^{0.3}$$

$$Nu = 41.6$$

Heat transfer coefficient (h_{air})

$$h_{air} = \frac{Nu k_{air}}{2R_i} = 60.1\ W/m^2 K$$

Resistance due to air (R_{air}) given is by:

$$R_{AIR} = \left(\frac{1}{2R_i h_{air} L_{air}} \right) = 2.25 \times 10^{-2} \frac{K}{W}$$

The total thermal resistance (R_T) is given by

$$R_T = R_{AIR} + R_{WALL} + R_{PCM} = 0.0228\ K/W$$

Putting the all-available value in effectiveness equation:

$$\varepsilon_{AIR} = 1 - exp\left(\frac{-1}{\dot{m} C_p R_T} \right) = 99\%$$

Final effectiveness of heat exchanger is

$$\varepsilon = \varepsilon_{HTF} * \varepsilon_{AIR} = 0.27 * 0.99 = 0.268$$

26.8 % is the overall effectiveness at 88% phase change is this ration is change then effectiveness also changes.

3) *Heat exchanger tank*

Table 3 Specification of Heat Storage Tank

Parameter	Value
Material	Stainless Steel 304S
Outer Diameter	310 mm
Height of tank	540 mm
Thickness of tank	5 mm
Total volume of Heat exchanger tank	0.0374 m^3
Total volume occupied by tubes	7.21×10^{-3} m^3
Volume available for PCM	0.03 m^3
Mass of PCM	29.94 ~ 30 kg

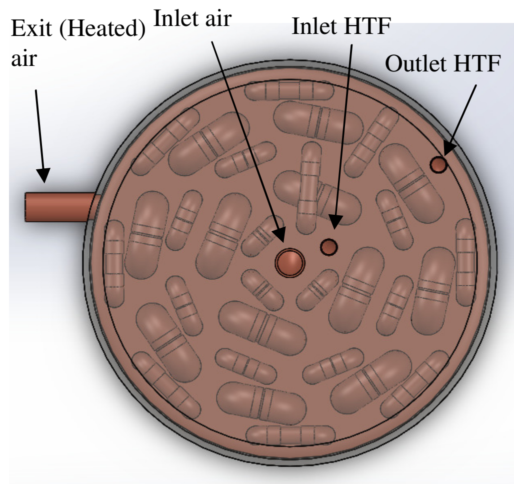

Fig. 7 Top view of PCM Storage unit

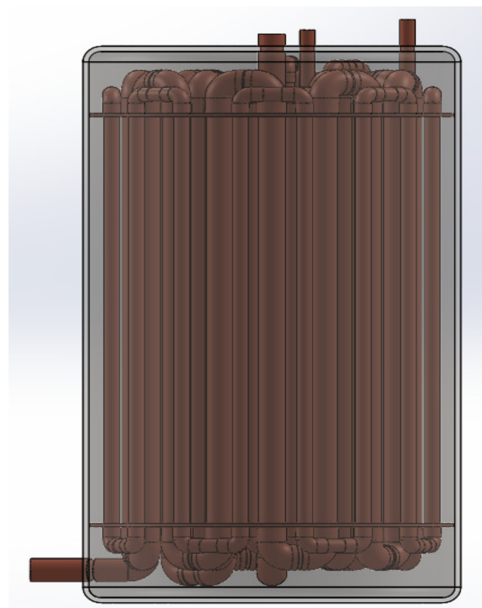

Fig. 8 Side view of PCM Storage Unit

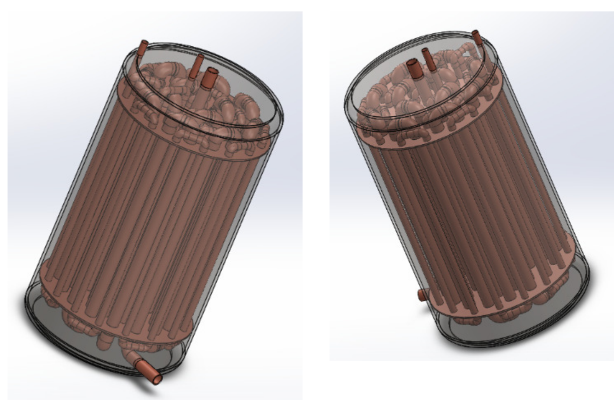

Fig. 9 Solid Model of PCM Storage Unit

Insulation is used to prevent thermal losses from the latent heat storage unit. Fibreglass material is suitable as an insulating material with thermal conductivity of 0.038 W/m K.

4) Pressure drop calculation

Pressure drop in HTF pipe and air pipe can be calculated by putting the appropriate value in the below equation.

We have selected EXCESS HEAD (K) method or also known as resistance coefficient, velocity head, or crane method [20].

K value for standard 90° elbow is 0.75

The head loss for single elbow;

$$h_L - \frac{KV^2}{2g} \tag{8}$$

For pipe loss:

$$h_p = f\frac{L}{D}\frac{V^2}{2g} \tag{9}$$

Total Head loss, $h_T = h_p + n \times h_L$

Where, n is number of elbows.

Power required to compensate the resulting head losses (10).

Power,

$$P = \frac{\dot{Q}(\rho g h_T)}{\eta} \tag{10}$$

D. Design of Drying Chamber

Material use for drying chamber is plywood. Dimensions of drying chamber is taken based on various research worked carried out in past. Plywood has thermal conductivity 0.13 W/m K use for construction of drying chamber, chimney and other supporting structure. Stainless steel wire mesh is use for construction of product holding tray [21]. There are four removable trays for product holding. 10 cm space is kept between each tray [22]. An air-tight wooden door was made with rubber seals at the edges and hinges as well as wooden locks. There is air inlet at the bottom of chamber and outlet at the top of chamber, outlet is use in second stage of drying where less humid air in circulation.

Table 4 Specification of Drying Chamber

Parameter	**Value**
Material of Drying chamber:	Plywood
Material for tray	Stainless steel
Dimensions for Chamber (Inside)	450×450×580 mm
Thickness for chamber wall	12 mm
Drying chamber capacity	118 litres
No of tray	4
Dimensions of perforated tray:	390×390×2
Distance between each tray	100 mm
Chimney	80×80×500 mm

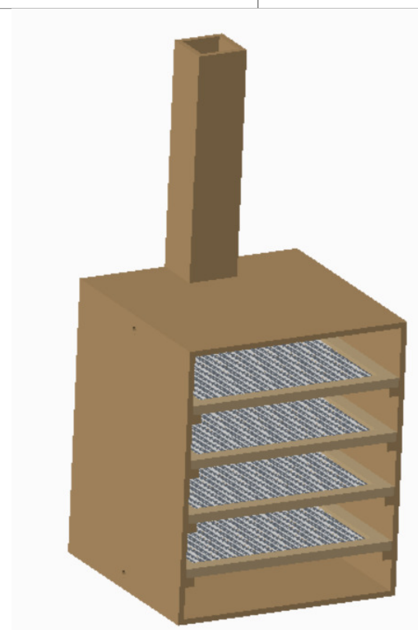

Fig. 10 Solid Model of Drying Chamber

IV. CONCLUSION

By following the above procedure, any solar dryer with ETC and heat storage mechanism can be designed. The designed novel solar dryer can dry up to 5 to 6 kg of grapes in a day. The capacity of the dryer changes with the change in drying product and change in weather phenomena like high sunshine, seasonal change, cloudiness etc.

Based on the given design, a Solar dryer can be fabricated. The novel Dryer continues the drying process in the off-sunshine hours of the day. The continued process has a good effect on food that is being dry as it will not regain moisture from its surroundings at night time when the process is stopped. It gives clean and high-quality drying in a minimum time duration.

Nomenclature

α	Absorptivity of collector
δ	phase change fraction
ε	heat exchanger effectiveness
ε_{HTF}	heat exchanger effectiveness HTF tube side
ε_{PCM}	heat exchanger effectiveness Air tube side
η	Efficiency
τ	Transmissivity of collector
A	heat transfer area (m^2)
A_s	Solar collector Area (m^2)
A_c	cross sectional area of the inner tube (m^2)
CF	compactness factor
C_p	specific heat of the HTF (kJ/ (kg K))
ΔT	temperature difference (K)
D	diameter of the tank (m)
d	inner tube diameter (m)
ρ_{PCM}	density of the PCM (kg/m^3)
ρ_{HTF}	density of the HTF (kg/m^3)
f	friction factor
G_s	Average solar insolation (W/m^2)
h_f	heat transfer coefficient of the HTF (W/ (m^2 K))
h_{fa}	Latent Heat of Vaporization (kJ/kg)
h_{air}	heat transfer coefficient of the Air (W/ (m^2 K))
h_L	head loss in elbow (m)
h_P	head loss in pipe (m)
h_T	total head loss (m)
k_w	thermal conductivity of the tube wall (W/ (m K))
k_{PCM}	thermal conductivity of the PCM (W/ (m K))
L	length of the tube (m)
L_f	length of the HTF tube (m)
L_{air}	length of the Air tube (m)
μ_f	dynamic viscosity of the HTF (kg/ (m s))
μ_{air}	dynamic viscosity of the Air (kg/ (m s))
m_i	initial moisture content in % in wet basis in dryer product
m_f	final moisture content in % in wet basis in dryer product
m_o	Amount of moisture to be removed from the product (kg)
$\dot{m}$	mass flow rate of HTF (kg/s)
Nu	Nusselt number
Pr	Prandtl number
ΔP	pressure drop of the system (kPa)
Q	Amount of Energy required for drying Process (kJ)
Q	volume flow rate (m^3/s)
R_T	total thermal resistance (K/W)
R_{HTF}	thermal resistance of the HTF (K/W)
R_{WALL}	thermal resistance of the tube wall (K/W)
R_{AIR}	thermal resistance of the Air (K/W)
R_{PCM}	thermal resistance of the PCM (K/W)
R_i	inner radius of the tube (m)
R_o	outer radius of the tube (m)
R	changing radius of the PCM during phase change (m)
R_{max}	radius of PCM when point of intersection with neighbouring phase change front (m)
Re	Reynolds number
V	velocity of fluid (m/s)
U	overall heat-transfer coefficient (W/ (m^2 K))

Abbreviation

HTF	Heat Transfer Fluid
ETC	Evacuated Tube Collector
NTU	Number of Transfer Units
PCM	Phase Change Material

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