Chapter 1  INTRODUCTION

Drying, also known as dehydration is defined as the process of reducing moisture content of a product. It is the most common method employed to preserve food for long durations (Doymaz 2007). The moisture level is reduced to a level below which micro-organisms cannot grow, hence the shelf life of the food is extended (Chkir et al 2015). Drying also brings other benefits. Dried product is much lesser in weight which helps in reducing material handling cost as it takes less transportation power and less storage space (Guine 2008). It becomes easier to carry food to remote areas which is one of the preferred choice for military. Various agricultural products such as grapes, chillies, turmeric, amla, tea, cocoa beans and dry fruits undergo drying process. It is estimated that 20% of the world's fruit and vegetable production is subjected to dehydration (Ramirez et al 2015).

Drying is an energy intensive operation. It accounts nearly 15% of the all the industrial energy utilisation (Colak and Hepbasli 2009). The process involves abstraction of water from the food product by supplying heat and usage of moving air to take away the vapours (Ekechukwu and Norton 1999). Traditionally it has been accomplished by burning wood or fossil fuels and under open sun. In the former method, heat produced during the combustion process is used to evaporate the water content. In open sun drying, the product is spread in a thin layer under the sun which provides the necessary energy for the required process (Togrul 2003). These days electric dryers are becoming prominent in industries. Nevertheless these can still be considered to be equivalent of burning fossils as most of the electricity is produced using coal. While burning fuels is expensive and damages the environment, open sun drying is time consuming and susceptible to contaminations (VijayaVenkataRaman et al 2012). Solar Crop Dryers have been developed which tend to overcome the shortcomings of open sun drying, permitting usage of clean energy.

A simple solar crop dryer consists of a cabinet covered with a glass in which product to be dried is kept. Greenhouse effect due to glass raises dryer temperature much higher than ambient temperature and thus faster drying is achieved (El-Sebaii and Shalaby 2012). Further, crop is protected from the invasion of insects and birds as it is covered. Several designs for solar dryers have been developed. These days indirect crop dryers are more prevalent in which crop is shaded from direct sunlight and instead, air heated by passing through solar collectors is used to raise the temperature of the crop (Sharma et al 2009). On basis of mode of operation, solar dryers are of two types- passive and active. Passive dryers depend on natural movement of air. These are inexpensive and easy to construct. Active solar dryers incorporate fans or pumps for the circulation of air. These are characterised with faster drying operation due to enhanced heat transfer coefficient at high mass flow rates but at the expense of increased input energy (Esakkimuthu et al 2013).

The limitation of equipments based on solar energy is that they can operate only when adequate amount of solar radiation is available. This causes huge inconvenience on cloudy days
when the available energy is not enough. It also prohibits night operations which is one of the major requirements of the industries. This limitation can be overcome by storing the excess energy during peak sunshine hours and utilising the same during off sunshine. The heat energy is stored in a storage medium, typically in form of sensible or latent heat. In sensible heat storage, thermal energy is stored by raising the temperature of solid or liquid (Singh et al 2010). The amount of heat storage depends on specific heat of the medium, the temperature change and amount of storage material. Latent heat storage is the heat absorption when a storage material undergoes a change of phase at constant temperature. One common example is melting of ice. It is characterised with high heat storage capacity per unit volume and isothermal behaviour during charging and discharging (Bal et al 2011). But due to high costs associated with latent heat storage, it is overshadowed by sensible heat storage. When storage is integrated, not only operational time increases but also efficiency of the system increases (Shalaby et al 2014).

In a crop dryer, product is at higher temperature than ambient. This imposes limitation on maximum energy that can be transferred by hot air to the product. In perspective of thermodynamics, even if we consider best case scenario, heat transfer will only occur until air comes down to crop temperature. Rest of the energy which has been used to elevate the temperature of air remains ineffective. Air will leave drying chamber of a solar crop dryer comparatively at higher temperature than ambient air due to which some energy will always be lost. If this energy could be collected and utilised, the performance of the dryer would improve.

Although there have been lot of design improvements in solar crop dryers, information regarding heat in exit air and its utilisation is still in scanty. This research is aimed towards evaluating the losses due to heat in exit air of a conventional forced circulation solar dryer and test sensible thermal storage as an option for heat recovery. It is proposed to have thermal energy storage unit at exit of the drying chamber which will extract heat from leaving air. During off-sunshine hours, this collected heat can be utilised by making the air to flow in reverse direction from storage to drying chamber.

Thus the present research has been conducted with the following specific objectives:

1. To investigate effect of sensible heat storage at exit of drying chamber of a conventional forced circulation crop dryer on thermal efficiency.
2. To investigate effect of auxiliary heating coupled with sensible heat storage at exit of drying chamber of a conventional forced circulation crop dryer on thermal efficiency.
3. To develop a computer model for predicting drying rate curve using hot air from solar air heater of conventional forced circulation solar crop dryer with storage.

The first two objectives have been achieved by fabricating and experimentally testing the proposed system. Based on these experimental results, computer program has been developed for the same setup to predict the drying rate curve using hot air from solar air heater.
Chapter 2 REVIEW OF LITERATURE

Literature study comprises of three sections- theory of drying, solar crop dryers and thermal storage. First section introduces drying process of food materials along with influencing factors. Second section presents various designs and performance characteristics of solar crop dryers. Finally thermal energy storage is discussed which provides basis for designing energy recovery unit used in current research work.

2.1 Theory of Drying

Drying is a complicated process involving simultaneous heat and mass transfer under the transient conditions (Guine and Fernandes 2006). It is a two step process. First, the migration of water from inside to the surface of the product takes place. Second process is the evaporation of water from the wetted surface (Hollick 1999).

Biological materials are capillary-porous in nature. Several mechanisms act in different combinations during migration of moisture. The mechanisms include liquid diffusion due to concentration gradients, liquid transport due to capillary forces, vapour diffusion due to partial vapour pressure gradients, liquid or vapour transport due to the difference in total pressure caused by external pressure and temperature, evaporation and condensation effects, surface diffusion and liquid transport due to gravity (Guine 2005).

A typical drying proceeds in two phases-constant drying rate period and falling drying rate period (Duc et al 2011). Constant drying rate is observed until the material is able to provide enough water for constant evaporation. After that, diffusion limits the migration of moisture thereby pushing drying into falling rate period (Seth and Sarkar 2004). Many products do not exhibit constant drying rate period as they are unable to keep surface saturated with water (Belessiotis and Delyannis 2011).

Drying is influenced by various internal and external factors. Internal parameters encompass material's density, permeability, porosity, sorption-desorption characteristics and thermophysical properties. External factors are temperature of working air, mass flow rate of air and its relative humidity (Kaya et al 2007, Sevik 2013). Since the internal factors are material properties, they hardly play any role in process control. The operation is controlled by the external parameters. Selection of higher temperature accelerates the process (Panchariya et al 2002). Wang et al 2007 carried out hot air dehydration of apple pomace in thin layer manner at a fixed bed velocity. Drying time required to reach the equilibrium moisture content were 330, 285, 224 and 178 min at 75, 85, 95 and 105°C respectively. Increase in relative humidity not only slows down the drying process, but also escalates equilibrium moisture content (Khatchatourian 2012). Study conducted by Zlatanovic et al 2013 on apple cubes at different temperature and relative humidity
values confirm this effect. Higher mass flow rate or air velocity shortens the drying time (Doymaz 2004). However, this effect is significant only upto a certain level. Thereafter, air velocity has proportionally less role in reduction of drying time. It is due to the fact that external resistance to moisture transport becomes negligible as compared to internal resistance which becomes the governing mechanism. Higher temperatures are more effective in removing the remaining moisture instead of excessive mass flow rate (Sabarez 2012). Slice thickness also affects the drying time (Dissa et al 2008). Sacilik and Elicin 2006 determined thin layer drying characteristics of organic apple slices. Samples having slice thickness 5 and 9 mm were dried in temperature range of 40 to 60°C. In case of 5 mm thick slices, the drying time required to reach the final moisture content came out to be 400, 300 and 240 min at the drying air temperatures of 40, 50 and 60°C respectively. Corresponding values for the slice thickness of 9 mm were 640, 560 and 460 min.

Apart from process duration, drying parameters also affect the quality of the final product. Excessive drying temperature can cause case hardening and over burnings (Kadam and Samuel 2006). Nutrient losses are also high at elevated temperatures (Marfil et al 2008).

Pretreatments have been reported to reduce energy, time and damage. Verma and Gupta 2004 examined the effect of various pre-treatments on amla for solar drying. It was observed that amla with flaking treatment retained the maximum amount of ascorbic acid, that is 76%. While untreated sample could only retain 27%. Treatments significantly improved colour, taste, texture and flavour when compared to control sample. Drying time was also reduced by pre-treatment.

The energy requirements for carrying out dehydration of a product depend upon its initial moisture content, desired final moisture level and all of the above factors discussed (Karim and Hawlader 2005).

### 2.2 Solar Crop Dryers

Solar crop dryers can be organised into four categories- direct type, indirect type, mixed mode and hybrid dryers. In direct type solar dryers, material to be dried is placed in an enclosure having a transparent cover on it. Heat is generated by the absorption of solar radiation on the product itself as well as by internal surfaces of the dryer (El-Sebaii and Shalaby 2012).

Direct exposure to solar radiation sometimes deteriorates the product quality (Mohajer et al 2013). Grapes when dried under direct solar radiation are vulnerable to surface cracking. Sweet potatoes also need to be protected from sunlight in order to avoid discolouration in the resulting product (Jairaj et al 2009). This has led to the development of indirect type solar crop dryers. Direct type solar dryers also suffer from reduced transmissivity due to moisture condensation inside glass cover (El-Sebaii and Shalby 2012) which is completely eliminated in indirect type dryers. These solar dryers consist of a solar collector and a separate drying chamber with an opaque top. The crop is placed in drying chamber mostly in a thin bed which spans the cabinet. Air is heated in solar collector, then it passes through the drying chamber producing drying effect (Janjai
Sreekumar et al. (2008) developed an indirect type crop dryer which operates in active mode. The dryer basically consists of two parts – flat plate collector and a drying chamber located underneath the collector. The aperture area of the dryer is 1.27 m². Selectively coated aluminium absorber plate shades the crop from sunlight. Two axial fans are provided for forced circulation of air. The dryer was first tested without load and axial fans switched off. The maximum temperature of the absorber plate, drying chamber and ambient was 97°C, 78.1°C and 33.7°C respectively. When the fans were operated in no load condition, the maximum temperature of absorber plate and air flowing over the absorber plate was found to be 80.5°C and 56.6°C. This clearly indicates that more heat has been extracted from the absorber plate due to forced circulation of air. The average air velocity inside the dryer was 1.2 m/s. The dryer was tested for drying 4 kg of fresh bitter gourd. The moisture content was reduced from 95% to 5% after 6 hours of drying. While open sun drying for the same took 2 days with an effective drying time of 11 h. In open sun drying, the colour of the product was completely lost, however it was retained in case of indirect dryer. Economic analysis was also conducted for the developed dryer and annual savings for drying same crop came out to be Rs. 31659. Furthermore, the dryer has pretty short payback time of 3.26 years. The cost of drying bitter gourd in solar dryer was 42% less than that of electric dryer.

Mixed-mode solar crop dryer consists of a separate solar collector and a drying chamber, both having transparent cover on top. Solar radiation is received in collector as well as in drying chamber (Janjai and Bala 2012). Since the material to be dried receives heat in two ways, drying time is considerably reduced. ELkhadraoui et al. (2015) analysed mixed-mode greenhouse dryer for drying pepper and grapes. The system comprised of a flat plate collector and east-west oriented even span greenhouse fitted centrifugal fan for forced air circulation. The moisture content of red pepper was reduced to 16% (wet basis) in 17 hours in greenhouse dryer whereas open sun drying took 24 hours for the same. Moisture content of sultana grapes was reduced to 18% (wet basis) in 50 hours in case of greenhouse dryer whereas open sun drying took 76 hours. Payback period was found to be 1.6 years.

Hybrid solar dryers combine solar radiation energy with auxilary conventional source of energy. They can be operated either only by solar energy, only by conventional energy source or in combination, depending upon design and load requirements (Vidana et al 2013). The energy source could be electricity, biomass, CNG or some fossil fuel (Fudholi et al 2010).

Prasad and Vijay (2005) constructed and tested solar-biomass hybrid dryer working on natural convection. The dryer consists of single glazed inclined cabinet receiving direct solar radiation. Three perforated trays aligned vertically, each having 0.991 m² area hold the product. Vents are provided for circulation of air. Below the cabinet there is a biomass burner. Stove heats the bottom of the drying cabinet setting up natural draft as well as providing heat for drying. It is used when solar radiation is not available although it can also be used in conjuction with solar.
energy. Testing was done with drying ginger, turmeric and guduchi. In each case hybrid mode was significantly faster than solar mode. Thermal efficiencies in case of solar only mode ranged from 13.5-29.1% depending upon the product. While hybrid mode operated at 7.5-15.59% efficiency, biomass only mode mode could only reach 3.4-7.1%.

Limited ventilation in natural draft solar dryers sometimes gives birth to extremely high temperatures in the drying area which leads to partially cooked food instead of dried. To overcome this problem, chimney is used to enhance natural circulation. Chimney operates by increasing force to boost the airflow through a structure. This buoyancy force is directly proportional to the difference between density of air within the chimney and the outside the environment (Chen and Qu 2014).

Zambrano and Alvarado (1984) evaluated a new design of chimney that reduces dangerous temperatures in natural convection solar dryers. The new chimney is truncated inverted cone shaped. The chimney has 4.3 m height, 0.3 m smaller diameter and 8° slant angle. It was tested against conventional cylindrical chimney having similar dimensions. Testing revealed that stack air velocity is more than twice in proposed chimney when compared to cylindrical chimney. Chimney was also tested for drying peaches. While temperature as high as 78°C was recorded using cylindrical stack, it came down to 66°C with conical chimney.

Numerous innovative designs for solar dryers have been developed to suite different requirements among which, indirect multi-shelf dryers, tunnel dryer, dryer integrated with photovoltaic cells are the popular ones (Anyanwu et al 2012).

Singh et al (2004) designed, constructed and tested multi-shelf portable solar dryer having unique capability of working in both direct as well as indirect mode. The dryer utilised the principle of natural circulation. Components have been designed in such a way that it is quite easy to mantle and dismantle. Thus the dryer can be disassembled while not in operation, thereby saving space. Inclined at 45°, the dryer has aperture area of 3.34 m². Seven perforated trays can hold 10-30 kg depending upon the product. Dryer could reach stagnation temperature of 75°C under the test conditions of 750 W/m² solar radiation and 30°C ambient temperature. The dryer can be operated either in batch mode or in newly introduced semi-continuous mode. In semi-continuous mode, the trays that get empty due to shrinkage of the product are filled with fresh product while partially dried product from different trays is combined. The dryer was tested in batch mode with 11.13 kg fenugreek leaves. Drying took 18.75 hours spread over three days. Thermal efficiencies were 28.55, 16.2 and 8.6% for first, second and third day respectively. According to authors, reduced values of efficiency on successive days are due to empty trays. In semi-continuous mode testing, 4.770, 3.180 and 6.360 kg fresh leaves were added on second, third and fourth drying days. Thermal efficiencies were 28.96, 27.6, 23.4 and 25.3% respectively for first to fourth days. Higher efficiency on latter days is due to fully loaded trays which shows the superior efficiency induced by semi-continuous mode.
A lot of research is being carried out to integrate thermal storage in solar drying applications. Packed beds represent the most suitable storage units for air based heating systems. A packed bed system consists of loosely packed solid material through which the heat transport fluid is circulated. Heated air flows from solar collectors to bed of particles from top to bottom during which thermal energy is transferred. This is known as charging phase. When energy is needed from storage, the air flow is reversed. This process is known as discharging phase. The storage system is well insulated and installed near the solar collectors to avoid heat losses (Singh et al 2010).

Tiwari et al (1994) investigated the effect of thermal storage on crop drying. Experiment was set up and simulated in the laboratory. Wheat grains were taken as drying product and its moisture content was brought down to 11% which is optimal for safe storage. Temperature of drying air was varied between 50 and 70°C with the help of heating coils and forced air circulation. Rocks 5-8 cm in diameter, having density 1750 kg/m³ and specific heat 0.81 kJ/kg-K were used as sensible heat storage medium. It was found that the temperature fluctuations are significantly reduced and the quality of final product was much better.

Shalaby and Bek (2014) tested the performance of indirect solar dryer in two cases. In the first case, it was tested using phase change material (PCM) and in the second case, free of PCM. The storage was incorporated at the bottom of drying chamber. It consists of two cylindrical containers filled with paraffin wax, along with uniformly distributed 32 copper tubes for air passage. The results indicated that use of PCM smoothened the drying temperature for approximately seven consecutive hours per day. Also the drying temperature achieved after 2 p.m. was 3.5-6.6°C more than that of second case. Even after the sunset, the temperature of drying air was 2.5-7.5°C higher than ambient for 5 hours atleast.

Jain and Tewari (2015) designed a natural convection indirect type solar crop dryer with integrated thermal storage. The dryer consists of flat plate collector, packed bed thermal storage, drying chamber and natural draft system. Packed bed storage is placed below drying chamber. It consists of cylindrical tubes filled with paraffin wax as PCM. Drying chamber consists of six rectangular drying trays placed in three rows and two columns. Natural draft system is located above drying chamber. It comprises of another flat plate collector with PCM placed underneath absorber plate. The draft system enhances the convection by setting up thermal buoyant forces. It heats up the leaving air to the temperature higher than that of incoming air and due to this temperature difference, natural currents are induced. During sunshine hours, air from flat plate collector at first passes through storage system where it loses a part of its heat to PCM, and rest it takes to the drying chamber which starts drying the crop. Moist air from drying chamber moves upward, comes in contact with natural draft system where it gets heated and finally leaves the dryer. During off sunshine hours, the energy stored in PCM is released which helps to continue drying for another 5-6 hours. The developers examined the performance of the dryer by drying mint leaves. They observed that during initial hours of the day, the temperature below the packed
bed (that of incoming air) was higher as compared to the temperature above it (that of air coming out of packed bed) which clearly reflects the charging phase. After 2 p.m. the temperature above packed bed was higher than that of incoming air which shows that energy stored in PCM helped to maintain the temperature in the range of 40-45°C. Thermal efficiency was calculated 28.2%. The dryer was also found to be economically viable with payback period of 1.5 year.

Ayyappan and Mayilsamy (2010) analysed tunel dryer coupled with sensible heat storage. It was found that use of heat storage reduced the drying time considerably. With thermal storage, time taken to reduce moisture level in copra to 7% from initial value of 52.3% took 52 hours as compared to 78 hours without using the heat storage. The open sun drying, on the other hand, took 172 hours which clearly indicates the advantage of using solar dryer.

Madhlopa and Ngwalo (2007) fabricated and tested indirect type natural convection solar crop dryer with biomass burner and integrated thermal storage. The dryer can be operated in three modes- solar, biomass and solar-biomass. The energy released by burning biomass is first transferred to pebble bed storage from where it is passed on to absorber plate (placed upon storage unit) and consequently to the drying air. The system was tested with pineapples having initial moisture content of 669% (dry basis). Solar-biomass mode came out to be the fastest among all modes to lower down the moisture to 16% while solar mode was slowest. Hybrid mode is most suitable for the sensitive crops which need to be dried in short time to preserve the quality. System efficiency was recorded highest (15%) for solar, intermediate (13%) for solar-biomass and lowest (11%) for biomass mode.

Ugwu et al (2015) designed and evaluated mixed mode solar kiln having pebble bed sensible heat storage. Single glazed, black-painted rock bed serves as both solar absorber and energy storage medium. The air passes through the space between absorber bed and glass cover, and gets heated. The air then flows to the drying compartment specifically designed for timber drying. During testing, the maximum drying chamber temperature was recorded 61.7°C. Solar kiln was able to reduce the moisture content of okpeye timber from 66.27% to 12.9% within 360 hours whereas open sun drying could only bring it down to 20.1% during the same period.

Das et al (2001) built and analysed a tray dryer with air recirculation. The laboratory model of the dryer consists of drying unit, air blower, heating chamber, air distribution system, air recirculation system and control panel. The drying unit has two chambers of size 300 mm x 300 mm x 400 mm separated by inlet air duct. Each chamber can hold 3 trays of size 300 mm x 300 mm x 40 mm. Space between trays is maintained at 50 mm for good air flow. Three rectangular slots (300 mm x 50 mm) are provided on each side of both drying chambers. These slots serve as inlet and outlet for each tray. Hot air is fed from the centre to reduce variation of moisture removal between inlet and exhaust end which is common in conventional tray dryers. Flow rate over trays is controlled using baffles. Air leaving drying chamber from sides is collected at the top in exhaust cum recirculatory system. From here it is sent for recirculation depending upon the ratio selected.
with the help of baffle. With air velocity of 1.5-1.7 m/s, 1kW heater, ambient temperature 25°C and inlet temperature set to 70°C, initial heating time of air in drying chamber took 25 minutes and 12 minutes for no recirculation and 100% recirculation respectively. Under load conditions, dryer was tested with potato chips having uniform thickness of 1 mm for temperatures of 55, 60, 65°C and varying loading rate of 5, 6, 7 kg/m². Recirculation was set to 95% while air velocity over trays was maintained in the range of 1.5-1.7 m/s. Heat utilisation factor and thermal efficiency were found to be in the range of 17-20% and 21-24% respectively.

2.3 Thermal Storage

Thermal storage generally involves heat transfer fluid (HTF) and energy accumulator. Temperature difference between HTF and material is required to release or store the energy. It involves 3 phases namely charging, standby and discharging.

Mawire et al (2010) evaluated heat transfer characteristics during charging for oil and glass pebble thermal storage. Size of the system was kept very small as it charges up faster than a larger system and enables rapid experiments under different conditions. Setup consists of glass tube in which glass and pebble were contained. Glass tube had external diameter of 40 mm, thickness 2.58 mm and height 450 mm. Oil used to charge thermal storage was heated with an electrical hot plate providing constant heat flux. The plate was maintained at uniform temperature. The oil leaving from the storage bed was circulated through hot plate and sent to storage again in a closed loop. Axial temperatures of oil/pebble storage was measured by thermocouples installed at various locations. Pebbles had density of 2500 kg/m³ and specific heat 789 J/kg-K. Experiments were performed at different oil flow rates of 4.24, 5.92 and 7.54 ml/s while hot plate being maintained at 200°C. The outlet temperature of oil leaving hot plate was around 155°C for all cases. Rise in temperature of storage was faster at higher flow rates. Heat transfer coefficients were 15.530, 21.770 and 27.363 kW/m²-K for respective flow rates at 200°C. Linear relationship was found between charging flow rate and volumetric heat transfer coefficient.

Srinivasan and Raghunathan (2013) analysed response of low aspect ratio packed beds at high Reynolds numbers. Packed beds of lengths 200 and 300 mm were characterised for Reynolds number range of 6000 to 85000. Runs were conducted at various configurations. Commercially available 5 and 9.5 mm steel ball bearings were chosen as packing material. Test section had internal diameter of 90 mm. Maximum inlet flow temperature varied for different configurations from 73.4 to 212°C. Experiments were conducted in an open loop using hydrogen air heater based system. Results revealed that gas density across the bed length was almost constant. Pressure drop increased with increase in Reynolds number as well as packing density. Thermocouples placed at radial locations indicated less than 2.5% temperature variation. Nusselt number varied axially along the bed. In case of 9.5 mm spheres, Nusselt number was found to be decreasing at centre of the bed compared to start and end whereas for 5 mm bearings, it followed increasing trend upto
centre, then started decreasing. The causes of variation were thought to be boundary layer separation and its reattachment, relaminarisation of the turbulent boundary layer, variation of Reynolds number across the axial length due to temperature changes. Average Nusselt number increased at higher Reynolds number.

Singh et al (2006) determined Nusselt number and friction factor corelations for packed bed storage system with large sized elements. Data required for determination of corelations, namely heat transfer rate and pressure drop was found experimentally for different set of parameters. Five shapes of storage elements were investigated at various flow rates and void fraction values. Storage tank with diameter 0.6 m and effective bed height of 0.75 m was constucted using 3 mm thick MS sheet. Electric heater was used to supply hot air. Pressure drop across the bed was measured using micromanometer. Sphericity values used in the experiment were 0.55, 0.63, 0.72, 0.80 and 1.00. At constant sphericity, Nusselt number as well as friction factor was found to be increasing with decrease in void fraction. It was due to more surface area of contact and enhanced turbulence. Holding void fraction constant, Nusselt number and friction factor both decreased as value of sphericity decreased from 1 to 0.8. Then they started increasing with further decrease in sphericity from 0.8 to 0.55. Change of flow patterns and area of contact were supposed to be responsible for such trend which were well explained in the paper. Further Nusselt number was found to be increasing with rise in Reynolds number whereas friction factor slightly decreased. Corelations for Nusselt number and friction factors in terms of sphericity, void fraction and Reynolds number were reported.

Often there exists a gradient in thermal storage tank. Material close to inlet port during charging is at higher temperature than particles near outlet. It is called thermal stratification. It enhances the efficiency of the system during charging as well as discharging (Oro et al 2013, Eamesi and Norton 1998). Factors that affect thermal stratification are location of inlet and outlet ports, variation in temperature and mass flow rate during charging and discharging, dimensionless numbers of heat and mass transfer (Haller et al 2009). Generally, thermal gradient peaks as the charging proceeds, then starts falling when charging progresses towards the end (Mawire and Taole 2011). It also decreases during standby mode as axial conduction leads to transfer of heat from high temperature rocks to low temperature side (Stamps and Clark 2012). To get best out of the system, starified charging and discharging techniques should be favoured (Glembin and Rockendorf 2012).

Phase changing materials can store much larger amount of heat as compared to sensible heat storage materials. This is because heat of fusion of the materials is many folds higher than their specific heat. For instance, melting of ice needs 80 times energy as compared to energy needed for raising temperature of water by one degree. It makes PCM based storage design much compact than sensible options. There are four phase transformations namely solid-solid transformation, solid-liquid transformation, liquid gas transformation and solid-gas transformation or sublimation. Of all these transformations, vapourisation and sublimation can store the highest
amount of energy per unit weight. However, requirement of large pressure vessels limit their use in practical systems. Most feasible is the solid-liquid transformation in which volume changes are almost negligible (Lorsch et al 1975).

There are several configurations in which PCM is employed. Some of them are double pipe, triplex tube, shell and tube, rectangular or slab container and encapsulation. Double pipe system involves two concentric tubes. In one tube, PCM is filled whereas heat transfer fluid (HTF) flows through the other (Agyenim et al 2010). Triplex tube configuration incorporates PCM in middle tube while HTF fluid passing through inner as well as outer tube. (Al-Abidi et al 2013). Similarly in shell and tube setup, PCM fills the tubes and HTF flows outside those tubes in a shell (Ismail and Abogderah 2000). Rectangular configuration has PCM contained in thin slabs while HTF flows parallel to them (Xu et al 2015). Encapsulation technique involves enclosure of PCM in small leakproof shells, which are then packed in a container. Huge surface to volume ratio is obtained in this technique (Salunkhe and Shembekar 2012).

Assis et al (2007) explored the process of melting of PCM encapsulated in spherical geometry through numerical as well as experimental investigation. Study was performed for shell diameters of 40, 60 and 80 mm at uniform wall temperature. For different cases it was kept 2 to 20°C above the melting temperature of PCM. The material chosen was commercially available RT27 having melting interval 28-30°C. Solid phase with flat top surface occupied 85% of capsule volume. Practical runs were conducted in a transparent tank filled with water. Temperature of bath was maintained with the use of electric heater and stirrer. Visualisation of melting process in glass shell was chosen as the investigation method. It was observed that solid phase typically descends as the time passed. Volumetric expansion was also observed. Melt fraction increased more rapidly when the temperature difference was higher. Melting was observed to be faster in case of smaller shells at similar conditions. Solid fraction sank in the liquid material. Natural convection was found between upper and lower part of the melt due to temperature difference between wall top and relatively cold solid surface at the bottom.

Regin et al (2009) numerically investigated behaviour of packed bed latent heat storage system. Packed bed consists of paraffin wax (melting point 58-60°C) filled spherical capsules in a cylindrical tank 1 m in diameter and 1.5 m in length. Water was selected as HTF. Study revealed that complete solidification time is longer as compared to melting. It is due to resistance in heat transfer induced by solid layer formation on inner wall of capsule during solidification. Higher HTF temperature resulted in shorter time for complete charging. Similar effect was caused by higher mass flow rate. Charging as well as discharging rate were higher for small capsules compared to larger. Melting of PCM in a range, rather than at a fixed temperature decreased the charging time. This is because of higher heat transfer rate in former case. Despite being heat transfer coefficient same, lower average temperature of bed caused by early melting resulted in larger temperature difference between fluid and storage material which enhances energy transfer.
Not only sensible heat storage (SHS) is a well-developed technology, the systems are easy to construct. But they suffer from low heat storage capacity per unit volume. Latent heat storage (LHS) systems provide high heat storage density as well as isothermal behavior during charging and discharging processes. Yet these are not popular for commercial use as SHS systems. It is because of poor heat transfer characteristics of the storage materials which demand complicated heat exchanger designs, ultimately increasing the storage cost. The combined sensible and LHS system reduces the difficulties experienced in both of the above individual techniques to some extent and possess the advantages of both the systems (Nallusamy et al 2007).

Okello et al (2014) studied the storage configuration combining PCM and rock particles, and compared it with rocks only. The container for thermal storage system was constructed using two vertical coaxial steel cylinders having diameters 0.3 and 0.4 m. Highly reflecting steel foils were inserted in space between cylinders to minimise radiative heat losses. Later on, the space between cylinders was sealed. Binary mixture of NaNO$_3$ and KNO$_3$ in 60:40 ratio, having melting point of 220°C was used as PCM. Mixture was contained in 4 copper tubes each having 0.9 m length and 0.5 diameter. Tubes were inserted axially in the main container. Space between main cylinder and PCM tubes was filled with crushed stones. Hot air was introduced from the top. Nine thermocouples were installed at different vertical positions 0.1 m apart to record the temperatures. At inlet air temperature of 350°C, charging temperature profiles, thermal degradation profiles and transient charging efficiencies were compared for both configurations. It was observed that bottom temperature for the bed containing rocks and PCM cylinders increased much faster as compared to thermal storage with rocks only. It was caused by enhanced heat conduction down the bed due to copper tubes. Thus the vertical heat flow along copper length was higher than horizontal heat flow along cylinder radius required for heating PCM. Better stratification was obtained in case of bed containing only rocks. In thermal degradation testing, thermal stratification loss in bed containing two materials was observed to be faster. Drop in transient charging efficiency was rapid in case of combined sensible and PCM heat storage. Overall, energy density in this design was increased but at the expense of thermal equalisation within storage volume.

Ling and Poon (2013) carried out in depth review of integrating PCMs in concrete. Methods of incorporating, effect on thermal as well as mechanical properties of concrete and stability of the mixture are presented in the paper. The predominant incorporation techniques include immersion, impregnation and direct mixing. Depending upon the integration technique, PCMs affect the properties of concrete like compressive strength and fire resistance. While direct mixing significantly reduces the compressive strength of concrete, immersion technique has no effect. Flammable PCMs like paraffin wax, if not properly encapsulated reduces fire resistance of concrete. Also not all PCMs are suitable for use because high alkali level in concrete degrades some PCMs. Organic PCMs are favoured because of their stability in alkali environment. As far as retention is concerned, there is some mass loss after some cycles in case immersion technique is
employed. For regular concrete, it could be as high as 30% after 20 heating and cooling cycles. But it could be controlled using autoclaved concrete blocks with just 5% loss reported.

Many industrial by-products exhibit excellent properties, much better than conventional materials, which make them good candidates for use in thermal storage. Along with better performance, use of these materials also solve problem of disposing industrial wastes.

Miro et al (2014) tested the suitability of a solid by-product coming from potash industry to be used for industrial sensible heat recovery. Complete analysis of thermo-physical properties was done at laboratory scale and then in a high temperature pilot plant of university of Lleida. Thermal cycles were performed with the storage material. The material had been selected for its low price, availability and similar thermal characteristics with other sensible heat storage materials. The substance was found to be a good candidate for temperatures up to 200°C. The product was identified as crystalline sodium chloride by X-ray diffraction. Specific heat capacity was measured 0.738 kJ/kg-K for the range of 100-200°C. Salt was tested for original granulated form 1-2 mm and treated form in which air voids were filled by water treatment. Salt in treated form was found to be more useful with thermal conductivity of 2.84 W/m-K and density 2050 kg/m$^3$.

Fernandez et al (2015) analysed the feasibility of using slag for thermal energy storage. It was found to be low cost, successful material suitable for temperatures up to 1100°C. Cost of slag is same as concrete while properties are better. Tables 2.1 and 2.2 enlist various thermal storage materials along with their properties.

### Table 2.1 Sensible heat storage materials and their properties.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m$^3$)</th>
<th>Thermal Conductivity (W/m-K)</th>
<th>Specific heat (kJ/kg-K)</th>
<th>TCLP$^+$ (°C)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>1000</td>
<td>0.58</td>
<td>4.19</td>
<td>20</td>
<td>Tatsidjodoung et al 2013</td>
</tr>
<tr>
<td>Glass</td>
<td>2700</td>
<td>0.78</td>
<td>840</td>
<td>20</td>
<td>Dhifaoui et al 2007</td>
</tr>
<tr>
<td>Rocks</td>
<td>2804</td>
<td>2.9</td>
<td>0.8</td>
<td>N.A.</td>
<td>Kurklu et al 2003</td>
</tr>
<tr>
<td>Brick</td>
<td>1600</td>
<td>1.2</td>
<td>0.84</td>
<td>20</td>
<td>Tatsidjodoung et al 2013</td>
</tr>
<tr>
<td>Concrete</td>
<td>2240</td>
<td>0.9-1.13</td>
<td>1.13</td>
<td>20</td>
<td>Tatsidjodoung et al 2013</td>
</tr>
<tr>
<td>Marble</td>
<td>2500</td>
<td>2.0</td>
<td>0.880</td>
<td>20</td>
<td>Tatsidjodoung et al 2013</td>
</tr>
<tr>
<td>NaCl (solid)</td>
<td>2160</td>
<td>7</td>
<td>0.85</td>
<td>N.A.</td>
<td>Pintaldi et al 2015</td>
</tr>
<tr>
<td>Magnesia fire bricks</td>
<td>3000</td>
<td>1</td>
<td>1.15</td>
<td>N.A.</td>
<td>Pintaldi et al 2015</td>
</tr>
<tr>
<td>Cast iron</td>
<td>7200</td>
<td>37</td>
<td>0.56</td>
<td>N.A.</td>
<td>Pintaldi et al 2015</td>
</tr>
<tr>
<td>Stainless steel</td>
<td>7816</td>
<td>16.3</td>
<td>460</td>
<td>20</td>
<td>Dhifaoui et al 2007</td>
</tr>
<tr>
<td>Aluminium</td>
<td>2700</td>
<td>205</td>
<td>0.896</td>
<td>N.A.</td>
<td>Aly and El-Sharkawy 1990</td>
</tr>
</tbody>
</table>

$^+$ TCLP stands for Temperature Corresponding to Listed Properties
Table 2.2 Latent heat storage materials and their properties.

<table>
<thead>
<tr>
<th>Material</th>
<th>Melting point (°C)</th>
<th>Density (kg/m³)</th>
<th>Latent heat (kJ/kg)</th>
<th>Thermal Conductivity (W/m-K)</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>n-Hexacosane (Paraffin)</td>
<td>56</td>
<td>770</td>
<td>257</td>
<td>0.21 (s)</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>N-Nonadecane (paraffin)</td>
<td>31.9</td>
<td>912(s)</td>
<td>222</td>
<td>0.21 (s)</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>Capric acid</td>
<td>32</td>
<td>878</td>
<td>152.7</td>
<td>0.153</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>Stearic acid</td>
<td>41-43</td>
<td>1007(s)</td>
<td>211.6</td>
<td>1.6</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>Palmitic acid</td>
<td>57.8-61.8</td>
<td>989(s)</td>
<td>185.4</td>
<td>0.162</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>Lauric acid</td>
<td>42-44</td>
<td>1007 (s)</td>
<td>178</td>
<td>1.6</td>
<td>Su et al 2015</td>
</tr>
<tr>
<td>Rubitherm GR 27 (GPCC*)</td>
<td>21-29</td>
<td>N.A.</td>
<td>64.8</td>
<td>N.A.</td>
<td>Rady 2009</td>
</tr>
<tr>
<td>Rubitherm GR 41 (GPCC)</td>
<td>31-45</td>
<td>N.A.</td>
<td>65.9</td>
<td>N.A.</td>
<td>Rady 2009</td>
</tr>
</tbody>
</table>

* Solid phase
* GPCC stands for Granular Phase Changing Composites

From the above literature review, it can be concluded that dryers operate at pretty much less efficiency. This translates to good scope for heat recovery operations. Till now, integration of thermal storage has been performed at inlet side to elongate the available drying time. Proposed storage location in current research not only extends the drying time but also improve overall efficiency of the system. Packed bed sensible heat storage is favoured because it does not require complicated heat exchanger, thereby saving cost and fabrication time.
Chapter 3  METHODOLOGY

The research has been conducted at Solar Energy Laboratory, School of Renewable Energy Engineering, Punjab Agricultural University, Ludhiana located at 30.90°N, 75.86°E. Experimental setup comprising of drying chamber, electric air blower, air heating unit and heat recovery system was fabricated and evaluated. In order to determine size requirements for thermal storage, heat in exit air during conventional drying process needs to be known which was found by conducting various trial runs. Results of the trial runs are given in Appendix A. Based on these results, storage unit was designed. For the present study, in order to simulate the heating process and to have control over the drying temperature, solar air heater has been replaced with electric heater. The experimental setup, instrumentation and testing procedure are described below:

3.1 Experimental setup

The crop dryer consists of a multi-tray drying chamber to hold the product to be dried, centrifugal blower to supply the air and heating unit to raise the temperature of drying air as shown in Fig. 3.1. This dryer is coupled to the thermal storage which acts as heat recovery unit. Fig. 3.2 shows actual view of the experimental setup. The details of these components are given below:

3.1.1 Drying chamber

Drying chamber is constructed using standard 20 mm x 20 mm x 3 mm angle iron frame surrounded by 5 cm thick thermocol sheet forming walls and at the same time providing insulation (Fig. 3.3). It can hold 10 trays each one having length 53 cm, width 17 cm and height 5 cm. Trays are placed one above another as shown in Fig. 3.4. Six millimeters clearance is provided between each tray and supporting base to ensure effortless removal. Base of the trays is made up of wire mesh (0.5 mm wire diameter with 4 pores per cm length) which holds the product, at the same time allows the passage of air. Hot air enters from the bottom and leaves from the top. The air distributor is located at the bottom which also acts as reservoir to reduce temperature fluctuations. With the net effective height of 76 cm, the dehydrating chamber can accommodate around 3-8 kg depending upon the product.

3.1.2 Air blower

For air flow through the system, single phase 2800 rpm centrifugal type air blower is used which has electrical rating of 190 W.

3.1.3 Air heating unit

Air heating unit is composed of three heating elements, each of them having 1 kW rating
and 23 cm length. These are installed in parallel configuration in a 80 mm diameter galvanised iron (GI) pipe. The cold air enters from one end, heats up as it passes over the length, and leaves from other end of the pipe at elevated temperature.

### 3.1.4 Thermal Storage

Rock bed storage has been selected as sensible heat storage unit. Easy availability, simplicity in design, high surface to volume ratio, minimal tendency to absorb water, long life and minimal cost are some of its benefits. Table 3.1 enlists other materials which were also considered while selecting the storage media, but could not get finalised due to their limitations.

**Table 3.1 Sensible heat storage options.**

<table>
<thead>
<tr>
<th>Material</th>
<th>Limitations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water</td>
<td>Requires heat exchanger</td>
</tr>
<tr>
<td>Soil dry</td>
<td>Low effusivity</td>
</tr>
<tr>
<td>Soil wet</td>
<td>Requires heat exchanger</td>
</tr>
<tr>
<td>Masonary bricks</td>
<td>Absorb moisture</td>
</tr>
<tr>
<td>Concrete</td>
<td>Absorbs moisture</td>
</tr>
<tr>
<td>Glass</td>
<td>Limited availability</td>
</tr>
<tr>
<td>Iron</td>
<td>Rusting may occur due to moist air</td>
</tr>
</tbody>
</table>

Small rocks have high bulk density, better heat transfer due to more surface to volume ratio, but they cause huge pressure drop. It might be possible that pumping power exceeds the recovered energy. On the other hand, large rocks require less pumping power but have inferior heat transfer characteristics as compared to small rocks. Therefore, medium sized rocks were selected which provide balance in both directions. The crushed rocks used are 65 mm industrial grade, same that are used on railway tracks and for construction of the roads.
Fig. 3.1 Schematic diagram of the experimental setup.

Fig. 3.2 Actual view of experimental setup.
Fig. 3.3 Pictorial view of drying chamber without trays.

Fig. 3.4 Pictorial view of drying chamber with trays.
Quantity of the storage media has been selected based on preliminary testing. The results are discussed in Appendix A. The storage was designed for accumulating about 4500 kJ of heat with temperature rise of 20°C. Following numerical relations were used for calculation purposes:

\[ M_s = \frac{E_{s,\text{targetted}}}{e_s} \]  \hspace{1cm} ...(3.1)  
\[ e_s = C_{p,s} (T_{s,2} - T_{s,1}) \]  \hspace{1cm} ...(3.2)  
\[ V_{\text{storage}} = \frac{M_s}{\text{Bulk density of stones}} \]  \hspace{1cm} ...(3.3)

where

- \( M_s \) is weight of storage material required (kg)
- \( E_{s,\text{targetted}} \) is total amount of energy for which storage is targetted (kJ)
- \( e_s \) is energy stored per kg of storage media (kJ/kg)
- \( C_{p,s} \) is specific of the storage material (kJ/kg K)
- \( (T_{s,2} - T_{s,1}) \) is possible temperature rise for thermal storage (°C)
- \( V_{\text{storage}} \) is volume of the storage tank (m³)

Assuming specific heat of the rocks 0.8 kJ/kg (Kurklu et al 2003), the material quantity came out to be around 275 kg which was packed in rectangular container.

Storage shell is constructed of 24 gauge GI sheet. It measures 60 cm x 60 cm in cross section with 135 cm total height. Constructed height has been kept more for provision of more storage media if needed. It is made air tight with seam joints and rivets. Supports are provided in mid section to prevent bulging. The system is supported on 30 mm x 30 mm x 3 mm angle iron frame. Bottom is fixed while top is kept removable. Gasket is placed between top lid and main body to prevent leakage. Ports having 50 mm diameter are provided on top and bottom to allow passage of air.

Air enters from the top, charges the rocks and leaves from the bottom in relatively colder state. During discharging, ambient air enters bottom enters and leaves from the top in hot condition. Four inch gap is provided between base and rock bed to allow proper distribution of air with the help of wire mesh mounted on base for supporting rocks. The storage unit is insulated with 5 cm thick thermocol on all sides.

Bulk density of the rocks in packing was 1450 kg/m³. It was determined using total volume occupied by all the rocks and their weight. Occupied volume was directly noted from the storage shell. True density of the rocks used is 2460 kg/m³. It was calculated from the volume of water displaced by the rock. Void fraction, that is the empty space left during packing of rocks in the system is 0.4 which was calculated as:

\[ \text{Void fraction} = 1 - \left( \frac{\text{Bulk density}}{\text{True density}} \right) \]  \hspace{1cm} ...(3.4)

Thermal conductivity and heat capacity have been cited in literature and are assumed to be 2.9 W/m-K and 0.8 kJ/kg respectively (Kurklu et al 2003). Table 3.2 summarises the storage specifications.
Table 3.2 Storage specifications.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Area of cross section</td>
<td>0.36 m²</td>
</tr>
<tr>
<td>Effective rock-bed length</td>
<td>0.53 m</td>
</tr>
<tr>
<td>Clearence between bottom of the tank and rock-bed</td>
<td>10 cm</td>
</tr>
<tr>
<td>Total height of storage tank</td>
<td>1.35 m</td>
</tr>
<tr>
<td>Amount of storage material</td>
<td>275 kg</td>
</tr>
<tr>
<td>Bulk density of stones</td>
<td>1460 kg/m³</td>
</tr>
<tr>
<td>True density of stones</td>
<td>2450 kg/m³</td>
</tr>
<tr>
<td>Void fraction of packed stones</td>
<td>0.4</td>
</tr>
<tr>
<td>Specific heat of stones</td>
<td>0.8 kJ/kg</td>
</tr>
<tr>
<td>Thermal conductivity of stones</td>
<td>1.2 W/m-K</td>
</tr>
<tr>
<td>Thickness of insulation</td>
<td>5 cm</td>
</tr>
<tr>
<td>Thickness of GI sheet</td>
<td>24 gauge</td>
</tr>
<tr>
<td>Ground clearence</td>
<td>45 cm</td>
</tr>
</tbody>
</table>

3.1.5 Connecting pipes
The complete fitting was done with 50 mm diameter GI pipes. Pipes were insulated with 4 layers of felt, each layer being 2 mm thick to prevent heat losses to the surroundings. Ball valves were used to change the circuit and control air flow rate.

3.2 Working of the setup
The setup has been designed to work in two modes namely storage charging and storage discharging mode. Fig. 3.5 shows air flow through system in charging mode and Fig. 3.6 shows the system during discharging mode. In storage charging or regular mode, that is during day time, valves 1, 3 and 5 are opened while valves 2 and 4 are closed. This directs the air from drying chamber to thermal storage before being thrown out. During this time, charging of the storage material occurs. In reverse or storage discharging mode, valves 1 and 3 are closed while valves 2, 4 and 5 are opened. Fresh air is sucked from bottom of storage, heated by it, performs dehydration in drying chamber and escapes from valve 2.
Fig. 3.5 Air flow in experimental setup during storage charging mode.

Fig. 3.6 Air flow in experimental setup during storage discharging mode.
3.3 Instrumentation

Various instruments were used for the measurement of temperature, air flow rate, weight of drying product etc. These are described below:

3.3.1 Temperature measurement

Air temperatures were recorded using mercury in glass thermometers having range 0-110°C and least count of 0.5°C. Measurement locations were outlet of the drying chamber, top port of the thermal storage and inlet of the blower. The ambient temperature was also measured.

3.3.2 Air flow measurement

Air flow rate was measured with the help of vane type anemometer (Make: TSI, Model: 8322-M-GB). During all the experiments, flow was measured at the exit locations which vary depending upon the circuit. For preliminary testing, it was at outlet of the drying chamber. For final testing, it corresponds to exit of the thermal storage unit during storage charging mode and exit of the drying chamber during storage discharging mode.

3.3.3 Weight measurement

Weight of the crop was recorded using digital weighing scale having range 0-10 kg and least count of 1 g. Weight of the individual stones was also measured with this scale to find out true density of stones. Another weighing scale of 500 kg measuring capacity and least count of 10 g was used for measuring weight of all the stones collectively while putting them in storage tank.

3.3.4 Input energy measurement

Although input energy was obtained through calculations, still for cross-checking purpose, single phase, digital electricity energy meter having least count of 0.1 kWh was installed. Meter measured combined energy input for air heating unit and the blower.

3.3.5 Volume measurement

In order to determine density of stones, volume of water as well as combined volume of stone and water was measured using a beaker having capacity 1 litre and least count of 10 ml.

3.3.6 Relative humidity measurement

Relative humidity of ambient air was measured using digital data logger (Make: Lascar, Model: EL-USB-2-LCD).

3.3.7 Temperature controller

Digital temperature controller (Make: Delta, Model: DTA-4848) with least count 0.1°C was
used to maintain the temperature of air at inlet of drying chamber to desired level. The controller acts as a switch to electric heater. When temperature of incoming air falls below desired level, the temperature controller turns on electric heater and switch the heater off when temperature reaches the desired value. J-type iron constantan thermocouple was used to sense inlet air temperature as input signal to temperature controller.

3.4 Experimental Procedure

The testing of the system was performed by drying carrots at a fixed air flow rate of 0.025 kg/s and four drying air temperatures of 55, 60, 65 and 70°C. For each experiment, same amount of product (5.850 kg) was dried at a uniform air inlet temperature and constant mass flow rate. The product to be dried was equally distributed in 9 trays. Data was recorded at the interval of one hour. For weighing, trays were taken out of the drying chamber and during that period, energy supply was shut off. In order to minimise recording time, combined weight of crop and trays was measured. Weight of empty trays was subtracted later on to get the actual quantity of the product. Minimizing the recording time was necessary to avoid heat losses from drying chamber and trays to surroundings so that continuity of the experiment could be preserved. Drying was continued until moisture content came under 10% on wet basis (wb). Three sets of experiments were performed during testing:

1. Set A: Drying without heat storage
2. Set B: Drying with heat storage
3. Set C: Drying with heat storage and auxiliary heating.

3.4.1 Set A: Drying without heat storage

In this set, drying was performed at four drying air temperatures of 55, 60, 65 and 70°C without any interruption to study the drying characteristics of the product.

3.4.2 Set B: Drying with heat storage

In this set of experiments, the stored energy during day time was used to heat the air during off-sun hours. All the experiments were started at 10 a.m. After 6 hours, the flow was reversed and electric heaters were switched off. Drying continued until the inlet air temperature remained 7°C higher than ambient temperature. Rest of the drying was continued on next day at selected constant temperature. Product was taken out of the drying chamber and kept in respective trays in the room.

3.4.3 Set C: Drying with heat storage and auxiliary heating

In this setup, during discharging of thermal storage the auxiliary heat was also supplied. This facilitated the use of stored energy at lower temperature which was not practical with Set B. Drying was carried out at uniform temperature by using electric heaters during reversed mode.
First, air got pre-heated with the stored energy. Remaining boost in temperature was provided by auxiliary heating. All experiments were started at 10 a.m. Flow was reversed after 6 hours. Rest of the parameters were kept same as earlier sets of the experiments. Drying was performed without any break until recorded moisture content came down under 10% on wet basis.

3.5 Data Reduction

3.5.1 Determination of Input heat energy

At any instant, energy given to raise the temperature of drying air is given as:

\[ q_i = m_a C_{p,a} (T_d - T_a) \]  

...(3.5)

where  
- \( T_d \) is drying temperature \(^\circ\)C
- \( T_a \) is ambient temperature \(^\circ\)C
- \( C_{p,a} \) is specific heat of air at constant pressure (kJ/kg-K)
- \( m_a \) is air flow rate (kg/s)

Energy requirements for the interval of one hour (\( Q_{i,h} \)) can be calculated by replacing mass flow rate per second with mass flow rate per hour, instantaneous \( T_d \) and \( T_a \) by hourly average respective temperatures \( T_1 \) and \( T_2 \). Mathematically,

\[ Q_{i,h} = M_a C_{p,a} (T_1 - T_2) \]  

...(3.6)

where  
- \( M_a \) is air flow rate per hour (kg/h)
- \( T_1 \), the mean inlet air temperature here is same as set on temperature controller \(^\circ\)C
- \( T_2 \) is average temperature of ambient air given as

\[ T_2 = \frac{T_{a,1} + T_{a,2}}{2} \]  

...(3.7)

Here \( T_{a,1} \) and \( T_{a,2} \) are temperature of ambient air recorded at start and end of hourly time interval.

Total input energy required for heating air during whole drying process (\( Q_{i,t} \)) is determined by the summation of hourly input energy.

3.5.2 Determination of sensible heat at exit of drying chamber

Heat in exit air at any instant is given as:

\[ q_w = m_a C_{p,a} (T_{out} - T_a) \]  

...(3.8)

where  
- \( T_{out} \) is the temperature of exit air \(^\circ\)C

For the interval of one hour, energy in exit air (\( Q_{w,h} \)) is determined from hourly mean exit air and ambient temperatures and hourly mass flow rate. Mathematically,

\[ Q_{w,h} = M_a C_{p,a} (\bar{T}_{out} - \bar{T}_a) \]  

...(3.9)

where  
- \( \bar{T}_{out} \) is average temperature of exit air for that particular hour given as:

\[ \bar{T}_{out} = \frac{T_{out,1} + T_{out,2}}{2} \]  

...(3.10)

\( T_{out,1} \) and \( T_{out,2} \) are temperatures of exit air at beginning and end of the hourly interval.

Total heat in exit air during drying process (\( Q_{w,t} \)) is calculated by summation of hourly heat in exit air.
3.5.3 Fraction of heat energy in exit air

The percentage of heat in exit air for each hourly interval is calculated as:

\[ F_h = \left( \frac{Q_{w,h}}{Q_{i,h}} \right) \times 100 \]  

...(3.11)

For complete drying process, \( F_t = \left( \frac{Q_{w,t}}{Q_{i,t}} \right) \times 100 \)

3.5.4 Determination of moisture content of the drying product

On wet basis, moisture content of the product is determined as:

\[ MC \, (\% \, wb) = 100 \frac{m_w}{m_w + m_d} \]  

...(3.12)

where \( m_w \) is mass of water content (g)

\( m_d \) is mass of dry matter (g)

On dry basis (db), moisture content of the product is determined as:

\[ MC \, (\% \, db) = 100 \frac{m_w}{m_d} \]  

...(3.13)

3.5.5 Determination of thermal efficiency during drying experiment

Thermal efficiency during drying experiment (\( \eta_d \)) is calculated using the following relation:

\[ \eta_d = \frac{2270 M_{w,rem}}{Q_{i,t}} \]  

...(3.14)

where \( M_{w,rem} \) is amount of water removed (kg)

3.6 Uncertainties in Measurements

Although efforts have been made to record data accurately, still some variations are induced due to instrumentation errors, least count and human limitations. Table 3.3 lists maximum uncertainties in recording different parameters.

**Table 3.3 Maximum uncertainties in measurements.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Maximum uncertainty in measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature measurement with thermometer</td>
<td>± 0.5°C</td>
</tr>
<tr>
<td>Weight measurement of the product</td>
<td>± 1 g</td>
</tr>
<tr>
<td>Inlet air temperature</td>
<td>± 0.5°C</td>
</tr>
<tr>
<td>Flow measurement</td>
<td>± 0.000278 kg/s</td>
</tr>
</tbody>
</table>

Uncertainty in calculations have been determined using the following relation (Holman 2014):

\[ \omega_R = \left[ \left( \frac{\partial R}{\partial x_1} \omega_1 \right)^2 + \left( \frac{\partial R}{\partial x_2} \omega_2 \right)^2 + \ldots + \left( \frac{\partial R}{\partial x_n} \omega_n \right)^2 \right]^{1/2} \]  

...(3.15)

where \( \omega_R \) is uncertainty in function R

\( x_1, x_2, \ldots x_n \) are independent parameters on which function R depends

\( \omega_1, \omega_2, \ldots \omega_n \) are uncertainties in respective independent parameters
Maximum uncertainties in hourly input energy and hourly energy in exit air were found to be 83.9 kJ and 79.8 kJ respectively. For total input and total exit energy during experiments, the maximum uncertainties were 268.6 kJ and 249 kJ respectively. The detailed uncertainty analysis is given as Appendix B.
The performance of the system was evaluated in three sets of experiments conducted for drying of carrot under pre-defined conditions. Each set comprises of drying at air temperatures of 55, 60, 65 and 70°C at constant mass flow rate of 0.025 kg/s.

4.1 Drying without thermal storage

This set of experiments was performed in last week of February 2016 with the aim of getting drying characteristics at different temperatures and to study the effect of drying temperature on energy consumption.

Figs. 4.1 and 4.2 present drying curves for these experiments. Initial moisture content on dry basis was 900.51, 919.69, 885.35 and 856.37% for drying experiments carried out at inlet air temperatures of 55, 60, 65 and 70°C. This variation in initial moisture levels was due to carrots purchased on different days for conducting each run. At higher inlet air temperatures, drying proceeds faster when compared to lower ones. Faster drying at higher temperatures is due to more available energy for water removal, increased water holding capacity of air and high vapour pressure difference between product and the drying air. Similar results have been obtained for red pepper by Akpinar et al (2003) and pomegranate arils by Mundada et al (2010).

Fig. 4.3 shows temperatures of the exit air for all four experiments. Individual curves chase increasing profiles with passage of time. It indicates energy is utilised to lower extent as the moisture content of the product reduces. The reason for this effect can be had from preliminary testing which is given in Appendix A. At any particular time, say hour 3, the temperature of exit air at 70°C drying temperature is the highest, followed by 65, 60 and 55°C respectively. The outcome replicates at any instant. It is due to difference in moisture levels of the product at that time. Moisture content was 77.18, 79.86, 82.8 and 83.04% respectively for highest to lowest drying temperatures. As already stated, lower moisture content of the product leads to reduced utilisation of energy which establishes this trend. There is an abrupt rise in temperature upto initial half an hour as shown in Fig. 4.3 for each experiment which is due to initial heating of the drying chamber as well as the product. Further it is observed that the outlet temperature at end of the drying process is lowest in case of drying at 55°C which goes on rising with increase in drying temperature. Highest value of 64.5°C is attained during dehydration at 70°C.

Ambient temperatures are plotted in Fig. 4.4. The ambient temperature peaked around noon with minimum value occurring in the morning. Variations were recorded to be 17.5-27.5, 17.5-26.5, 16-26 and 21-27°C for drying at 55, 60, 65 and 70°C respectively. Similarly variations in ambient relative humidity were observed to be 41.5-79.5, 52.25-81, 53.5-79.75 and 37.5-64.5% respectively for these experiments.
Fig. 4.1 Moisture content on dry basis for carrot drying at different inlet air temperatures.

Fig. 4.2 Moisture content on wet basis for carrot drying at different inlet air temperatures.
Fig. 4.3 Exit air temperature during carrot drying at different inlet air temperatures.
Fig. 4.4 Ambient temperature during carrot drying at different inlet air temperatures.

Fig. 4.5 Ambient relative humidity (RH) during carrot drying at different inlet air temperatures.
Total input heat energy for drying carrots up to moisture content 10% (wb) at 55, 60, 65 and 70°C was 36772.3, 37870.9, 37882.6 and 36181.7 kJ respectively. Corresponding values for exit air energy were 16159.5, 16915.5, 18817.3 and 16702.9 kJ. Thus the fraction of heat energy in exit air came out to be 43.94, 44.67, 49.67 and 46.2% respectively for these drying temperatures. However there were some differences in drying parameters like ambient temperature, ambient relative humidity, initial and final moisture content which limit the precise one on one comparison. These parameters are listed in Table 4.1. So as to remove the effect of differences in ambient temperature and moisture content, calculations were normalised to initial moisture content 89.5% (wb) and reference ambient temperature of 23.78°C.

Table 4.1: Average ambient temperature, average ambient relative humidity as well as initial moisture content for carrot drying at different inlet air temperatures.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Drying temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>55</td>
</tr>
<tr>
<td>Initial moisture content (% wb)</td>
<td>90</td>
</tr>
<tr>
<td>Average ambient temperature (°C)</td>
<td>23.75</td>
</tr>
<tr>
<td>Average ambient relative humidity (%)</td>
<td>63.19</td>
</tr>
</tbody>
</table>

Fig. 4.6 shows energy calculations normalised to moisture content reduction of 89.5% (wb) to 10% (wb) and ambient temperature of 23.78°C. Input heat energy increased with rise in temperature up to 65°C and after that it decreased. Amount of heat energy in exit air as well as percentage of heat in exit air followed the same route. But before some conclusion is drawn, the remaining factor also needs to be considered. Since normalising ambient relative humidity accurately is beyond the scope of this research work, so it is only applied by generalisation. Thus the effect of normalising RH to that of 60°C on energy consumption of others would be slightly decreasing, slightly increasing and moderately increasing for 65, 55 and 70°C respectively. Taking into consideration all of the above factors, no clear effect of temperature can be seen on energy consumption and utilisation as the variations are in narrow range.
It can be concluded that drying temperature does not have strong effect on utilisation of energy during drying process. The reduced time in higher temperature drying is offset by high energy demand per unit time, thus overall energy requirements remain almost same. The performance testing of solar dryer carried for constant temperature drying can also be extended to variable temperature drying by considering its mean temperature. It is because the fraction of energy in exit air is not a strong function of inlet air temperature, rather it depends on moisture content of the product. Thus same amount of energy will be wasted while drying at fluctuating temperature and constant temperature, provided mean temperature for former drying equals the fixed value of latter and flow rates are identical. Another conclusion that can be drawn is that at same flow rate, amount of thermal storage material needed for heat recovering purpose would be lower for drying at higher temperature. This is because since heat energy in exit air is same for all drying temperatures, and the available temperature difference is more for high temperature, it lowers quantity of material required to store same amount of heat.

Thermal efficiencies were also calculated for drying carrots upto 10% moisture content on wet basis. These came out to be 32.12, 31.26, 31.12 and 32.45% at drying temperatures of 55, 60, 65 and 70°C with average value of 31.74%.
4.1.2 Drying with thermal storage

Experiments were conducted in the first two weeks of March 2016 to see the effect of thermal storage on performance of the dryer. Experiments were started at 10 a.m. and air flow was reversed after 6 hours to utilise stored energy.

Fig. 4.7 shows moisture content and temperatures at various locations of the setup for drying process conducted at 70°C. As shown, air inlet temperature remains constant at 70°C during first 6 hours. After this period when flow was reversed and electric heaters were switched off, temperature of inlet air changed. This time, boost in temperature was provided by thermal storage. Air temperatures were recorded at different locations of fitting to track heat losses. It has been observed that initially there is some difference between temperature of air just at top port of the thermal storage and inlet of blower which later on diminishes. This effect is caused by pre-heating of the pipes. Initially pipes are at ambient temperature. The heat absorbed by pipes to attain equilibrium with passing air results in drop in temperature of air. Once steady state is attained, temperatures at two locations become equal. Further there is difference in temperature of air at inlet to blower and inlet to drying chamber, i.e. the temperature at inlet to the drying chamber is higher than that at inlet to the blower. This boost is believed to be provided by the blower which uses sucked air for cooling electric motor. Moreover the energy required for this increase in temperature was found to be within wattage of the blower, somewhat less, and the temperature rise was nearly consistent. These arguments strengthen the proposed reason. The sharp decline in drying temperature for first half an hour during reverse flow can be seen which is due to air distributor. The heat was stored in air distributor while constant temperature drying which was released during this stage. The temperature profile of air leaving recovery unit followed typical trend observed in thermal storage units. The process was terminated after 2.5 hours of reversed flow as the temperature rise provided by storage reduced to just 7°C which was not enough for carrying out dehydration. The remaining drying was continued on next day. Moisture regain was observed during night. It took only 1 hour to achieve moisture content less than 10% (wb).

Drying curves for all the four trials are plotted in Fig. 4.8. First 6 hours follow constant temperature drying for each case. This phase is shown with solid lines. The profile of the curves upto this point are similar to previous set of the experiments conducted on carrots. Dotted lines represent the reversed flow. Its duration had been 0.5, 1.75, 3.25 and 2.5 hours respectively for drying at 55, 60, 65 and 70°C. In this phase, temperature of inlet air varied according to the boost provided by the heat unit. Since reversed flow time as well as temperature varies in each case, data can not be related directly among experiments. Even it can not be compared with previous set of experiments (Set A). The curves can only be used to monitor the drying processes individually. Breaks are observed between moisture content of the product for all cases, although these are hard to see on graph due to scale. These breaks are due to moisture lost or gain during night.
Temperature of air heated by the thermal storage unit available at blower can be traced from Fig. 4.9. This is the effective temperature which is available as useful gain. Energy savings are computed based on this temperature. It can be seen that drying at higher temperature leads to higher recovery temperature. It is due to more amount of energy wasted during same time in case of elevated temperatures, which heats the stones to larger extent. Also the recovery time is longer for higher temperatures. This is because for higher temperature of delivered air, minimum required temperature difference of 7°C prevails for more time (Fig. 4.10). However there is an exceptional case of 65°C drying experiment which lasts longer than that of 70°C. It is due to lower ambient temperature on that day. Due to rain, ambient temperature dropped significantly which allowed 7°C temperature difference to last longer as well as higher available energy, thus more energy was extracted. The repetition of this experiment was avoided due to diminishing availability of the crop and changing environmental conditions which may result in improper comparison between earlier and later experiments. But an interesting side was revealed by this anomaly that recovery performance is a big function of ambient conditions. Peak in air temperature at inlet of the blower is attained after 15 minutes from start in most cases. It is due to some amount heat absorbed at the start of the process for heating of pipes.

Input energy, energy in exit air and recovered energy during the experiments are given in Fig. 11. It is clear that input energy, unlike previous set of trials, is not consistent. The difference is due to ambient conditions, extent of drying, initial moisture content plus moisture gain/removal during nights. Heat energy in exit air for first 6 hours, which was used for charging rock bed storage, reduces with fall in drying temperature which is quite expected. Net amount of energy recovered generally increased with rise in drying temperature except for 65°C which is understood.

Fig. 4.12 illustrates percentage of heat load taken by thermal storage. It was calculated by dividing energy recovered with total input energy which also includes recovered heat. It is evident that higher drying temperatures enable more energy savings using this setup. The higher percentage of heat load for 65°C is due to lower ambient temperature on that day as mentioned above.

Thermal efficiencies for drying experiments at 55, 60, 65 and 70°C were 41.25, 42.36, 37.13 and 41.04%. Thus the average thermal efficiency was 38.08% which is 6.34% higher than drying without thermal storage.
Fig. 4.7 Inlet air temperature, ambient temperature, temperature at exit of storage, temperature at inlet of blower and moisture content during carrot drying at 70°C with thermal storage.
Fig 4.8 Variation in moisture content during carrot drying with thermal storage.
Fig 4.9 Temperature of air at blower inlet and ambient temperature for different experiments during carrot drying with thermal storage.

Fig 4.10 Rise in temperature (ΔT) of drying air by thermal storage measured at blower inlet during carrot drying with thermal storage.
Fig 4.11 Total input energy, energy in exit air and recovered energy for carrot drying with thermal storage.

Fig 4.12 Energy savings at different drying temperatures by using thermal storage at exit of the drying chamber.
4.1.3 Recovery supplemented by auxiliary heating

This analysis was performed in third week of March 2016. Fig. 4.13 shows drying curves for these experiments. These are similar to normal drying curves. The reason for similarity is that drying occurred at constant temperature. The slight variation in drying time as compared to first set of drying performed on carrots is due to initial moisture content, structural changes as the crop was migrating towards off season and different ambient conditions like RH.

Fig. 4.14 shows effective temperature of air heated by thermal storage. The trend is similar to previous one, that is, higher drying temperatures lead to higher recovery temperatures. However, there is one difference in this case from the previous one. Now recovery time is not controlled by thermal storage, but dependent on total drying time. In other words, recovery continued until the product was dried to required moisture content. Recovery time for higher temperature drying was short in comparison with dehydration at lower temperatures because the drying occurred faster in former conditions. At lower temperature drying, even two degree temperature rise was being utilised which was retarded in previous experimental structure. On the other hand, drying at higher temperatures terminated earlier, thus there was still some energy left in the rocks which remained unrecovered.

Figs. 4.15 and 4.16 compare recovered energy for different temperatures. As usual, waste energy under consideration and recovered energy follow the increasing path with rise in drying temperature. It should also be noted that the results are influenced by ambient temperatures, which affects availability of stored energy. So direct comparison should be carefully done compensating for those differences.

Thermal efficiencies for these experiments were 42.42, 41.72, 38.44 and 36.73% in case of drying at 55, 60, 65 and 70°C respectively. Average thermal efficiency came out to be 37.22% which is almost same as that of drying with thermal storage only. This may be because although the auxiliary heating has advantage to recover heat up to lower temperatures, but the air from thermal storage without auxiliary heating still has capability to remove some moisture. So due to combined effect of these factors, thermal efficiency for the two cases remains almost same.
Fig 4.13 Variation of moisture content during carrot drying with thermal storage and auxiliary heating.

Fig 4.14 Temperature of air at blower inlet and ambient temperature for different experiments during carrot drying with thermal storage and auxiliary heating.
Fig 4.15 Total input energy, energy in exit air and recovered energy corresponding to different inlet air temperatures during carrot drying with thermal storage and auxiliary heating.

Fig 4.16 Energy savings at different drying temperatures by using thermal storage at exit of the drying chamber and auxiliary heating.
4.2 Comparison of auxiliary heating vs reversed flow mode only

It can be seen that auxiliary heating enable more energy to be extracted. Strange weather conditions for 65°C as discussed earlier leads to unfair comparison for that case. For all other experiments, the trend is clear. Care should be taken while making comparison even at same temperature since amount of recovered energy depend on ambient conditions. For instance, ambient temperature in both cases of 70°C drying experiments were significantly different. Surrounding temperature while heat recovery with auxiliary heating was lower than that of without auxiliary heating, which automatically translates to more available energy despite being stones at similar temperature.

![Comparison of different heat recovery methods.](image-url)

**Fig 4.17** Comparison of different heat recovery methods.
Chapter 5 DEVELOPMENT OF COMPUTER MODEL

Computer program has been developed in Freemat\(^1\) (version 4.0) which can be used to theoretically predict the performance of the solar dryer having thermal storage at exit of drying chamber. Fig. 5.1 shows line diagram of the setup which is same as that one used in practical testing except that electric heaters are replaced by solar air heaters. Thus the computer model evaluates the same setup for variable temperature drying. The effect of thermal storage dimensions can also be studied through this model and can be optimised for best performance. In addition to theoretical relations available in literature, experimental data has been used wherever possible. The flowchart of computer program is shown in Fig. 5.2 while lines of code are given in Appendix C. Mathematical relations as well as data used in development of computer model are given in upcoming section which are followed by simulation results.

\[\text{Fig. 5.1} \quad \text{Line diagram of setup used for computer simulation.}\]

\(^1\) Freemat is free and open source alternative to MATLAB. Coding structure of both is compatible with each other with only slight modifications required in some cases.
5.1 Mathematical Relations

5.1.1 Ambient temperature

Hourly ambient temperature is calculated by fitting sine curve between maximum and minimum temperatures for the day assuming that minimum temperature occurs at 2 a.m. while maximum temperature occurs at 2 p.m. Average maximum and minimum values for representative day of the month were provided by School of Climate Change and Agricultural Meteorology, PAU, Ludhiana.

\[ T_{a,\text{time}} = T_{a,\text{day}} + A \sin [(t_{\text{absolute}} - 8) (\pi/12)] \] ... (5.1)

\[ T_{a,\text{day}} = \frac{(T_{a,\text{max}} + T_{a,\text{min}})}{2} \] ... (5.2)

\[ A = \frac{(T_{a,\text{max}} - T_{a,\text{min}})}{2} \] ... (5.3)

where \( T_{a,\text{time}} \) is ambient temperature at any time

\( T_{a,\text{day}} \) is average ambient temperature for the day

\( T_{a,\text{max}} \) is maximum ambient temperature for the day

\( T_{a,\text{min}} \) is minimum ambient temperature for the day

5.1.2 Total solar radiation

Solar radiation intensity for Ludhiana has been assumed to be same as Delhi. Table 5.1 shows hourly diffuse and total solar radiation on flat surface for the month of March (Sukhatme 1990). From these values, solar radiation on inclined surface has been calculated using the following relations (Tiwari 2013):

\[ I_t = I_b R_b + I_d R_d + \rho_g R_r (I_b + I_d) \] ... (5.4)

\[ I_b = I - I_d \] ... (5.5)

\[ R_b = \cos \theta_i / \cos \theta_z \] ... (5.6)

\[ R_d = (1 + \cos \beta) / 2 \] ... (5.7)

\[ R_r = (1 - \cos \beta) / 2 \] ... (5.8)

\[ \cos \theta_i = \cos \phi \cos \psi \cos \Omega + \sin \psi \sin \phi \] ... (5.9)

\[ \cos \theta_z = (\cos \phi \cos \beta + \sin \phi \sin \beta \cos \gamma) \cos \psi \cos \Omega + \cos \psi \sin \Omega \sin \beta \sin \gamma \]

\[ + \sin \psi (\sin \phi \cos \beta - \cos \phi \sin \beta \cos \gamma) \] ... (5.10)

\[ \psi = 23.24 \sin [(360/365)(284 + n)] \] ... (5.11)

where

- \( I_t \) is total solar radiation intensity on a tilted surface
- \( I \) is total solar radiation intensity on horizontal surface
- \( I_b \) is beam radiation on horizontal surface
- \( I_d \) is diffused radiation on horizontal surface
- \( R_b \) is tilt factor for beam radiation
- \( R_d \) is tilt factor for diffused radiation
- \( R_r \) is tilt factor for reflected radiation
- \( \rho_g \) is reflectivity of the ground
θ, is incidence angle for solar radiation on inclined surface
θ, is zenith angle
β is angle of inclination of the surface
φ is latitude of location
ψ is declination angle
Ω is hour angle
γ is surface azimuth angle
n is day of the year counted from 1\textsuperscript{st} January

<table>
<thead>
<tr>
<th>Time</th>
<th>Total solar radiation (kW/m\textsuperscript{2})</th>
<th>Diffused radiation (kW/m\textsuperscript{2})</th>
</tr>
</thead>
<tbody>
<tr>
<td>6 a.m.</td>
<td>0.001</td>
<td>0.001</td>
</tr>
<tr>
<td>7 a.m.</td>
<td>0.066</td>
<td>0.043</td>
</tr>
<tr>
<td>8 a.m.</td>
<td>0.260</td>
<td>0.113</td>
</tr>
<tr>
<td>9 a.m.</td>
<td>0.475</td>
<td>0.159</td>
</tr>
<tr>
<td>10 a.m.</td>
<td>0.655</td>
<td>0.189</td>
</tr>
<tr>
<td>11 a.m.</td>
<td>0.781</td>
<td>0.210</td>
</tr>
<tr>
<td>12 p.m.</td>
<td>0.845</td>
<td>0.222</td>
</tr>
<tr>
<td>1 p.m.</td>
<td>0.844</td>
<td>0.226</td>
</tr>
<tr>
<td>2 p.m.</td>
<td>0.769</td>
<td>0.215</td>
</tr>
<tr>
<td>3 p.m.</td>
<td>0.635</td>
<td>0.197</td>
</tr>
<tr>
<td>4 p.m.</td>
<td>0.458</td>
<td>0.163</td>
</tr>
<tr>
<td>5 p.m.</td>
<td>0.249</td>
<td>0.113</td>
</tr>
<tr>
<td>6 p.m.</td>
<td>0.164</td>
<td>0.041</td>
</tr>
<tr>
<td>7 p.m.</td>
<td>0.001</td>
<td>0.001</td>
</tr>
</tbody>
</table>

5.1.3 Temperature rise by solar air heater
Temperature rise of air by flat plate solar air heater is found by determining useful heat gain and then dividing it by heat capacity of air (Sukhatme 1990).

\[ q_u = F_R A_p \left[ S - U_l (T_{fi,c} - T_a) \right] \]  \hspace{1cm} \text{(5.12)}

\[ S = \tau \alpha I_l \]  \hspace{1cm} \text{(5.13)}

\[ F_R = \left( \frac{m_{fp} C_{pa}}{U_l A_p} \right) \left\{ 1 - \exp \left\{ - F' U_l A_p / m_{fp} C_{pa} \right\} \right\} \]  \hspace{1cm} \text{(5.14)}

\[ F' = (1 + U_l / h_c)^{\frac{1}{4}} \]  \hspace{1cm} \text{(5.15)}

\[ h_e = h_{fb} + (h_r + h_{fb}) / (h_r + h_{fb}) \]  \hspace{1cm} \text{(5.16)}

\[ h_r = 4 \sigma T_{fm}^{3/4} \left\{ (1 / c_p) + (1 / c_b) - 1 \right\} \]  \hspace{1cm} \text{(5.17)}

\[ h_{fp} = h_{fb} = Nu (k_c / D_c) \]  \hspace{1cm} \text{(5.18)}
\[ \text{Nu} = 0.0158 \text{ Re}^{0.8} \]  
\[ \text{Re} = \rho_a V_f D_e / \mu \]  
\[ V_f = m_f / (\rho_a L_b H_s) \]  
\[ T_{\text{rise}} = q_u / m_f C_{pa} \]  
\[ T_{fo} = T_a + T_{\text{rise}} \]  

where \( q_u \) is useful heat gain from the collector
\( F_R \) is collector heat removal factor
\( A_p \) is area of absorber plate
\( S \) is solar flux absorbed in absorber plate
\( U_l \) is overall heat loss coefficient
\( T_{in,c} \) is inlet air temperature to the solar air heater
\( \tau \) is transmissivity of glass cover
\( \alpha \) is absorptivity of the absorber plate
\( m_f \) is mass flow rate of air
\( C_{pa} \) specific heat of the air
\( F' \) is collector efficiency factor
\( h_e \) is effective heat transfer coefficient between the absorber plate and air stream
\( h_{cb} \) is convective heat transfer coefficient between absorber plate and air stream
\( h_r \) is equivalent radiative heat transfer coefficient
\( h_{fb} \) is convective heat transfer coefficient between bottom plate and air stream
\( T_{in} \) is mean air temperature
\( \epsilon_p \) is emissivity of the absorber plate surface
\( \epsilon_b \) is emissivity of the bottom plate surface
\( k_a \) is thermal conductivity of air
\( D_e \) is equivalent diameter of the duct
\( \rho_a \) is density of air
\( \mu \) is viscosity of air
\( V_f \) flow velocity of air in solar air heater
\( L_b \) is length of solar air heater
\( H_s \) is space between bottom plate and absorber plate
\( T_{\text{rise}} \) is rise in temperature of air by solar air heater
\( T_{fo} \) is temperature of air leaving solar air heater

For program calculations, \( T_{in} \) is taken to be equal to \( T_a \) as it does not affect results significantly. \( U_l \), which is constant for a design, is assumed \( 7 \text{ W/m}^2\text{-K} \) for present flat plate solar air heater.
5.1.4 Drying rate and moisture content

Firstly correlations are developed using the experimental data reported in chapter 4. From these relations, instantaneous drying rate is determined in the program. Following equation has been used to determine drying rate (DR) for actual drying process.

\[
DR = \frac{M_{w,\text{rem}}}{(DM) (t_{dh})} \quad \ldots (5.24)
\]

\[
M_d = M_{p,\text{ini}} - MC_{\text{ini, wb}} \left(\frac{M_{p,\text{ini}}}{100}\right) \quad \ldots (5.25)
\]

where \(M_{w,\text{rem}}\) is amount of water removed
\(t_{dh}\) is drying time required for water removal
\(M_{p,\text{ini}}\) is initial weight of the crop
\(MC_{\text{ini, wb}}\) is initial moisture content on wet basis

Equations 5.26 - 5.30 are used for predicting purposes. \(R^2\) and standard error for these fittings are listed in Table 5.1.

\[
DR_{70} = 1.854 \text{ for } MC_{\text{wb}} > 87.2\%
\]

\[
DR_{70} = -0.1126102 + 0.01891595 (MC_{\text{wb}}) - 0.000388011 (MC_{\text{wb}})^2 + 0.00000487 (MC_{\text{wb}})^3 \text{ for } 87.2\% > MC_{\text{wb}} > 10\% \quad \ldots (5.26)
\]

\[
DR_{65} = 1.6933 \text{ for } MC_{\text{wb}} > 85\%
\]

\[
DR_{65} = -0.1078560 + 0.01736696 (MC_{\text{wb}}) - 0.000351231 (MC_{\text{wb}})^2 + 0.0000044009 (MC_{\text{wb}})^3 \text{ for } 85\% > MC_{\text{wb}} > 10\% \quad \ldots (5.27)
\]

\[
DR_{60} = 1.486 \text{ for } MC_{\text{wb}} > 86.2\%
\]

\[
DR_{60} = -0.1127181 + 0.01730623 (MC_{\text{wb}}) - 0.000347401 (MC_{\text{wb}})^2 + 0.0000027656 (MC_{\text{wb}})^3 \text{ for } 86.2\% > MC_{\text{wb}} > 10\% \quad \ldots (5.28)
\]

\[
DR_{55} = 1.41 \text{ for } MC_{\text{wb}} > 85\%
\]

\[
DR_{55} = -0.07963629 + 0.01284641 (MC_{\text{wb}}) - 0.000220256 (MC_{\text{wb}})^2 + 0.0000027656 (MC_{\text{wb}})^3 \text{ for } 85\% > MC_{\text{wb}} > 10\% \quad \ldots (5.29)
\]

Here \(DR_{70}\), \(DR_{65}\), \(DR_{60}\) and \(DR_{55}\) are drying rates at 70, 65, 60 and 55°C and \(MC_{\text{wb}}\) is moisture content on wet basis

<table>
<thead>
<tr>
<th>Drying temperature (°C)</th>
<th>(R^2)</th>
<th>Standard Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>55</td>
<td>0.9989</td>
<td>0.01492</td>
</tr>
<tr>
<td>60</td>
<td>0.9994</td>
<td>0.01366</td>
</tr>
<tr>
<td>65</td>
<td>0.9991</td>
<td>0.02088</td>
</tr>
<tr>
<td>70</td>
<td>0.9995</td>
<td>0.01852</td>
</tr>
</tbody>
</table>

The drying rate at intermediate temperatures is determined by assuming that it is linearly dependent on drying temperature. For example, drying rate for temperature between 65 and 70°C is computed as:
DR = DR_{70} - [(DR_{70} - DR_{65}) / 5] (70 - T_d) \quad \ldots (5.30)

Similarly, DR for temperature lying between other ranges can be determined.

Apart from above relations, some other equations are also used which are:

\[ M_{w,\text{ini}} = M_{p,\text{ini}} - DM \quad \ldots (5.31) \]
\[ MC_{db} = 100 \frac{M_p}{DM} \quad \ldots (5.32) \]
\[ M_w = \frac{(MC_{wb}) \times (DM)}{100-\text{MC}_{wb}} \quad \ldots (5.33) \]
\[ M_{w,\text{rem}} = (DR) \times (t_d, h) \quad \ldots (5.34) \]
\[ f_{\text{exit}} = 93 - 0.82 \times MC_{wb} \quad \ldots (5.35) \]
\[ T_{ex} = T_a + \left[ (T_d - T_a) \times \left( \frac{\text{waste}_{pc}}{100} \right) \right] \quad \ldots (5.36) \]

where

- MC_{db} is moisture content on dry basis
- T_{ex} is temperature of air leaving drying chamber
- f_{\text{exit}} is percentage of heat in exit air

### 5.1.5 Thermal storage temperatures

Thermal storage temperatures are computed per second time step with the help of following formulae (Sukhatme 1990):

\[ T_{in,s} = T_{ex,i} + \left[ \frac{(T_{ex,i} - T_{ex,i+1})}{60} \right] \times t_{\text{minutes}} \quad \ldots (5.37) \]
\[ q_{\text{element}} = h_v \times \text{(element Length) \times (area of cross section) \times (T_f - T_s)} / 1000 \quad \ldots (5.38) \]
\[ h_v = 650 \times \left( \frac{G}{D_s} \right)^{0.7} \quad \ldots (5.39) \]
\[ G = m_f \times \text{area of cross section} \quad \ldots (5.40) \]
\[ T_{\text{rise,se}} = \frac{q_{\text{element}}}{\left[ (m_{\text{element}} \times C_{ps}) \right]} \quad \ldots (5.41) \]
\[ m_{\text{element}} = \text{(Bulk Density) \times (element Length) \times (area of cross section)} \quad \ldots (5.42) \]
\[ T_{f,\text{leaving}} = T_{f,\text{entering}} - T_{\text{rise,se}} \quad \ldots (5.43) \]

where

- T_{in,s} is inlet temperature to storage
- T_{ex,i} is temperature of air leaving drying chamber at start of hourly interval
- T_{ex,i+1} is temperature of air leaving drying chamber at end of hourly interval
- t_{\text{minutes}} is time passed in minutes
- q_{\text{element}} is heat transferred to a bed element
- h_v is volumetric heat transfer coefficient
- T_f is fluid temperature at any instant
- T_s is temperature of stones at any instant
- G is mass velocity
- D_s is average diameter of stones
- T_{\text{rise,se}} is temperature rise of bed element in consideration
- m_{\text{element}} mass of bed element in consideration
- T_{f,\text{leaving}} temperature of air leaving a bed element
- T_{f,\text{entering}} is temperature of air entering a bed element
Fig. 5.2 Flowchart of computer program for predicting drying rate curve using air from solar air heater and combining heat recovery system.
Fig. 5.3 Flowchart of sub-routine for calculation of total solar radiation intensity on inclined surface.
Fig. 5.4 Flowchart of sub-routine for calculation of air temperature at outlet of solar air heater.
Fig. 5.5 Flowchart of sub-routine for calculations of drying rate and exit air temperature.
Fig. 5.6 Flowchart of sub-routine for calculations of temperature of thermal storage and temperature of air leaving thermal storage.
5.2 Results of computer model

The system shown in Fig. 5.1 was simulated for drying carrots on a representative day in the month of March. The parameters and inputs used for simulation are listed in Table 5.3. To see the benefit of using thermal storage, another comparative simulation was performed in which there is no thermal storage.

### Table 5.3 Parameters fixed for carrot drying simulation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial weight of the crop</td>
<td>5853 g</td>
</tr>
<tr>
<td>Initial moisture content</td>
<td>90 % wb</td>
</tr>
<tr>
<td>Maximum ambient temperature during the day</td>
<td>27°C</td>
</tr>
<tr>
<td>Minimum ambient temperature during the day</td>
<td>13°C</td>
</tr>
<tr>
<td>Mass flow rate of air</td>
<td>0.025 kg/s</td>
</tr>
<tr>
<td>Length of solar air heater</td>
<td>2.2 m</td>
</tr>
<tr>
<td>Width of solar air heater</td>
<td>1.1 m</td>
</tr>
<tr>
<td>Space between absorber and back plate of solar air heater</td>
<td>1 cm</td>
</tr>
<tr>
<td>Transmissivity of glass cover</td>
<td>0.95</td>
</tr>
<tr>
<td>Absorptivity of absorber plate</td>
<td>0.95</td>
</tr>
<tr>
<td>Reflectivity of ground</td>
<td>0.2</td>
</tr>
<tr>
<td>Top loss coefficient (U_t)</td>
<td>6.2 W/m²·K</td>
</tr>
<tr>
<td>Bottom loss coefficient (U_b)</td>
<td>0.8 W/m²·K</td>
</tr>
<tr>
<td>Rock diameter</td>
<td>6.5 cm</td>
</tr>
<tr>
<td>Bed element count</td>
<td>10</td>
</tr>
<tr>
<td>Reverse time</td>
<td>After 6 hours</td>
</tr>
<tr>
<td>Drying terminating condition</td>
<td>Drying temperature falls below 45°C</td>
</tr>
</tbody>
</table>

The results are presented in Figs. 5.7-5.9. It can be seen that drying rate increases with time for the first three hours (Fig. 5.7). It is because of a rise in inlet temperature for up to 1 p.m. Thereafter, drying rate starts decreasing with a fall in solar air heater outlet temperature. Drying with thermal storage proceeds for a longer duration. Drying rate in the case of thermal storage is also higher than without storage after 6 hours. It is because thermal storage results in a higher temperature of drying air (Fig. 5.8). For 7th hour, while mean air temperature at the inlet of the drying chamber in case of no thermal storage was 48.23°C, it was 57.73°C when thermal storage was used.

Fig. 5.9 shows rock-bed temperature for different locations at different times during carrot drying simulation. Each element temperature increases up to 6 hours, that is until charging occurs. Thereafter it starts falling as heat extraction takes place. Temperature at the top side of the bed is also higher than bottom side, which shows thermal stratification.
Fig. 5.7 Drying rate (DR) and moisture content (MC) for simulated carrot drying with and without thermal storage.

Fig. 5.8 Inlet air temperature for crop dryer with and without thermal storage.
Fig. 5.9 Predicted storage bed temperatures of different elements during carrot drying simulation (Charging mode 0-6 hours; Discharging mode after 6 hours).
Chapter 6 SUMMARY

For improving thermal performance of conventional forced circulation solar dryer, a novel concept of using thermal storage at the exit of drying chamber has been proposed. Earlier designs of solar dryers integrated with thermal storage incorporate storage unit before the drying chamber. This reduces energy input to drying chamber during day time. The stored energy is released afterwards for drying. The present concept extracts heat from air leaving drying chamber, which otherwise goes waste. Thus, it provides the benefit of extended drying time and higher energy utilisation. Keeping this in view, the present study was undertaken with following specific objectives:

1. To investigate effect of sensible heat storage at exit of drying chamber of a conventional forced circulation crop dryer on thermal efficiency.
2. To investigate effect of auxiliary heating coupled with sensible heat storage at exit of drying chamber of a conventional forced circulation crop dryer on thermal efficiency.
3. To develop a computer model for predicting drying rate curve using hot air from solar air heater of conventional forced circulation solar crop dryer with storage.

For achieving the above objectives, an experimental setup consisting mainly of an air heating arrangement, drying chamber, rock-bed thermal storage and electric air blower was fabricated. For comparative evaluation of different experiments, heated air to drying chamber was supplied at constant temperature using electric heater instead of using solar air heater. Preliminary experiments were conducted to determine the heat energy in exit air for conventional dryer and for estimation of reasonable storage size. During preliminary testing, the heat energy in exit air was found to increasing with drying time. The average heat energy at exit of drying chamber varied between 48.21-61.7% of the heat energy at inlet of drying chamber. The thermal storage selected was rock-bed with total weight 275 kg.

To determine the amount of heat recovered by thermal storage, with and without auxiliary heating, the experiments were conducted for drying of carrot at air mass flow rate of 0.025 kg/s and drying chamber inlet air temperature of 55, 60, 65 and 70°C.

On the basis of above results, a computer model was developed to predict the drying rate of carrot using forced circulation solar dryer for any geographical location has also been developed. The following conclusions have been drawn from the present study:

- The heat energy at inlet to drying chamber of conventional forced circulation solar dryer increases with increase in air flow rate. The heat energy at inlet of drying chamber for drying of turnips increased from 28570 to 32714 kJ as flow rate increased from 0.0167 to 0.0317 kg/s.
- The sensible heat leaving the drying chamber of conventional forced circulation solar dryer
increases with increase in air flow rate. The heat energy at exit of drying chamber for drying of turnips increased from 13773 to 20185 kJ as flow rate increased from 0.0167 to 0.0317 kg/s.

• Fraction of heat in exit air is more at higher air flow rates. During drying of turnips, this fraction was 48.21% and 61.7% at air flow rates 0.0167 and 0.0317 kg/s respectively.

• Heat energy at exit of drying chamber varied in very narrow range with change in temperature from 55 to 70°C for carrot drying which shows that drying temperature does not have strong effect on energy utilisation.

• By using thermal storage at exit of the drying chamber, energy in exit air can be recovered. This technique is more effective at higher drying temperature. Energy savings were 1.08% and 7.28% for carrot drying at 55 and 70°C respectively.

• The effect of using auxiliary heating during heat recovery was found to be beneficiary. Recovery continued even at lower temperature rise due to thermal storage, thus overall more amount of energy was extracted. During carrot drying at 55°C, 5.2% energy savings were achieved by using auxiliary heating while this value was only 1.08% without auxiliary heating.

For future work, optimisations which were left in this study due to limitations imposed by time as well as resources are recommended. For instance, thermal storage can mounted above drying chamber to reduce bends in pipe which saves pumping power. Recovery for latter half of the drying process, which are left in this testing, would yield higher savings. These things should be addressed in future work. And despite being this study carried on solar air heaters, it is equally applicable to convective dryers used in industries. Hence it can be used to enhance operational efficiency of these dryers as well.
REFERENCES


properties on drying kinetics and stability of cactus/brewer’s grains mixture fermented with lactic acid bacteria. *Food and Bioproducts Processing* **94**:10-19.


