

CROP DRYING POTENTIAL OF SOLAR CHIMNEY DRYERS

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Received: December 13, 2019; Revised: April 04, 2020; Accepted: April 29, 2020

Abstract

In this study the potential of crop drying using the solar chimney was examined. The heat trapped under the solar collector is used to heat large volumes of air and this air is allowed to flow over the food product to remove the moisture and also take away from the surrounding of the product. A mathematical modeling based on the effective moisture removal rate was developed. The prediction showed good agreement with the experimental result. The solar chimney dryers for Nakhon Ratchasima province in Thailand were examined. It was found that they offer the drying air temperature that is within the allowable temperature limit for food product (15-20°C higher than the ambient temperature). The improvement of collector efficiency from 5% to 10% can increase the system potential notably. Additionally, it was found that the dryers studied here are not economically practical for electricity generation. The proposed model can be used in a design of a solar chimney dryer for any specific drying capacity.

Keywords: Solar chimney, Solar crop drying, Mathematical model, Solar dryer, Natural convection

Introduction

Approximately, 15% of the world population is undernourished (VijayaVenkataRaman *et al.*, 2012). As a result, the preservation of food and agricultural crops are essential for the economy. One of the traditional options is drying them. Generally, this is accomplished by using ovens or open-air drying. However,

drying in ovens is expensive and damages the environment. In addition, open-air drying is not only susceptible to the unpredictability of the weather but also provides poor quality products caused by contamination by dust, insect attack and infection by micro-organisms (Murthy, 2009). To enable farmers to produce

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high quality products, efficient-and-affordable drying method is necessary.

The solar chimney is a simple solar-energy-based electricity generation technology (Koonsrisuk and Chitsomboon, 2009a). A typical solar chimney power plant consists of a solar collector, a chimney and a turbine as shown in Figure 1. The solar collector is a round, transparent roof, elevated above the ground and open at its periphery. The chimney is located at the center of the collector and the turbine is installed at its base. The solar collector serves to convert solar energy coming through the roof into thermal energy of air underneath the roof. Air flows by natural convection as it enters through the roof perimeter and leaves at the chimney top. This creates an air updraft that drives the turbine to generate electricity. Several technology performance and feasibility have been studied and reported in the literature (e.g. Haaf *et al.*, 1983; Schlaich *et al.*, 1995; Bilgen and Rheault, 2005; Kashiwa and Kashiwa, 2008; Nizetic *et al.*, 2008; Fluri *et al.*, 2009; Koonsrisuk and Chitsomboon., 2009c; Zhou *et al.*, 2009, 2012; Koonsrisuk *et al.*, 2010; Lorente *et al.*, 2010; Cao *et al.*, 2011; Panse *et al.*, 2011).

As air is heated by the heat trapped in the collector, the temperature and velocity of the air increase with the distance from the collector inlet towards the collector outlet (Koonsrisuk and Chitsomboon, 2007). If agricultural products are properly distributed on the ground of the collector, the flowing heated air can remove the product moisture by convection. Warm air can contain more moisture. Consequently, the moisture holding capacity of the heated air increases with the distance. This makes solar chimney be a promising crop dryer. Ferreira *et al.* (2008) carried out the experimental study of a solar chimney for food drying. The solar chimney prototype with a chimney height and diameter of 12.3 m and 1 m, respectively, was built. Its collector, with a diameter of 25 m, was set 0.5 m above the ground. Drying tests of coffee grains, bananas and tomatoes were performed. The solar heat flux, ambient temperature, flow temperature and mass flow rate during the tests

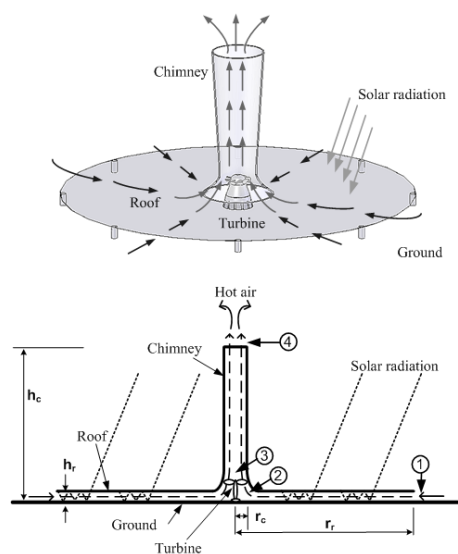


Figure 1. Schematic layout of conventional solar chimney power plant

were reported. It should be noted that there is no turbine installed in the prototype because the purpose of the study was to assess the food drying performance.

In the present paper, an attempt has made theoretically toward potential assessment of crop drying by the solar chimney system in Nakhon Ratchasima province, Thailand. Such a system is hereinafter called a solar chimney dryer. The mathematical models for solar chimney power plants proposed by Koonsrisuk and Chitsomboon (2013) and Koonsrisuk (2012) are adapted here. The model validity is justified by comparing its prediction with the experimental results from Ferreira *et al.* (2008). Then the variables influencing the system performance are identified. The results obtained here are expected to provide a promising replacement for conventional drying technologies.

Materials and Methods

Mathematical Model

In order to evaluate the crop drying potential of the solar chimney dryer, the relevant flow properties are computed. Those

flow properties are evaluated using the mathematical model proposed by Koonsrisuk and Chitsomboon (2013), which is the model for conventional solar chimneys with constant collector height. The heights at the collector inlet and outlet of the physical model in Ferreira *et al.* (2008) are different. As we will adopt the size of that model, the mathematical model in Koonsrisuk and Chitsomboon (2013) is modified by the model propose in Koonsrisuk (2012), which is the model for sloped solar chimney systems. In addition, friction losses are included in the current study (Koonsrisuk, 2013). Here are those equations:

$$\int_{col} dp = \frac{\dot{m}^2}{\rho_{ave,col}} \int \frac{dA}{A^3} - \frac{\dot{m}q''}{\rho_{ave,col} c_p T_{ave,col}} \int \frac{dA_r}{A^2} + \frac{1}{2} f \frac{\dot{m}^2}{\rho_{ave,col}} \int \frac{dr}{A^2 h_{roof}} \quad (1)$$

$$T_2 = \frac{1}{(c_p + 1.82\omega_2)} \left(\frac{\eta_{col} q'' A_r}{\dot{m}} - \frac{\dot{m}^2}{2\rho^2} \left(\frac{1}{A_2^2} - \frac{1}{A_1^2} \right) + (\omega_2 - \omega_1) \times \right. \\ \left. (h_w - 2,501.3) + (c_p + 1.82\omega_1)(T_1 - 273.15) \right) + 273.15 \quad (2)$$

$$T_3 = T_2 \left(\frac{p_3}{p_2} \right)^{\frac{\gamma-1}{\gamma}} \quad (3)$$

$$p_3 = p_4 + \frac{1}{2} (\rho_3 + \rho_4) \cdot g \cdot h_c + f \frac{\dot{m}^2}{2(\rho_3 + \rho_4) \pi^2 r_c^5} h_c \quad (4)$$

$$T_4 = T_3 - \frac{g}{c_p} h_c \quad (5)$$

$$\rho_2 = \frac{p_2}{RT_2}, \rho_3 = \frac{p_3}{RT_3}, \rho_4 = \frac{p_4}{RT_4} \quad (6)$$

$$\dot{W}_{ext} = \frac{\dot{m}}{(\rho_2 + \rho_3)/2} (p_2 - p_3) \quad (7)$$

The derivation of Equation (2) is provided in Appendix. Referring to Equation (A1) in Appendix, this model uses a simplified approach and solves the equations making use of a constant effective moisture removal rate, $\dot{m}_w$, to describe the drying rate of the whole drying product. The study assumes the product as being thermally stationary, although the product temperature continuously changes during the drying process. However, because

the temperature change rate of the crop is slow relatively, a quasi-stationary model assuming that the drying rate is constant would be applicable. Although using this simplified model does not give us the drying characteristic as a function of time, such a model reveals the drying temperature and drying capacity of the system. The model results can be used to determine the appropriate system size for a specific desired drying capacity.

It should be also noted that the turbine extraction power, $\dot{W}_{ext}$, is included in this model (see Equation (7)). Basically, $\dot{W}_{ext}$ is set equal to zero in this study. It is included to examine the possibility of using the dryer to generate electricity while it is working.

In order to close the system of equations above we need the auxiliary relations, they are (Çengel and Boles, 2014):

$$h_w \approx h_{f@T_2} \quad (8)$$

$$\dot{m}_w = \dot{m}(\omega_2 - \omega_1) \quad (9)$$

$$\omega_1 = \frac{0.622\phi_1 p_{g1}}{p_1 - \phi_1 p_{g1}} \quad (10)$$

$$\omega_2 = \frac{0.622\phi_2 p_{g2}}{p_2 - \phi_2 p_{g2}} \quad (11)$$

$$p_g = p_{sat@T} \quad (12)$$

In this study, the process is characterized by T_2 , which is unknown before we could obtain the converged solution. Because h_w and p_{g2} depend on T_2 (see Equations (8) and (12)), we estimate T_2 , use a lookup table with interpolation to calculate h_w and p_{g2} and then iterate until we obtain a converged solution.

Description of the Simulation

The solar chimney crop-drying system studied here has the same dimension as described in Ferreira *et al.* (2008). The reasons are that its capacity is acceptable based on the reported performance and it can be locally manufactured with low initial investment

capital. According to Maia *et al.* (2009), the efficiency of conversion of solar energy into useful thermal energy of this system is 5%. This very small value would be due to the fact that the solar chimney efficiency depends on the system size (Koonsrisuk and Chitsomboon, 2009b) and the proposed system has a very small size relatively.

The ambient conditions of Nakhon Ratchasima province are described here. The Department of Alternative Energy Development and Efficiency of Thailand (2012) reported that the average solar radiation of Nakhon Ratchasima province is 17.94 MJ/m²-day. The average total sunshine hours are 7.67 h/day for Nakhon Ratchasima province (Wikipedia, 2012). Accordingly, the solar insolation used in this study is 649.4 W/m². The relative humidity and temperature are 70.5% and 27.9°C, respectively (Trek Thailand Team, 2012).

According to the formulations described in mathematical model, the MATLAB code was developed. The systems of equations are solved by using Newton-Raphson method. The iterative scheme starts with an initial guess of the values of the unknowns. The process is continued until the unknowns do not change from one iteration to the next, within a specified convergence criterion of 1×10^{-5} (Koonsrisuk and Chitsomboon, 2013). It should be noted that the friction factor is set to $f = 0.008428$ (Koonsrisuk, 2013) for all tested cases.

Validation

The experimental results by reference (Ferreira *et al.*, 2008) were used to validate the proposed mathematical model. Drying tests of bananas were performed from March 31st to April 9th in Ferreira *et al.* (2008). The average ambient temperature, solar radiation, ambient air relative humidity are 27.5°C, 16 MJ/m², 21%, respectively, during that time. Using the proposed model, it was found that the simulated mass flow rate is 1.39 kg/s while the paper reported the value of 1.36 kg/s. It was also found that the drying air temperature of air is 11°C higher than that of the ambient air, while the value of 10°C was reported in

Ferreira *et al.* (2008). The agreement between the predicted and measured results would be justified the validity of the model. In addition, the predicted effective moisture removal rate is equal to 0.0058 kg/s. The value of 0.0058 kg/s represents the average water evaporation rate from the product during the entire drying period.

Results and Discussion

Cassava is considered as one of the most important economic crops of Thailand. It was reported that cassava is mainly cultivated in Nakhon Ratchasima province (Sirirote *et al.*, 2010). Consequently, we choose cassava to be the drying product in this study. Generally, the initial moisture of cassavas is 60-65% (The Agricultural Future Trading Commission of Thailand, 2012). Commercially, the required moisture content of cassava chips is 14% and it takes 2-3 days to dry cassava chips in the open air (The Agricultural Futures Exchange of Thailand, 2012).

The temperature and velocity of the air increase with the distance from a collector inlet towards a collector outlet. To obtain the appropriate drying temperature level, we choose to spread the cassava chips uniformly over the collector floor from $0.5r_r$ to $r_r - r_c$. The location of $0.5r_r$ is at a middle location between collector inlet and outlet and the location of $r_r - r_c$ is at the collector outlet. Consequently, the drying floor area is about 120 m². For each square meter of drying floor area, 10-12 kg of fresh cassava chips should be spread (Best and Ospina, 1989). As a result, the capacity of the proposed solar chimney dryer is about 1,000 kg (1 ton).

Let us assume the initial moisture of 65%, final moisture of 14% and effective drying rate of 0.0058 kg/s, a 1 ton of fresh cassava chips should be dried in 28.4 h. Therefore a drying process should be finished within 3 days. This is the same order as that of a typical open-air drying system. One advantage of the solar chimney dryer over the open-air sun drying is that solar chimney dryers can operate during the night. At night,

the energy stored in the ground throughout the day can heat the air in the collector. In addition, solar chimney dryers can function on cloudy or rainy days. As the solar chimney dryer does operate in diffused sunlight, the air inside a collector will always have a higher temperature than the ambient temperature.

The results from the proposed mathematical model are shown in Figure 2 to Figure 5, each figure shows 4 graphs. One is for the case when d_r is varied, $h_c = 12.3$ m and $\eta_{col} = 0.05$, one is for the case when d_r is varied, $h_c = 12.3$ m and $\eta_{col} = 0.1$, one is for the case

when $d_r = 25$ m, h_c is varied and $\eta_{col} = 0.05$, and one is for the case when $d_r = 25$ m, h_c is varied and $\eta_{col} = 0.1$. The horizontal axis of those figures is the collector diameter (for the cases of varying d_r) or the chimney height (for the cases of varying h_c).

The mass flow rate ($\dot{m}$) and drying air velocity (V_2) are shown in Figures 2 and 3, respectively. It is obvious that they share the same trends. Both mass flow rate and velocity increase with the collector diameter/chimney height. In addition, they increase significantly with the collector efficiency.

Figure 4 shows the profile of the temperature difference between the drying air and ambient air. When h_c is varied, temperature decreases with the chimney height. Order of magnitude analysis reveals that $\eta_{col} q'' A_r / \dot{m}$ has the highest significant effect on the value of T_2 (see Equation (2)). According to Figure 2, $\dot{m}$ increases with h_c . Therefore, an increase of h_c results in the decrease of ΔT . In the case of $\eta_{col} = 0.05$, on the other hand, temperature decreases with collector diameter when d_r is less than 30 m and then it increases with collector diameter when d_r is greater than 30 m. Further investigation reveals that this trend is the result of the combined effect of $\dot{m}$ and A_r values when d_r is increasing. However, ΔT increases

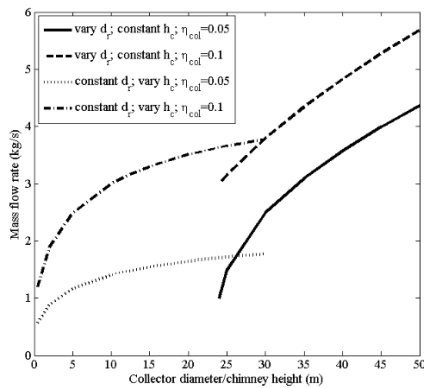


Figure 2. Influence of collector diameter, chimney height and collector efficiency on mass flow rate

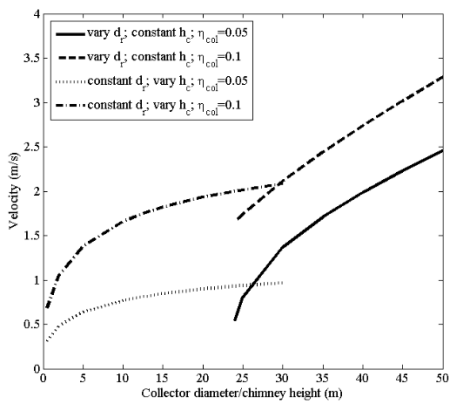


Figure 3. Influence of collector diameter, chimney height and collector efficiency on drying air velocity

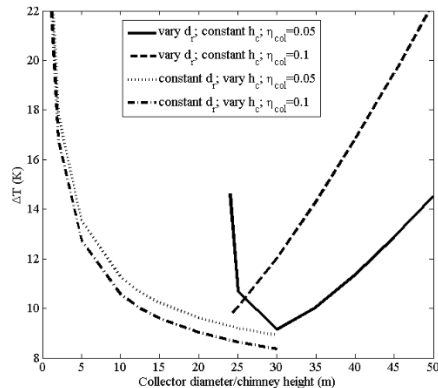


Figure 4. Influence of collector diameter, chimney height and collector efficiency on temperature difference between drying temperature and ambient temperature

with d_r monotonically when $\eta_{col} = 0.1$. In other words, it can be seen in Figure 4 that the change of collector efficiency from 0.05 to 0.1 does not have a significant effect for the case of varying h_c . In contrast, it is notable when d_r is varied.

The maximum allowable temperature for every food product is usually 15–20°C higher than the ambient temperature (Murthy, 2009). According to Figure 4, the chimney height and collector diameter should be greater than 2 m and less than 40 m, respectively, when $\eta_{col} = 0.1$.

The relative humidity of air after passing through the chips is given in Figure 5. The plot shows that increasing the system size appropriately can increase the drying potential of the system (by reducing the drying air relative humidity). The effects of increasing η_{col} , which are shown in Figures 2–5, look promising. The collector cover of Ferreira *et al.* (2008) was made of a plastic film. The replacement of plastic film with glass should increase the collector efficiency significantly, although the initial cost will be increase correspondingly. This would be economically justified in the long run depending on the specific drying capacity desired.

It should be emphasized that $\dot{W}_{ext}$ is set equal to zero for all the above cases. The possibility of electricity generation of solar chimney dryers is interesting and deserves a further investigation. It was found that, after providing its thermal energy to the drying chips, the baseline case ($d_r = 25$ m, $h_c = 12.3$ m and $\eta_{col} = 0.05$) does not have enough potential to generate electricity. With the appropriated combination of d_r , h_c and η_{col} , the dryer has the potential to generate electricity. Based on the ranges of d_r , h_c and η_{col} studied here, however, the maximum electricity that can produce is 40 W, which is not economically practical.

Conclusions

In this paper we try to assess the drying potential of a solar chimney. A simplified mathematical model is proposed. The theoretical result agrees very well with the

experimental result. We examine the solar chimney dryer for Nakhon Ratchasiman province in Thailand. It was found that the predicted drying air temperature is in the allowable temperature limit for food product. Agricultural products generally require drying through a consistent application of relatively low heat. This study shows that the solar chimney dryer can satisfy that requirement. Therefore the solar chimney system is a promising alternative drying system. However, it is not recommended to use the dryers studied here to produce electricity.

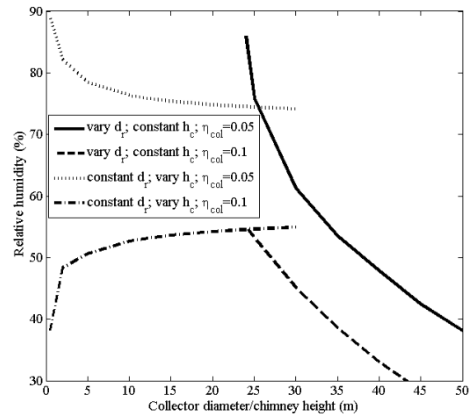


Figure 5. Influence of collector diameter, chimney height and collector efficiency on relative humidity of air

Acknowledgements

This research is sponsored by the Royal Golden Jubilee (RGJ) Ph.D. Program of the Thailand Research Fund (TRF).

Appendix

Derivation of Equation (2)

According to the conservation of energy principle, we have

$$\eta_{col} q'' A_r = (H_2 - H_1) + \frac{\dot{m}^3}{2\rho^2} \left(\frac{1}{A_2^2} - \frac{1}{A_1^2} \right) - \dot{m}_w h_w \quad (A1)$$

where Çengel and Boles (2014)

$$H = \dot{m}(c_p(T - 273.15) + \omega h_g) \quad (A2)$$

$$h_g = 2,501.3 + 1.82(T - 273.15) \quad (A3)$$

$$\dot{m}_w = \dot{m}(\omega_2 - \omega_1) \quad (A3)$$

Substituting and rearranging, we obtain

$$T_2 = \frac{1}{(c_p + 1.82\omega_2)} \left(\frac{\eta_{\text{col}} q^* A_c}{\dot{m}} - \frac{\dot{m}^2}{2\rho^2} \left(\frac{1}{A_c^2} - \frac{1}{A_f^2} \right) + (\omega_2 - \omega_1)(h_w - 2,501.3) \right) + 273.15$$

Nomenclature

A	flow area, m ²
A _r	roof area, m ²
c _p	specific heat capacity at constant pressure, kJ/(kg.°C)
d _r	collector diameter, m
f	friction factor
g	gravitational acceleration, m/s ²
H	total enthalpy, J
h _c	chimney height, m
h _f	specific enthalpy of saturated liquid, kJ/kg
h _g	specific enthalpy of saturated vapor, kJ/kg
h _{roof}	roof height above the ground, m
h _w	specific enthalpy of moisture, kJ/kg
$\dot{m}$	mass flow rate, kg/s
$\dot{m}_w$	moisture removal rate, kg/s
q''	insolation, W/m ²
p	pressure, Pa
p _g	saturation pressure of water, Pa
r	coordinate in roof radius direction
R	ideal gas constant, J/kg.K
r _c	chimney radius, m
r _r	collector radius, m
T	absolute temperature, K
$\dot{W}_{\text{ext}}$	power extracted by turbine, W
≈	approximately equal

Greek Symbols

γ	specific heat ratio
Δp	pressure drop, Pa
ΔT	temperature difference between ambient and collector outlet, K
η _{col}	collector efficiency

η _{dryer}	dryer efficiency
ρ	density, kg/m ³
φ	relative humidity
ω	absolute humidity, kg water vapor/kg dry air

Subscripts

1,2,3,4	position as depicted in Figure 1
ave	average value
col	collector
sat	saturation

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