Design of the double pass solar collector for drying

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Abstract. Solar energy is the free and nearly endless source which can be applied in various fields. Viet Nam locates near the equation and has enormous solar radiation, so this is the good chance for us to harness it. Moreover, Viet Nam is also the agricultural country with the large number of export product. With the large advantage in having enormous solar radiation, we can save up the large amount of energy for drying and other applications. In this paper, the 2 double pass solar collectors which have fins and without fins for drying were designed and simulated by CFD Ansys in some different conditions. The result displays that with the same size, the outlet temperature of the solar collectors with fins is 2 times higher than the without fins one. This temperature can reach 94.6°C with solar radiation 1026.32W/m², mass flow rate 0.12kg/s and can reach 68.63°C with the same radiation and mass flow rate 0.18kg/s, so it is eligible for agricultural drying.

1. Introduction

The sun is a giant thermal ball that always reacts in the core, providing enormous heat. The sun's surface temperature of 6000°C corresponds to a radiation intensity of 70,000 to 80,000 kW per m², but the earth receives only a small fraction of this radiation intensity. If the total energy received from the sun to the earth of 174 petawatts (PW) per year could be completely exploited, that would meet more 10,000 times than the total energy used on earth in 1 year. The solar radiation intensity outside the atmosphere reaches to 1360 W/m². However, it is lost when penetrating the atmosphere, reduced to about 700 to 1000 W/m² depending on the region and season. Solar energy can be harvested for heating [1], lighting, cooking and a wide range of applications in life.

The direct drying process applied for drying agricultural products has the advantage of simplicity, low-cost. However, it is influenced by factors such as dust, insects and rain. Therefore, the chamber is used for drying in order to help minimize these effects. Moreover, the solar energy can be used during day time and resistors can be used during times of lack of sunlight, rain storms or at night in order to save energy. When using a solar-based drying system in a closed chamber, the internal temperature will be higher due to the greenhouse effect [2]. In the solar-based drying system, air is passed through the absorber plate, heated and fed into the drying chamber. However, a backup heat source is required in order to avoid product deterioration due to the effects of the weather [3]. This system achieves high operating efficiency due to the important part named the heat absorbing plate and increasing the heat transfer area by using metal and wavy fin.

A solar-based drying system with V-profile collector for drying tea and chili was experimented by Fudholit in [4]. The system is tested with solar radiation conditions of 700 W/m², total heat transfer area of...
15m², mass flow of 0.3 kg/s and average outlet air temperature of 50°C. In this system, an auxiliary resistor of 10 kW is used in condition of solar energy absence. Fresh tea leaves with an initial moisture content of 87%, must be reduced to moisture content of 54% in order to maintain the tea color. The auxiliary resistance is used when the temperature is lower than 50°C and the air flow through the system of 15m³/min. The initial weight of the fresh tea leaves is 10.03 kg before drying and is 2.86 kg after drying. Another solar-based drying system for corn grain was designed by BM Santos, MR Queiroz and TPF Borges in [5]; with a flat collector of cobblestones with a thickness of 20 cm, heat transfer area of 1.8m², glass thickness of 5mm and air flow when operating of 2.1m³/min. It is operated under condition of outlet temperature of 50°C in the city of Campinas, Brazil.

Azharul Karim [6] pointed out that when the double pass collector is used for the flat-profile, it has significantly higher efficiency than the single pass collector and the double pass collector should not be used for V-profile collectors. K. J. Grecia designed and evaluated the mango solar dryer with thermal energy storage and recirculated air [7]. After the comparison with other results, they claimed that their hybrid solar dryer can reduce the drying time from 7.17 hours to 5.32 hours. L. Diana, A. G. Safitra, D. Ichsani and S. Nugroho simulated the air flow through the prism obstacles inside the solar heater channel, they concluded that the obstacle can create the back flow, so air temperature can be increased inside the channel[8]. Ekadewi A. Handoyo, Djatmiko Ichsani, Prabowo and Sutardi conducted the experiment the solar air heater having V-corrugated absorber plate with obstacles bent vertically at any angle [9]. They found that the obstacles raised the solar air heating performance, but pressure drop also went up. The optimum angle in their experiment was found 30°. Based on the basic dimensions of 0.9m by 1.9m, two models of flat and finned collector are designed by Inventor software in this paper. Then, the CFD Ansys software is used in order to calculate simulation parameters of two models at several different times in order to compare two performances.

The rest of this paper is organized as follows. Designs of simulation model for solar-based drying system are described in section 2. In section 3, an evaluation of experimental results is presented. Section 4 concludes the paper.

2. Simulation Model for Solar-Based Drying System

The model was designed using the Inventor software, version 2018 and was simulated using Ansys Fluent software, version 2020. The first model is a flat collector (as shown in figure 1) with the dimension of 1000mm by 2000mm by 5mm (2m² of the upper surface and 2m² of the under surface); the material of steel; an absorption coefficient of ε=0.9; coated glass sheet thickness of 5mm; a penetration coefficient of τ=0.9; the insulation wooden wall thickness of 40mm and the dimension of inlet and outlet of 200mm by 1000mm.

![Figure 1. The flat collector model](image1)

![Figure 2. The finned collector model](image2)
The second model (as shown in figure 2) has the same dimension, but the flat collector is replaced with the one with steel fins with a total of 10 blades. Each fin has size of 3mm thickness, 50mm height and 100mm fin spacing.

The two models were simulated by the diagram as shown in figure 3 and meshed calculation with 174,930 nodes and 162,950 elements as shown in figure 4.

**Figure 3.** The diagram for simulation.

**Figure 4.** Meshing calculation on the models.
The simulation was performed with Roseland radiation model and Solar Ray Tracing. The coordinates are simulated with latitude of 10.769° and longitude of 106.662° in Ho Chi Minh City. The air flow in the system is calculated with a k-epsilon model. All residual values of parameters consist of continuity, velocity, energy and k-epsilon are set to $10^{-6}$. Solar radiation simulated at 9am, 13pm, and 16pm on January 15 in order to evaluate system parameters. The materials used in the system have parameters listed as in Table 1.

| Material type | Density (kg/m$^3$) | Specific heat (J/kg.K) | Coefficient of thermal conductivity (W/m.K) |
|---------------|--------------------|------------------------|-------------------------------------------|
| Glass         | 2321               | 750                    | 1.15                                      |
| Steel         | 8030               | 502.48                 | 16.27                                     |
| Wood          | 700                | 2310                   | 0.173                                     |

According to [10], the collector plate radiation is obtained as in Eq. (1).

$$q_1 = \varepsilon I_{\text{solar}}$$

where $\varepsilon$ is the absorption coefficient of collector and $I_{\text{solar}}$ is the intensity of solar radiation through the glass.

Therefore, we have the following collector heat equilibrium as in Eq. (2).

$$\eta = q_1 - q_2$$

where $\eta$ is heat increase in air temperature, $q_1$ is absorbed collector heat and $q_2$ is heat loss.

There are 3 types of heat losses could be listed as follows: loss through coated glass, side loss and heat loss through bottom. The boundary conditions for determining the value of heat loss through the surrounding environment in this model are: convection heat coefficient from the wall, from the cover glass to the surrounding environment $\alpha = 20\text{W/m}^2\text{.K}$; ambient air temperature of 29°C. The surrounding walls are considered to receive no solar radiation.

3. Evaluation of Experimental Results
Radiation values of 948.917 (W/m²), 1026.32 (W/m²), 848.967 (W/m²) are calculated by Ansys program at 9am, 13pm, 16pm on Jan 15, respectively. The outlet air temperatures at every moment are also have different values. The inlet air velocity is also changed. According to [9,10], the mass flow rate can be calculated based on air velocity $v$ (m/s), duct area (m²) and density $\delta$ (kg/m³) by using Eq. (3).

$$G = v \times F \times \delta$$

The mass flow rates have the results of 0.12 (kg/s) and 0.18 (kg/s) according to the inlet dimension of 900mm by 1000mm and simulated inlet velocity of 0.5m/s and 0.75m/s, respectively.

3.1 The outlet temperature
The outlet temperature values at 9:00, 13:00 and 16:00 with mass flow rate of 0.12kg/s and 0.18kg/s, respectively are displayed in figure 5.

With the flow rate 0.12 kg/s (figure 5a), the model 1 which has the highest outlet temperature at 13:00 is 44.8°C due to having the highest number of radiation. However, according to the model 2 at the same period, the higher outlet temperature acquired is 94.6°C as a result of being up 1.5 times of area (4m² of the upper surface and 2m² of the under surface) as the first one. At 9:00, the outlet temperature is lower than the one is at 13:00, reach 41.5°C, 89.6°C correspond to the model 1 and model 2, respectively, because the radiation is 948.97 W/m². At 16:00, radiation is lowest with its value 848.967 (W/m²), so the outlet temperatures which are also low are 35.36°C, 73.66°C correspond to the model 1 and model 2.
Figure 5. The outlet air temperature with the mass flow rate 0.12 kg/s and 0.18 kg/s.

(a)  
(b)  

Figure 6. The temperature variation in plane of model 1 (a) and model 2 (b).

When the mass flow is up to 0.18 kg/s (figure 5b), the outlet temperature is down. Obviously, at 13:00, the outlet temperature of the model 1 is 40°C and the model 2 is 68.6°C. At 9:00, the outlet temperature is 37.79°C, 61.77°C correspond to the model 1 and model 2, respectively. At 16:00, the outlet temperature of model 1 and model 2 are 33.43°C and 55°C, respectively. The variation of temperature in the model 1 and model 2’s plane at 13:00 with the highest radiation and mass flow rate 0.18 kg/s is predicted in figure 6.

Obviously, air temperature is highest near the collector surface. In term of the finned collector, because the heat transfer area is two times higher than that of the flat plate, the radiation is significantly concentrated
at there, result in high temperature on collector’s surface. Therefore, the outlet temperature is higher than that of the plate collector.

3.2 Glass surface’s temperature
When solar sunlight strikes on the glass surface, a large portion of it transmit through glass to heat the collector, other portions will heat the surface and reflect from it. The higher glass transmission coefficient, the more radiation will penetrate the glass and heat collector. The glass surface’s temperature is displayed in figure 7.

![Figure 7](image)

**Figure 7.** The glass surface’s temperature with the mass flow rate of (a) 0.12 kg/s and (b) 0.18 kg/s.

The glass surface’s temperature variation depends on the outlet temperature and radiation at the period time. In term of finned collector, glass surface’s temperature which reaches the highest value at 13:00 is 75°C with the mass flow rate 0.12kg/s and the surface temperature of model 1 also reaches 37.6°C with the same mass flow rate. When the mass flow rate increases to 0.18kg/s, the outlet temperature is low and the glass surface’s temperature is also low, namely 61.4°C of model 1 and 36.15°C of model 2 at 13:00.

3.3 Air velocity variation
The air velocity varying along equipment correspond to 0.5 m/s and 0.75 m/s is depicted in figure (8a, 8b) and figure (8c, 8d), respectively.
Figure 8. Air velocity along the equipment of model 1 correspond to 0.5 m/s (a,b) and 0.75 m/s (c,d). Air velocity variation inside the finned collector corresponds to 0.5 m/s and 0.75 m/s is displayed in figure (9a, 9b) and figure (9c, 9d), respectively.

Figure 9. Air velocity along the equipment of model 2 correspond to 0.5 m/s (a,b) and 0.75 m/s (c,d).

As a result of using finned collector, air flow will be turbulent. Therefore, heat transfer coefficient will not only be surge, but the efficiency is also increased. For that reason, the aim of fin is to raise heat transfer area and heat transfer coefficient.

4. Conclusion
In this paper, the authors design 2 models of flat plate collector and finned collector by using Inventor software and then simulate heat and aerodynamic transmission. The use of a finned collector improves heat transfer efficiency up to twice as much as a flat panel. The highest air temperature coming out of the finned collector can reach up to 94.6°C by 13:00, with a flow of 0.12 kg/s. When using the finned type, the original size of the device does not change, thus saving the usable area.

In addition, the paper shows that the application of simulation software helps designers save time in designing products, as well as a preliminary assessment of equipment efficiency in operation.

Acknowledgments
This project is supported by Van Lang University (VLU), Electronics and Electrical Engineering Division, Ho Chi Minh City 700000, Vietnam.

References
[1] Nguyen D T and Do T N 2020 The efficiency of solar water heating system with heat pump software application designed for resorts in Vietnam Vietnam Journal of Science, Technology and Engineering vol 62 issue 2 pp 56-64
[2] Rajkumar P 2007 Drying kinetics of tomato slices in vacuum assisted solar and open sun drying methods Dry Technol. vol 25 pp 1349-1357
[3] Fadhel M I, Sopian K, Daud W R W, and Alghoul M A 2010 Performance analysis of solar-assisted chemical heat-pump dryer Solar Energ. vol 84 pp 1920-1928
[4] Fudholi A, Sopian K, Othman M Y, Ruslan M H, AlGhoul M A, et al. 2008 Heat transfer correlation for the V-Groove solar collector *8th WSEAS International Conference on SIMULATION, MODELLING and OPTIMIZATION (SMO ’08)* Santander, Cantabria, Spain, September 23-25, 2008.

[5] Santos B M, Queiroz M R and Borges T P F 2005 A solar collector design procedure for crop drying *Brazilian Journal of Chemical Engineering* vol 22 issue 02 pp 277 – 284

[6] Karim A Performance investigation of flat plate, v-corrugated and finned air collectors *Solar Energ.* vol 31 issue 4 pp 452-470

[7] Grecia K J, Albert Luce A, Buenaventura M A, Ubando A, and Gue I H V 2019 Design and evaluation of a mango solar dryer with thermal energy storage and recirculated air *IEEE 11th International Conference on Humanoid, Nanotechnology, Information Technology, Communication and Control, Environment, and Management (HNICEM)* pp 1–5 (Laoag: IEEE)

[8] Diana L, Safitra A G, Ichsani D, and Nugroho S 2019 CFD analysis of airflow through prism obstacles inside solar air heater channel *Journal of Physics: Conference Series 1577* 012038

[9] Handoyo E A, Ichsani D, Prabowo and Sutardi 2014 Experimental studies on a solar air heater having V-corrugated absorber plate with obstacles bent vertically *Applied Mechanics and Materials Vol. 493* pp 86-92

[10] El-Sebaii A A, Aboul-Enein S, Ramadan M R I, Shalaby S M and Moharram B M Investigation of thermal performance of-double pass-flat and v-corrugated plate solar air heaters *Solar Energ.* 36 pp 1076-1086