Design of a solar dryer with fins and baffles for rice-cracker drying

Panuwat Pawakote* and Atit Koonsrisuk
School of Mechanical Engineering, Institute of Engineering, Suranaree University of Technology, Nakhon Ratchasima, 30000, Thailand

*E-mail: jay.panu@hotmail.com

Abstract. The rice-cracker is one of the popular snacks in Thailand. To produce it, drying is a necessary preparative step. This research aims to investigate the performance of a solar dryer with fins and baffles installed in the air flow channels of the dryer for rice crackers. The cover made of a 4 mm thickness poly methyl methacrylate (PMMA). A mathematical model was developed and used to design a solar dryer. The simulations revealed that a minimum of 2.25 m² solar collector area is required for solar radiation of 628.93 W/m² and the mass flow rate of 34 kg/hr and outlet temperature of 64.9°C are obtained. In addition, the collector efficiency is 52.01%.

1. Introduction
The rice-cracker is one of the popular snacks in Thailand. It is made of sticky rice soaked in water for 6 hours, steamed, and mixed with watermelon juice or muskmelon juice. After that it is shaped in a mould and then dried under open sun as shown in figure 1. The open sun drying provides drying with low-cost, however, it relies on the weather. Also, the products may contaminate by dust. The production is suffered by slow drying rate.

Figure 1. Rice cracker drying under open sun.

The solar drying, which the products are kept in a cabinet and dried by heated air created by greenhouse effect in a solar collector, is one of the promising alternative for rice-cracker drying. Yeh et al. [1] conducted a solar drying using a solar collector flat plate. They studied 3 cases: 1) solar collector with fins, 2) solar collector with fins and baffles, 3) solar collector without fins and baffles. The tested solar intensity were at 830 W/m² and 1100 W/m². It was found that the efficiency of Case 1 was 59%, Case 2 was 72%, and Case 3 was 47%. Abene et al. [2] also did an experiment with grape
drying. They compared 3 cases: 1) without obstacle (WO), 2) with obstacle of type WDL1, 3) with obstacle of type TL (transverse-longitudinal). It should be noted that, although calling with different names, Case 3 is the solar collector with fins and baffles. The experiment reveals that Case 1 needed 13 hours and 20 minutes, Case 2 needed 6 hours and 53 minutes, and Case 3 needed 5 hours and 50 minutes for drying. It is obvious that again the case with fins and baffles provides the fastest rate of drying.

AHMED-ZAID et al. [3] studied the drying of onions. They compared 3 cases: 1) without obstacle (WO), 2) with obstacle of type WDL1, 3) with obstacle of type TL (transverse-longitudinal). The experiment reveals that Case 1 needed 13 hours and 30 minutes, Case 2 needed 7 hours, and Case 3 needed 5 hours and 48 minutes for drying. Meanwhile, the study of [4] shows that drying of apple sliced using a solar collector without fins and baffles needs longer time. Again, the study of [5] reveals that the efficiencies of the solar collector with fins and without fins were 40.02% and 34.92%, respectively. The study of [6] indicates that the different cover materials provide different performance. These cited articles provide the impression that drying using the solar collector with fins and baffles is a very promising option.

As a result, the objective of this study is to design a solar collector with fins and baffles for rice-cracker drying. First, a mathematical model was developed and used to calculate the required dimensions for the collector that provide the heated air temperature of 60°C. Then a prototype of solar collector was built and tested.

2. Design concept of the prototype
In the literature, it is evident that solar collectors equipped with fins and baffles provide a high temperature comparatively. Accordingly, this study aims to design a rice-cracker dryer equipped with fins and baffles. The proposed design consists of a transparent cover, black-color absorber, insulation, and fan. To help the heated air flows easily in zig-zag pattern as shown in figure 2, a fan is a part of the prototype proposed as shown in figure 3.

![Figure 2. Airflow paths in the solar collector.](image)

![Figure 3. Schematic diagram of proposed solar collector with fins and baffles.](image)

3. Mathematical Modeling
A code was developed using Matlab to predict the performances of the proposed prototype. A set of equations used are as follows [7-10]:

\[ A_c = BL \]  \hspace{1cm} (1)
\[ A_f = 2H(L - 0.4) - (15W_2)t \]  \hspace{1cm} (2)
\[ A_b = 15(2W_2W)t \]  \hspace{1cm} (3)
\[ Re = 2m/\mu(B + H) \]  \hspace{1cm} (4)
\[ Nu = 0.0158(Re^{0.8}) \]  \hspace{1cm} (5)
\[ D_{e,f} = 2H/(1 + (H/w_1)) \]  \hspace{1cm} (6)
\[ h = (kNu)/D_{e,f} = h_1 = h_2 \]  \hspace{1cm} (7)
\[ h_w = 5.7 + 3.8V \]  \hspace{1cm} (8)
where $D_{e.f}$ refers to the hydraulic diameter and $h$ is the heat transfer coefficients between the air and wall. The fin efficiency ($\eta_f$) and baffle efficiency ($\eta_b$) can be calculated from equations (9) and (10).

$$\eta_f = \left( \tanh \left( \frac{Mw_2}{2} \right) \right) / \text{Mw}_2$$  \hspace{1cm} (9)

where $M = \sqrt{2h_z/kt}$

$$\eta_b = 26.361(w/D_{e.f})^{-0.454}(L/I)^{-0.634}$$  \hspace{1cm} (10)

$$\phi = 1 + (A_f/A_c)\eta_f + (A_b/A_c)\eta_b$$  \hspace{1cm} (11)

$$\theta_r \approx 4\sigma T_{f,m}^2 / [(1/\varepsilon_r) + (1/\varepsilon_q) - 1]$$  \hspace{1cm} (12)

where $\alpha = \varepsilon = 0.95$, $\varepsilon = 0.94$ in equation (12).

It should be noted that, in equation (12), the average temperature of the surface absorbing solar radiation ($T_{P,m}$) is assumed to be equal to the flow average temperature ($T_{f,m}$).

The heat loss coefficient from the top surface of the solar collector ($U_t$) can be calculated using equation (13) [3, 4].

$$U_t = \left\{ \frac{2(T_{P,m} - 520)}{\left( \frac{T_{P,m}}{I_d} \right) 0.43 \left( 1 - 0.100 \frac{T_{P,m}}{T_{P,m}} \right)} + \frac{1}{h_w} \right\}^{-1}$$  \hspace{1cm} (13)

$$+ \sigma (T_{P,m} + T_o) \left( T_{P,m}^2 + T_o^2 \right) / \left( (\varepsilon_p + 2x0.00591h_w) - 1 \right) + [2x2 + (1 + 0.089h_w - 0.1166h_w \varepsilon_p)(1 + 0.07866x2) - 1 + 0.133 \varepsilon_p] / \varepsilon_p - 2$$

where

$$U_b = \frac{U_b}{f_{L}} \approx 0$$  \hspace{1cm} (14)

$$U_L = \frac{(U_b + U_c)(h_1h_2\phi + h_1h_2\phi h_1h_2\phi + U_b h_1h_2\phi)}{h_1h_2h_3h_4U_1 + h_1h_2h_3h_4U_2 + h_1h_2h_3h_4U_3 + h_1h_2h_3h_4U_4}$$  \hspace{1cm} (15)

$$F' = \frac{h_r h_1 + h_2 h_3 h_4 + h_2 h_3 h_4 + h_2 h_3 h_4}{U_1 + h'_r + h'_r + h'_r + h'_r + h'_r + h'_r + h'_r + h'_r + h'_r}$$

$$F_R = (mC_p/A_c U_L) (1 - \exp [ - (U_lF' A_c / mC_p)])$$  \hspace{1cm} (16)

The efficiency and outlet temperature of the solar panel could be estimated using equations (17) and (18) respectively [8],

$$\eta = \frac{q_{in}}{h_o} = F_R[A_p \tau_g^2 - U_L(T_{f,i} - T_o)] / I_0$$  \hspace{1cm} (17)

$$T_{f,o} = F_R A_c (I_o \tau_o A_p - U_c(T_{f,i} - T_o)) / mC_p + T_{f,i}$$  \hspace{1cm} (18)

where $T_s = T_{f,i}$ and $\tau_g = 0.875$.

4. Dimensions of the proposed prototype

The expected outlet temperature of the proposed prototype is 60°C. Using the mathematical model proposed, the dimensions of the prototype are as shown in table 1, where the layout of the prototype is displayed in figure 4.
Table 1. Dimensions of the prototype.

| Parameter                                  | Value  | Unit   |
|--------------------------------------------|--------|--------|
| Ta (ambient temperature)                   | 40     | °C     |
| I (incident solar radiation)               | 628.93 | W/m²   |
| L (collector length)                       | 1,840  | mm     |
| B (collector width)                        | 1,220  | mm     |
| W (width of baffles)                       | 190    | mm     |
| W1 (distance between fins)                | 287.5  | mm     |
| W2 (height of fins)                        | 60     | mm     |
| t (thickness of baffles)                   | 10     | mm     |
| l (distance between baffle to baffle)      | 20     | mm     |

Figure 4. (a) Top view and (b) front view of the prototype.

5. The prototype
According to the dimensions shown in table 1, the prototype was built. Its fan is powered by a PV panel next to the collector as depicted in figure 5. The measuring instruments were calibrated and equipped. The specifications of the instruments is listed in table 2.

Figure 5. Prototype built.

The details of the prototype are as follows:
(1) A cover was made from a poly methyl methacrylate (PMMA) sheet with thickness of 4 mm.
(2) An absorber plate was made from a 1-mm-thick iron sheet painted black on its top side. Its absorptivity and emissivity are 0.95 and 0.95, respectively.
(3) A fin, baffle, and insulation were made from a plastwood sheet with emissivity of 0.91-0.93 and thermal conductivity of 0.19 W/mK.
(4) A cooling fan with mass flow rate of 18.3 cfm (NMB Model 06015KA-24N).
Table 2. Specifications of the measuring instruments.

| Equipment                      | Description               | Accuracy         |
|-------------------------------|---------------------------|------------------|
| Thermometer (OEM TER040)      | Temperature Gage Meter    | ±1°C             |
| Humidity/Temperature (Honeywell HIH6131-021-001) | Measurement Humidity/Temperature | ±4%RH,±1°C     |
| Thermometer /Hygrometer       | Measurement Humidity/Temperature | ±4%RH,±1°C     |

6. Results

The ambient, experimental outlet temperature, and predicted outlet temperature are displayed in figure 6 and figure 7. It can be seen that the trends and values of the experimental and the predicted temperatures are agree well. The errors are 12.90% and 12.23 % on 25 and 26 November 2017, respectively. Their discrepancy could be due to heat losses from the walls of the air passages, where these losses are not included in the modelling. It should also be noted that the trends of the outlet temperatures are in accordant to the ambient temperature.

![Figure 6. Comparisons on 25 Nov 2017.](image)

![Figure 7. Comparisons on 26 Nov 2017.](image)

The system efficiencies are shown in figure 8. It is obvious that the efficiencies on both days are approximately the same and the values are relatively constant. This is in accord with intuition as the outlet temperature changes with the ambient temperature.

![Figure 8. System efficiency versus time.](image)

The Nusselt number (Nu) and Reynolds number (Re) of the system are shown in figure 9 and figure 10. It can be seen that both numbers have the same trend. Their trends are opposite to the trend of the ambient temperatures shown in figure 6 and figure 7. This could be due to the change of the flow properties with the ambient temperature.
Figure 9. Nusselt number versus time.

Figure 10. Reynolds number versus time.

7. Conclusion
This study investigates the performances of a solar collector with fins and baffles numerically and experimentally. It was found that the experimental efficiency and outlet temperature are about 47.27% and 60.1°C, respectively. While the predicted efficiency and outlet temperature are about 52.01% and 64.9°C, respectively. The discrepancy could be due to heat losses from the walls of the air passages.

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Nomenclature
$A_b$ Total surface area of baffles (m$^2$)
$A_c$ Surface area of collector (m$^2$)
$A_f$ Total surface area of fins (m$^2$)
$C_p$ Specific heat of air at constant pressure J/kg.K at 313 K
$D_{ef}$ Equivalent diameter of conduit with fins, (m)
$F^*$ Collector efficiency factor of solar air heater
$F_R$ Heat-protection factor for solar air heater
Height of air tunnel in solar collector (m)

Convective heat transfer coefficient for fluid flowing over flat plate (W/m² k)

Radiant heat transfer coefficient between two parallel plates (W/m² k)

Convective heat transfer coefficient for air flowing over outside surface cover (W/m² k)

Thermal conductivity of air, insulator (W/m² k)

Thickness of insulator (m)

Mass flow rate of air (kg/s)

Quantity defined (m)

Number of fins on absorbing plate

Useful gain of energy carried away by air per unit time (W)

Reynolds number

Fluid temperature (K)

Inlet and outlet of solar air heater (K)

Average value of Tf (K)

Temperature of absorbing plate, bottom plate (K)

Average value of Tp (K)

Loss coefficient from surface of edges and bottom of solar collector to ambient (W/m² k)

Overall loss coefficient (W/m² k)

Loss coefficient from top of solar collector to ambient (W/m² k)

Air velocity in the flow channel (m/s)

Wind velocity (m/s)

Absorptivity of absorbing plate

Collector efficiency

Baffle efficiency

Fin efficiency

Dimensionless quantity defined

Transmittance of glass cover

Emissivity of glass cover, absorbing plate, bottom plate

Absorptivity

Air density (1.127)(kg/m³) at 313 K

Air viscosity (19.15 x 10⁻⁶) (kg/m.s) at 313 K

Stefan-Boltzmann constant (5.68 x 10⁻⁸) (W/m² K⁴)