ISSN: 0975-833X

Available online at http://www.journalcra.com

International Journal of Current Research
Vol. 8, Issue, 09, pp.39253-39259, September, 2016

INTERNATIONAL JOURNAL OF CURRENT RESEARCH

RESEARCH ARTICLE

NUMERICAL SIMULATION AND TESTING OF A PASSIVE THERMAL SOLAR COLLECTOR ON OKRA (*HIBISCUS ESCULENTUS*)

[*,1]Karma Bako Rimamcwe and [2]Chavan, U. D.

[1]Department of Food Science and Technology, Federal University of Technology, Yola, Nigeria
[2]Department of Food Science and Technology, Mahatma Phule Krishi Vidyapeeth, Rahuri, India

ARTICLE INFO

Article History:
Received 29th June, 2016
Received in revised form
23rd July, 2016
Accepted 05th August, 2016
Published online 30th September, 2016

Key words:

Flat plate collector, Passive solar dryer, Black body, *Hibiscus esculentus.*

ABSTRACT

A flat plate collector, passive solar dryer with dimension 0.8m x 0.6m x 0.4m has been design and constructed with locally available materials such as plywood, aluminum surfaced roofing sheet lined on the inside, with the floor plate painted black as the black body. The black paint used has an absorbance of 0.96. A comparative test carried out on the drying Okra (*Hibisicus esculentus*) revealed an extent of 91% moisture removal by the solar cabinet drying and 86% for open air drying as against 89% literature standards. These methods are simple and illustrate the fact that constructions of efficient passive solar dryers are possible and achievable by our local users on a do-it-yourself basis, and this will minimize cost and over dependence on electricity for drying vegetables at home.

Citation: Karma Bako Rimamcwe and Chavan, 2016. "Numerical simulation and testing of a passive thermal solar collector on okra (*Hibiscus esculentus*)", *International Journal of Current Research*, 8, (09), 39253-39259.

INTRODUCTION

Energy is the capacity to do work. Today's civilization is sustained by energy utilization; this is obvious from the degree of dependence of the modern man on machinery in the provision of all the ingredient of life. Energy exists in various forms: mechanical, electrical, potential, chemical, solar and tidal each possessing different qualities and capabilities (Lapedes, 1976). Solar energy as an exception is the world's most abundant and renewable source of energy (Sayigh, 1977). Nigeria receives about 4.851 x 1012 KWh of energy per day from the sun (Nigerian Meteorological Station, 2006). This is equivalent to about 1.082 million tons of oil equivalent per day, and is about 4000 times the current daily crude oil production based on energy unit, according to Bala *et al.* (2000) as reported by Ikuponisi (2005). Solar thermal energy can cover a substantial part of the world's energy use in a cost effective and sustainable way (Gerhard, 2010). There has been a great negligence in utilizing solar energy especially in developing tropical countries, with over dependence on bio-fuels, hydro power and wood resources (biomass) with consequential results of global warming and environmental pollution which possess threat to human survival (UNEP, 2001). Also in the face of current global crude oil crash there is the need to develop and harness the potentials solar energy offers (Edwin, 2011). Any long-term vision for economic development must include solar thermal technologies to save finite energy sources (Gerhard, 2010). Hence the objective of this study was to design and construct a cost effective passive thermal solar dryer, develop a dryer that could maximize the microclimate and to configure a dryer that could minimize to barest minimum contamination of the drying product for an enhanced shelf life of the drying products.

Corresponding author: Karma Bako Rimamcwe,
Department of Food Science and Technology, Federal University of Technology, Yola, Nigeria.

MATERIALS AND METHODS

Materials Used

Aluminium surface roofing sheets, plywood, glass, mirror, door magnetor, non-carcinogenic black paint, top gum sealant, non-corrosive nails and screws, door hinges, polystyrene foam, wire mesh and gauze.

Design and Mathematical simulation

The direct mode cabinet type flat plate collector with booster was the type employed in this study. It was assembled from various components: the cabinet, glass cover, absorber plate, racks, insulation material and a reflector with compromise that are suitable and available Figure 1.

Figure 1. Assembled Solar cabinet dryer showing contents on drying racks with reflector to boost solar collector, with a louvered like vented door

The Cabinet

The cabinet is in rectangular form bearing side's plane walls, ventilated wall door, a vented netted back plane wall with vents strategically sized and positioned at 0.12m and 0.22m respectively from the floor for moisture removal. The cabinet also supports the absorber plate as black body overlaid on it floor. Also it supplies shelving that supports drying racks carrying products housed in the air heater, and as a perfect system the cabinet is provided with a lid, a single glazing.

Cabinet Area

A simple direct mode design (Fodor, 2006; Shaffer, 1999) with dimensions 0.8m x 0.6m x 0.4m was adopted for these studies; 16mm thick plywood was used. From the design there was a 30mm depression from top of the cabinet to house the glazing, leaving an effective inside volume as 0.8m x 0.6m x 0.37m.

Ease moisture evacuation:

1. Two sides plane wall, lined internally with aluminum roofing sheet, to minimize transient conductive heat loss for even distribution of heat within the air heater.
2. A vented back wall with vents strategically positioned to case moisture evaluation.
3. A multilayer door wall with louvered opening angled at 30^0 to create a natural local cyclonic convective air flow within the cabinet.
4. A multilayer floor, re-insulated with expanded polystyrene foam.

Being a direct mode type, the area of the collector was configured same to that of the receiver to achieve direct influx of solar radiation.

The effective outsider area of the cabinet was expressed mathematically as:

1.Area of two sides

$$2 \times L \times \; H - 0.03 \qquad \text{Eq 1}$$

2.Floor area

$$L \times \; W - 0.032 \qquad \text{Eq 2}$$

3.Area of vented end wall

$$W \; H - 0.03 \; - 2L_V H_V \qquad \text{Eq 3}$$

4.Area of louvered door wall

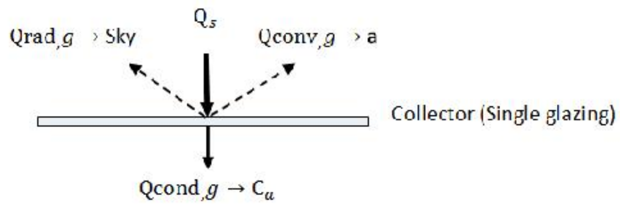
$$W - 0.032 \;\; H - 0.08 \; - L_{LV} H_{LV} \qquad \text{Eq 4}$$

The total effective external area of the cabinet equal sum of all four areas, where L (length), H (height), W (Width), L_V (length of vents), H_V (Height of vents), L_{LV} (Length of louvered door vent), H_{LV} (Height of louvered door vent).

Dryer components: Glass Cover, Air Heater and Absorber Plate

To suppress the convective and radiative losses from the dryer, single glazing was adopted in Figure 1. Glass has the property of being transparent in the visible spectrum, ultraviolet rays, of solar radiation and is opaque to infrared heat (Rajput, 2006) that allows most of the shortwave UV-rays to pass through and not allowing the long wavelength infrared rays from the absorber plate to return to the sky. The convective heat transfer is also reduced due do the air-gap. The cumulative heat generates high temperature within the air heater housing the racks on which test products are placed for drying.

Energy Balance across Dryer Components:



1.Glazing

The glazing serves as the collector which accepts the total global solar incident radiation. The energy balance under transient condition was correlated as:

$$M_g C_g \frac{d_T}{d_t} = Q_S - Q_{Conv,g\rightarrow a} - Q_{rad,g\rightarrow Sky} - Q_{Cond,g\rightarrow Ca} \qquad \text{Eq 5}$$

The accepted incident global solar radiation as correlated by Habon (2003).

$$Q_S = I_b + \frac{I_d}{C_R} \qquad \text{Eq 6}$$

Where C_R is the concentration ratio (for flat plate collector, C_R is assume as 1). Equation 6 becomes:

$$Q_S = I_b + I_d = I_g \qquad \text{Eq 7}$$

I_g is the total global radiation in W/m^2

The convective heat lost from glass to surrounding air, $Q_{Conv,g\to a}$, was correlated as :

$$Q_{Conv,g\to a} = h_a A_g \; T_g - T_a \qquad \text{Eq 8}$$

Where $A_g$the effective glass is surface area in m^2, and h_a is the convective heat transfer in W/m^2K of the surrounding air (Holman, 1985). For flat plate collector which is horizontal and heated by the sun naturally, h_a is evaluated using the empirical correlation described by (Rajput, 2006).

$$h_a = 1.35 \; \frac{\Delta T}{L}^{\;1\;\;4} \qquad \text{Eq 9}$$

The radiant heat transferred from glass to the sky, $Q_{rad,g\to Sky}$ was correlated as;

$$Q_{rad,g\to Sky} = A_g\sigma\varepsilon_g \; T_g{}^4 - T_{Sky}{}^4 \qquad \text{Eq 10}$$

Where A_g is effective glass surface area, in m^2; σ is Stefan-Boltzmann constant 5.67 × $10^{-8}\,W/m^2K^4$; ε_g is emissivity of Glass, T_g is temperature of glass in *K*; $T_{Sky=}T_\alpha$ is the Sky temperature in *K*.

The transient heat conduction by the glass,g to cabinet air (Ca), $Q_{Cond,g\to Ca}$ was correlated as reported by Holman (1985).

$$Q_{Cond,g\to Ca} = -K_g A_g \; \frac{T_g - T_{Ca}}{\Delta_{Xg}} \qquad \text{Eq 11}$$

Where K_g is thermal conductivity of glass; A_g is effective glass surface area, m^2; T_g temperature of glass, T_{Ca} is temperature of the cabinet air in *K* and Δ_{Xg} is the thickness of glass, in *m*. Equation 5 becomes:

$$M_gC_g\frac{dT}{dt} = I_g - h_aA_g \; T_g - T_a \; - A_g\sigma\varepsilon_g \; T_g{}^4 - T_{Sky}{}^4 \; - K_gA_g \; \frac{T_g - T_{Ca}}{\Delta_{Xg}} \qquad \text{Eq 12}$$

2.For Air Heater

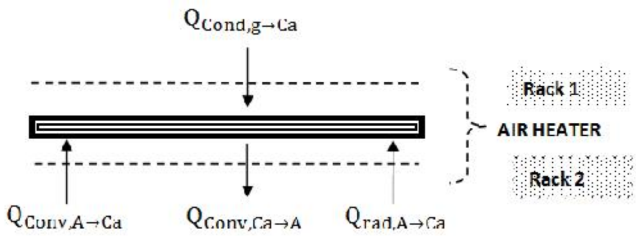
The air heater contains the racks that support the food items. The energy balance within the air heater would also define the energy balance of the racks and food items being a sum. Hypothetically the energy balance under transcend condition was correlated thus,

$$M_{Ca}C_{Ca}\frac{dT}{dt} = Q_{Cond,g\to Ca} - Q_{Conv,Ca\to A} + Q_{Conv,A\to Ca} + Q_{rad,A\to Ca} \qquad \text{Eq 13}$$

Similarly correlated:

$$1.Q_{Cond,g\to Ca} = K_gA_g\frac{T_g - T_{Ca}}{\Delta_{Xg}} \qquad \text{Eq 14}$$

$$2.Q_{Conv,Ca\to A} = h_aA_A \; T_{Ca} - T_A \qquad \text{Eq 15}$$



Where, T_A is the absorber temperature, in *K;* T_{Ca} is the cabinet air temperature, in *K.*

$$3.Q_{Conv,A\to Ca} = h_aA_A \; T_A - T_{Ca} \qquad \text{Eq 16}$$

$$4.Q_{rad,A\to Ca} = A_A\sigma\varepsilon_A \; T_A{}^4 - T_{Ca}{}^4 \qquad \text{Eq 17}$$

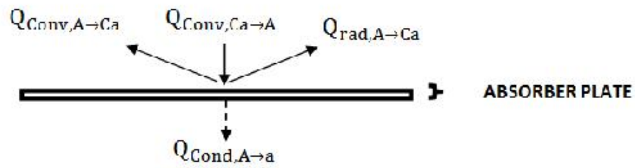
Thus equation 13 becomes

$$M_{Ca}C_{Ca}\frac{dT}{dt} = -K_gA_g \; \frac{T_g - T_{Ca}}{\Delta_{Xg}} \; - h_aA_A \; T_{Ca} - T_A \; + h_aA_A \; T_A - T_{Ca} \; + A_A\sigma\varepsilon_A \; T_A{}^4 - T_{Ca}{}^4 \qquad \text{Eq 18}$$

Note: $\boldsymbol{T_g - T_{Ca}}$ **;** $\boldsymbol{T_{Ca} > T_g \equiv -ve}$

3. For Absorber

The absorber is the thermal component of the dryer, painted black as the blackbody, which serves as the receiver. The energy balance under transient condition is hypothetically represented thus:



$$M_AC_A\frac{dT}{dt} = Q_{Conv,Ca\to A} - Q_{rad,A\to Ca} - Q_{Conv,A\to Ca} - Q_{Cond,A\to a} \qquad \text{Eq 19}$$

Also similarly correlated as:

1.$Q_{Conv,Ca\to A}$, as in equation 15
2.$Q_{rad,A\to Ca}$, as in equation 17
3.$Q_{Conv,A\to Ca}$, as in equation 16
4. $Q_{Cond,A\to a}$, the transient conductive heat loss from the receiver to the bottom outside air as correlated by Holman (1985) stating that:

$$Q_{Cond,A\to a} = \frac{T_A - T_a}{\frac{\Delta X_1}{K_1A_A} + \frac{\Delta X_2}{K_2A_A} + \frac{\Delta X_3}{K_3A_A}} \qquad \text{Eq 20}$$

Thus equation 19 becomes:

$$M_AC_A\frac{dT}{dt} = h_aA_A \; T_{Ca} - T_A \; - A_A\sigma\varepsilon_A \; T_A{}^4 - T_{Ca}{}^4 \; - h_aA_A \; T_A - T_{Ca} \; - \frac{T_A - T_a}{\frac{\Delta X_1}{K_1A_A} + \frac{\Delta X_2}{K_2A_A} + \frac{\Delta X_3}{K_3A_A}} \qquad \text{Eq 21}$$

4.Drying Racks

The racks are two in number shelf supported at 0.1m and 0.2m height respectively from the floor of the cabinet. The racks carry the product, housed in the air heater under the evaporative influence of natural air convection as the primary working fluid. The racks are made of the wire mesh painted black also boosting absorbed radiation for an enhanced dryer efficiency. On the rack, 1kg of Okra (*Hibiscus esculentus*) occupied an effective area of 0.4m^2. The area configured for the racks is 0.43m^2, given a tolerance of 0.03m^2 to ensure good spacing between the produce to effect uniform drying. For the two racks per batch, the facility could dry 2kg of Okra.

Efficiency of the Dryer

The efficiency was computed in terms of the drying rate of the dryer when compared to its open air counterpart, by assessing the rate of moisture lost from the test produce under the two set condition, using the correlation.

$$M_c \ \frac{w}{w} = \frac{W_w - W_d}{W_w} \times 100 \qquad \text{Eq 22}$$

Where, W_w' is Wet weight in g, W_d dry weight in, g.

Statistical Analysis

Data obtained was analyzed by completely randomized design (CRD) given by Panse and Sukhatme (1967) with suitable replication.

RESULTS AND DISCUSSION

The numerical geometry and energy balance across the dryer components were simulated the results obtained are outlined under the sub headings:

Design Analysis and Calculation

The design of the passive cabinet solar dryer involved some considerations which includes.

1. The area of the cabinet and insulation.
2. The volume of the cabinet
3. Numerical energy balance across the dryer components

Area of vented end wall

The dimension 0.8m x 0.6m x 0.4m of the cabinet was used to evaluate the area of the cabinet.

From equation 1

Area of two sides = 0.8 x 2 (0.4- 0.03) = 0.592m^2

From equation 2

Floor Area = 0.8 x (0.6 – 0.032) = 0.454m^2

From equation 3

Area of vented and wall = 0.6x (0.4–0.03)-2(0.4x 0.05) = 0.182m^2

From equation 4

Area of louvered door = (0.6 – 0.032) x (0.4 -0.08) – (0.3 x 0.2) = 0.122m^2

Total area of the Cabinet = Sum of all the four areas

= 0.592+0.45+0.182+0.122 = 1.35m^2

The total area of insulation is also equivalent to the total effective area of the cabinet excluding the vents equal to 1.35m^2.

Volume of the Cabinet

The volume was calculated based on the effective inside dimensions. The dimension of this study is 0.8m length x 0.6m Width x 0.4m height with 0.03m depression at the top to house the glazing and has 0.016m thick insulation for the two side's walls and end, also a multi-layer wall of (0.16m + 0.018m + 0.016m) for the floor of the cabinet. Therefore, the total effective inside volume of the cabinet equal's product of inside length, width and height derived as:

Effective inside Volume = 0.734 x 0.568m x 0.32m = 0.133m^3

Numerical Equation for Energy Balance across the Dryer Components

1.For Glazing

From equation 12

$$M_g C_g \frac{dT}{dt} = I_g - h_a A_g \ T_g - T_a \ - A_g \sigma \varepsilon_g \ T_g{}^4 - T_{Sky}{}^4 - K_g A_g \ \frac{T_g - T_{Ca}}{\Delta_{Xg}}$$

Where:

$M_g = Mass\ of\ Glass = 9Kg$
$C_g = Specific\ Heat\ used = 870\, J/Kg\, K$
$I_g = Global\ solar\ radiation = 225.46\, W/m^2$
$h_a = Heat\ transfer\ coefficient\ of\ air = 50\, W/m^2K$
$A_g = Area\ available\ for\ glass = 0.376 m^2$
$\sigma = Stefan\ Boltzmann\ Constant$
$= 5.67 \times 10^{-8}\, W/m^2K^4$
$\varepsilon_g = emissivity\ of\ glass = 0.94$
$\Delta_{Xg} = Thickness\ of\ glass = 0.004$
$T_{Sky} = 0.0552 T_a{}^{1.5}$, (Habou, 2003)

Upon substitution, equation above becomes

$$7560 \frac{dT_g}{dt} = 225.46 - 18.8 \ T_g - T_a \ - 2.008 \times 10^{-8} \ T_g{}^4 - T_{Sky}{}^4 \ - 98.7 \ T_g - T_{Ca}$$

Table 1. Effect of drying time (hrs) on moisture lost at average relative humidity 38.7 to 44.6 %

Days	Time, hrs	Average Solar Radiation (W/m^2)	Absorber Temperature (^0C)	Ambient Temperature (^0C)	% Cumulative Moisture Lost in SCD	% Cumulative Moisture Lost in OAD
First day	1pm	162.9	38.00	28.00	10.00	16.00
	2pm		42.00	28.00	15.10	23.00
	3pm		40.00	28.00	25.00	31.00
Indoor	3pm-9am		nr	nr	30.00	30.00
Second day	10am	241.1	32.00	27.00	47.50	58.00
	11am		46.00	28.00	60.00	63.00
	12pm		40.00	28.00	62.50	63.00
	1pm		50.00	30.00	67.50	70.00
	2pm		60.00	31.00	82.50	78.00
	3pm		48.00	31.00	85.00	82.50
Indoor	3pm-9am		nr	nr	85.00	82.50
Third day	10am	272.4	46.00	27.50	90.00	85.50
	11am		50.00	29.00	90.50	86.00
	12pm		62.00	30.50	90.50	86.00
	1pm		60.00	31.50	91.00	86.00
	2pm		60.00	31.90	91.00	86.00
	S.E.				0.07	0.23
	CD @ 5%				0.21	0.67
	CV%				0.44	1.44

NB: SCD - Solar Cabinet Drying; OAD - Open Air Drying, nr – Not recorded

Table 2. Cost estimate analysis

Material	Dimension	Quantity	Cost in Naira (₦)
Aluminium Sheet	0.768 x 0.568x 0.0005 m	1	200
Plywood	2.44 x 1.22 m	1	3,400
Glass	0.714 x 0.526	1	1,000
Mirror	0.800 x 0.320	1	800
Door magnetor		1	50
Black paint		2	140
Top sealant		2	450
Nails	2 inch, 1½ inch		300
Door hinges		2	10
Screw		4	10
Polystyrene foam		1	300
Wire mesh		2	50
Wire gauze			300
10 % contingency			701
Total Cost (₦)			7,711 (approx $19)

2.For Cabinet Air Heater

From equation 18,

$$M_{Ca}C_{Ca}\frac{dT}{dt} = -K_gA_g\ \frac{T_g - T_{Ca}}{\Delta x_g}\ - h_aA_A\ T_{Ca} - T_A\ + h_aA_A\ T_A - T_{Ca}\ + A_A\sigma\varepsilon_A\ T_A{}^4 - T_{Ca}{}^4$$

Where,

$M_{Ca} = Mass\ of\ Cabinet\ air = 0.1493\ Kg$

$C_a = Specific\ heat\ capacity = 1.009\ KJ/K_gK$

$h_{Ca} = heat\ transfer\ coefficient\ of\ cabinet = 50\ W/m^2K$

$h_A = Heat\ transfer\ function\ for\ absorber = 1.35\ \frac{\Delta T}{L}^{1\ 4}$

$A_A = Area\ available\ for\ absorber = 0.42m^2$

$\varepsilon_A = Emissivity\ of\ absorber = 0.96$

Upon substitution equation 18 becomes

$$0.15\frac{dT_{Ca}}{dt} = 98.7\ T_g - T_{Ca}\ - 21.8\ T_{Ca} - T_A\ + 1.772\ T_A - T_{Ca}\ ^{5\ 4} + 2.286 \times 10^{-8}\ T_A{}^4 - T_{Ca}{}^4$$

3.For Absorber Plate

From equation 21

$$M_AC_A\frac{dT}{dt} = h_aA_A\ T_{Ca} - T_A\ - A_A\sigma\varepsilon_A\ T_A{}^4 - T_{Ca}{}^4 - h_aA_A\ T_A - T_{Ca}\ - \frac{T_A - T_a}{\frac{\Delta_{X1}}{K_1A_A} + \frac{\Delta_{X2}}{K_2A_A} + \frac{\Delta_{X3}}{K_3A_A}}$$

Where:

$M_A = Mass\ of\ Absorber = 0.5901K_g$

$A_A = Area\ of\ the\ absorber = 0.42m^2$

$\varepsilon_A = Emissivity\ of\ absorber = 0.96$

$K_2 = K_1 = Thermal\ Conductivity\ of\ plywood = 0.13\ W/mK$

$K_2 = Thermal\ Conductivity\ of\ polystyrene\ foam = 0.0398\ W/mK$

$\Delta_{X1} = Thickness\ of\ plywood = 0.016 = \Delta_{X3}$

$\Delta_{X2} = Thickness\ of\ Polystyrene\ foam = 0.018m$

$\Delta_{XA} = Thickness\ of\ absorber = 5 \times 10^{-5}m$

Upon substation becomes

$$354.06\frac{dT_A}{dT} = 21\ T_{Ca} - T_A\ - 2.286\times 10^{-8}\ T_A{}^4 - T_{Ca}{}^4\ - 6.9\ T_A - T_{Ca}\ {}^5\ {}_4 - 0.6T_A + 4.15T_{Sky}{}^2\ {}_3$$

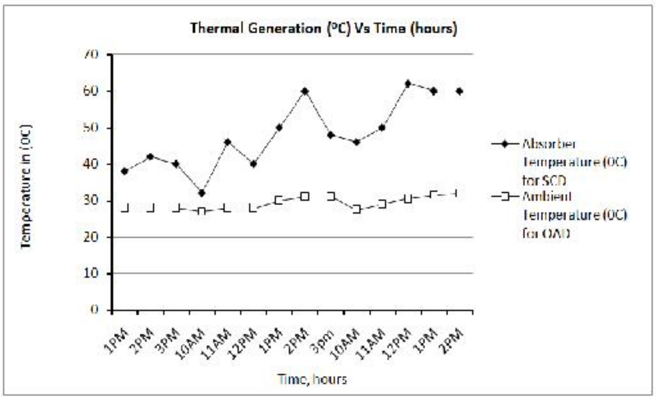


Figure 1. Comparative thermal generation curve

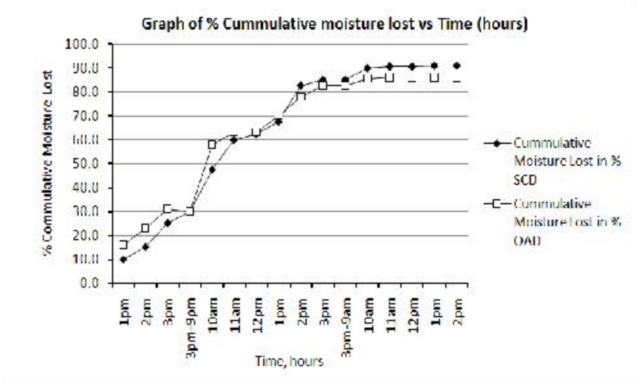


Figure 2. Effect of drying time (hr.) on moisture lost

Drying is a function of temperature, relative humidity and the extent of convective wind current which serves as the working fluid that carries off moisture laden evaporated from drying materials as a result of temperature difference exerted as the food sample by it drying environment. The open air drying (OAD) had the advantage of more exposure to prevalent ambient wind current which constantly takes off the evaporated moisture from the food sample. The solar cabinet drying (SCD) on the other hand had the benefit of maximizing the micro climate in the following respect. Higher thermal generation status almost doubling the ambient temperature, which the open air drying (OAD) is directly dependent on as shown in Table 1 and Figure 1. After 1pm-3pm at first and between 3pm-9am the next day, the SCD recorded desorption of 5.0g moisture whereas the OAD absorbed 5.0g of moisture, Table 1. This indicates that the SCD still had some residual thermal energy and accumulated moisture laden within the cabinet which had gradually vented out during the period.

1. The second day, 11am -12noon, was cloudy. As such there was no appreciable moisture lost for the OAD whereas a desorption of 25g moisture was accounted for SCD; as the proportion of direct diffuse radiation depends on cloud cover, moisture, and dust particle content in the atmosphere and on other environmental parameters (Gerhard, 2010)
2. After drying the second day, keeping time 3pm to 9am, the third day, 5g desorption was accounted for the SCD, whereas no appreciable change was noted for OAD.
3. On the third day both samples had almost reach their drying peaks, as shown by a steady slope in Figure 2 between 10am to 12noon. The two set samples were left for an additional 2hours after which a desorption of 5g moisture was against recorded for the SCD, with no significant weight change for the OAD.
4. Figure 2 shows the extent of drying for SCD as 91% which expresses better compliance to literature standard of 89% (Okada, and Okada 2001), than 86% for OAD. This suggests that products from SCD would have better and extended shelf life, than that from OAD.
5. From design configuration SCD has the advantage of being fly free, which makes it contamination proof and healthier. Whereas the products from OAD are expose to both dust and insect infestation which may constitute contaminant with consequence high microbial load on end product.
6. The cost analysis of producing a locally made passive solar dryer at 7,711 Naira (approx. $19) Table 2 is much cheaper than purchasing an industrial made solar dryer at the cost of 120,000 Naira ($ 286). This proves that locally made solar dryers are possible, cost effective and affordable for the common man ensuring hygiene and improving healthy livelihood.

Conclusion

The passive design adopted for this work has expressed quantum efficiency, both in thermal generating status and the extent of moisture removal of 91% (Moisture content 9%) for Solar Cabinet Drying (SCD) and 86% for Open Air Drying (OAD) as against the literature standard of 89%. 9% moisture content for product in SCD suggest a more stable end product with prolong shelf life due to low water activity inhibiting microbial growth giving a promise of a healthier product compared to 14% moisture content for OAD.

REFERENCES

Bala, B.K. and Woods, J.I. 1994. Simulation of Indirect Natural Convection Solar Drying of Rough Rice, J. of Solar Energy, Elsevier Science Ltd., vol. 53, No. 3., pp. 259-266.

Edwin, Cartidge 2011. Saving for a rainy day. *Science*, vol 334, pp. 922-924

Fodor, Eben 2006. “Build a Solar Dehydrator”, *Mother Earth News*, vol. 2006, pp. 66-70

Gerhard, Faninger 2010. The potential of Solar Thermal Technologies in a sustainable Energy Future. International Energy Agency (EIA) Solar Heating and Cooling Programme

Habou, D; Asere, A.A. and Alhasanssan, A.M. 2003. A comparative Study of the drying rate of Tomatoes and pepper Using Force and natural convection solar dryers, *Nigerian Journal of Solar Energy, Solar Energy Society of Nigeria*, vol. 14, pp. 36-40.

Holman, J.P. 1985. Heat Transfer, McGraw-Hill inc., Singapore 5th ed., pp. 1-13, 60-98

Ikuponisi, F.S. 2003. Status of Renewable Energy in Nigeria, One-SKY-Canadian Institute of sustainable living, Canada, November 21-27, pp. 1-32.

Lapedes, N. 1976. Encyclopedia of Energy McGraw Hill Book Company inc., New York, pp. 714-730.

Nigerian Metorological Station 2006. Wheather Data Sheet for September, 2006, Yola, Adamawa State.

Okada, J.C. and Okaka, A.N.C. 2001. Food Composition Spoilage Shelf life Extension, O Janco Academic Publishers Enugu, Nigeria, pp. 14

Panse, V.G. and Sukhate, P.V. 1985. Statistical methods for Agricultural Workers. 4th ed., New Delhi.

Rajput, R.K. 2006. Heat and Mass Transfers, S. Chand and Company Ltd, India, 3rd ed., pp. 504-876

Sayigh, A.A.M. 1977. Solar Energy Engineering, Academic Press, New York, pp. 83-99. Online source: www.lotusenergy.com

Shaffer, Marcella 1999. “Solar Food Drying”. *Back woods Home Magazine* No. 58

UNEP 2001. United Nations Environmental Protection Annual Solar Report.

****