International Scholars Journals

Advances in Food Science and Technology ISSN: 6732-4215 Vol. 6 (6), pp. 001-009, June, 2018. Available online at www.internationalscholarsjournals.org © International Scholars Journals

Full Length Research Paper

Design and construction of a solar drying system, a cylindrical section and analysis of the performance of the thermal drying system

Ahmed Abed Gatea

Department of Agricultural Mechanization, College of Agriculture, University of Baghdad, Iraq.
E-mail: ahmedabd192000@yahoo.com.

Accepted 28 March, 2018

In this work, a solar drying system of a cylindrical section which consists of a solar collector flat plate, drying chamber cylindrical section and a fan was built and designed for the purpose of drying 70 kg of bean crop. And there was an analysis of a thermal solar collector with flat plate absorption being obtained at the maximum temperature outlet of 71.4°C at 11 am. At radiation intensity, 750 W/m² for air flow rate of 0.0401 kg/s was obtained at ambient temperature of 34°C. The maximum average value of thermal efficiency of the solar air collector obtained from the calculation is 25.64% at air flow rate of 0.0675 kg/s, the maximum daily efficiency drying system was 18.41% at air flow rate of 0.0405 kg/s.

Key words: Solar energy, Solar Dryer Cylindrical Section (SDCS), thermal analysis, drying beans.

INTRODUCTION

In many parts of the world there is a growing awareness that renewable energy has an important role to play in extending technology to the farmer in developing countries in order to increase their productivity (Waewsak et al., 2006). Solar thermal technology is a technology that is rapidly gaining acceptance as an energy saving measure in agriculture application. It is preferred to other alternative sources of energy such as wind and shale, because it is abundant, inexhaustible, and non-polluting (Akinola 1999; Akinola and Fapetu, 2006; Akinola et al., 2006).

Solar air heaters are simple devices to heat air by utilizing solar energy and it is employed in many applications requiring low to moderate temperature below 80°C, such as crop drying and space heating (Kurtbas and Turgut, 2006). Drying processes play an important role in the preservation of agricultural products. They are defined as a process of moisture removal due to simultaneous heat and mass transfer (Ertekin and Yaldiz, 2004). According to Ikejiofor (1985) two types of water are present in food items: the chemically bound water and the physically held water. In drying, it is only the physically held water that is removed. The most important reasons for the popularity of dried products are longer shelf-life, product diversity as well as substantial volume reduction. This could be expanded further with improvements in product quality and process applications.

The application of dryers in developing countries can reduce post harvest losses and significantly contribute to the availability of food in these countries. Estimations of these losses are generally cited to be of the order of 40% but they can, under very adverse conditions, be nearly as high as 80%. A significant percentage of these losses are related to improper and/or untimely drying of foodstuffs such as cereal grains, pulses, tubers, meat, fish, etc. (Bassey, 1989; Togrul and Pehlivan, 2004).

Traditional drying, which is frequently done on the ground in the open air, is the most widespread method used in developing countries because it is the simplest and cheapest method of conserving foodstuffs. Some disadvantages of open air drying are: exposure of the foodstuff to rain and dust; uncontrolled drying; exposure to direct sunlight which is undesirable for some foodstuffs; infestation by insects; attack by animals; etc (Madhlopa et al., 2002).

In order to improve traditional drying, solar dryers, which have the potential of substantially reducing the

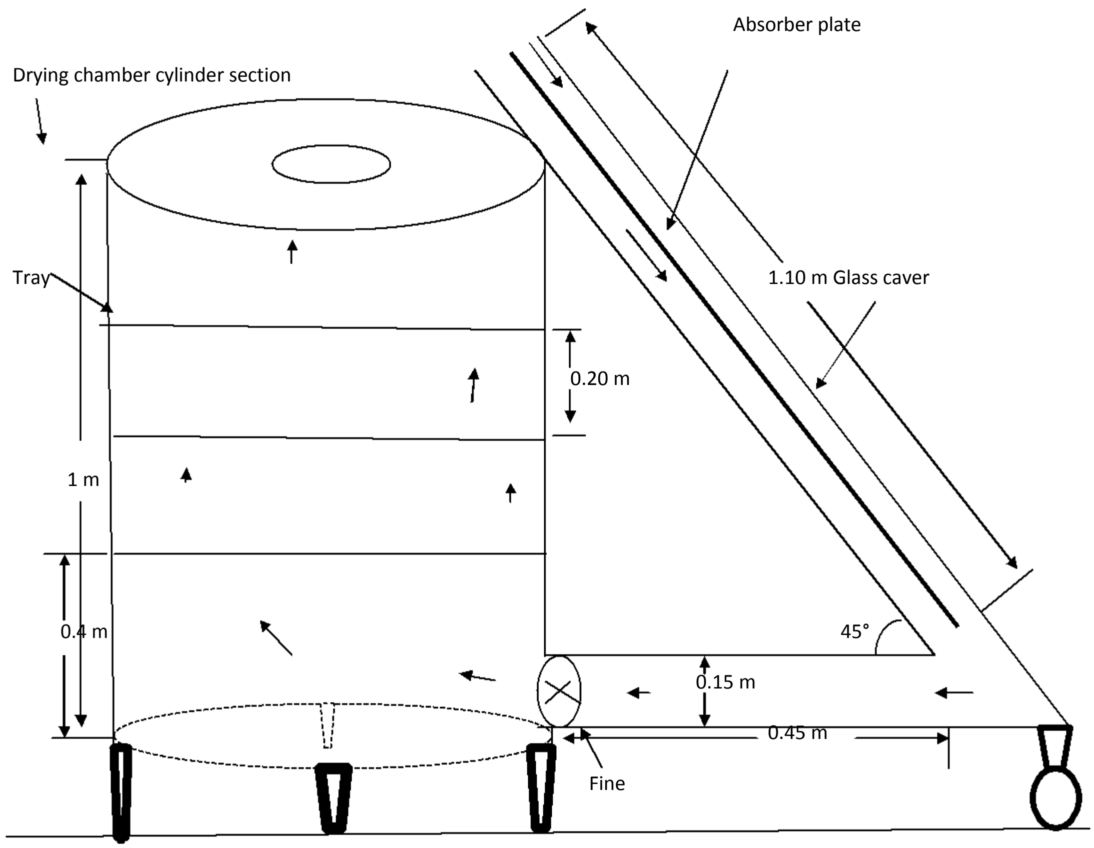

Figure 1. Sectional view of the solar drying system, a cylindrical section.

above-mentioned disadvantages of open air drying, have received considerable attention over the past 20 years (Bassey, 1989). Solar dryers of the forced convection type can be effectively used. They, however, need electricity, which unfortunately is non-existent in many rural areas, to operate the fans. Even when electricity exists, the potential users of the dryers are unable to pay for it due to their very low income. Forced convection dryers are for this reason not going to be readily applicable on a wide scale in many developing countries. Natural convection dryers circulate the drying air without the aid of a fan. They are therefore, the most applicable to the rural areas in developing countries.

Solar drying may be classified into direct, indirect and mixed-modes. In direct solar dryers the air heater contains the grains and solar energy passes through a transparent cover and is absorbed by the grains. Essentially, the heat required for drying is provided by radiation to the upper layers and subsequent conduction into the grain bed.

In indirect dryers, solar energy is collected in a separate solar collector (air heater) and the heated air then passes through the grain bed, while in the mixed-mode type of dryer, the heated air from a separate solar collector is passed through a grain bed, and at the same time, the drying cabinet absorbs solar energy directly through the transparent walls or roof. Therefore, the objective of this study is to develop a mixed-mode solar dryer in which the grains are dried simultaneously by both direct radiation through the transparent walls and roof of the cabinet and by the heated air from the solar collector. The performance of the dryer was also evaluated.

Objectives

The specific objectives of this study were:

1) Designing and construction of a solar drying system with a cylindrical section.
2) Analysis of thermal performance of solar collector with a flat plate.
3) Studying the efficiency of solar drying system for drying beans.

MATERIALS AND METHODS

Solar drying system

The materials used for the construction of solar dryer system are cheap and easily obtainable in the local market. Figure 1 shows the essential features of the solar dryer, consisting of the solar collector

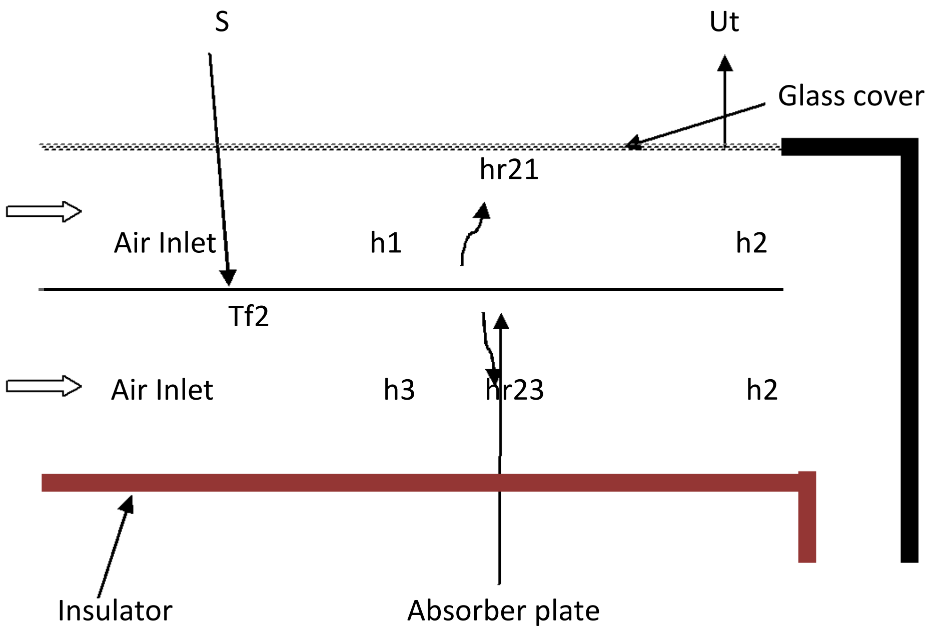

Figure 2. Heat transfer process in a vertical section of the solar air collector.

(air heater), the drying cabinet cylindrical section drying trays, and fan. The system has been installed in one of the buildings in the area of Zaafaraniya - southwest, Baghdad (Coordinates: 33°16'6"N 44°29'41"E).

Solar collector system, with length of 1.10 m and width of 1.10 m, consists of absorber, galvanized steel, single glass cover, back plate, and insulation. The system is framed with the aluminum logs. The single layer of typical glass covered with thickness of 0.4 cm is applied on the top surface of the collector. Backside and edges of the collector are insulated with 5 cm of mineral wool. These minimize heat losses. In operation, air flows through the space between absorber and back plate set just above the bottom/back insulation. The heated air then goes up to the drying chamber. Used dryer air specifications are 150-watt, 3300 rpm. The distance between absorber and glass cover is 4 cm, and between absorber and back plate is 8 cm. The solar collector system faced south and tilted 45°from the horizontal level.

Drying cabinet was manufactured in the cylindrical section in a laboratory located in Al Nasr Street - Baghdad. Its length was 1 m and diameter, 1 m; it was made of aluminum containing three trays. The distance between the tray and the other is 0.20 m, being linked to the drying chamber and solar collector by duct air flow dimensions of 0.15 *0.15 *0.45 m, which was developed within the fan duct at the foreward side of the chamber to pull air from inside the collector and push it into the chamber.

EXPERIMENTAL SET-UP

The performance of the solar drying system was evaluated by conducting tests at three volumetric air flow rate (0.0405, 0.0540 and 0.0675 kg/s) during three days to dry 70 kg of beans. The following parameters were measured: (a) radiation incident on the solar collector, (b) air temperatures at various locations in the collector and dryer, and moisture content of mass dried. International standards (ASABE/ASAE S423 DEC, 2007) for testing the performance of solar air collector were used. To measure the temperature of air at various locations of the collector and dryer, K-type thermocouples were installed at various points along the length and breadth of solar drying system. The drying test started at 8:00 am and stopped at 16:00 pm.

THERMAL ANALYSIS

The structure of a double flow solar air heater with single glass was covered and the energy flow diagram of such system is shown in Figure 2.

The two air streams are flowing equally steadily and simultaneously above and below the absorber plate. The energy equations are considered under equal flow rate of flowing fluid, convective heat transfer co-efficient (h1 = h3) and average fluid temperature ($Tf\ 1=Tf\ 2=Tf$). The energy equations are given for the absorber plate, cover plate and the air streams, respectively. For absorber plate;

$$Q_u = S - U_L(T_1 - T_a) - U_B(T_3 - T_a) \tag{1}$$

For the cover plate;

$$U_T(T_1 - T_a) = h_1(T_{f1} - T_1) + h_{r21}(T_2 - T_1) \tag{2}$$

$$U_B(T_3 - T_a) = h_1(T_{f2} - T_3) + h_{r23}(T_2 - T_3) \tag{3}$$

For the air stream;

$$Q_u = h_2(T_2 - T_{f1}) + h_2(T_2 - T_{f2}) - h_1(T_{f1} - T_1) - h_1(T_{f2} - T_3) \tag{4}$$

This equation can be rewritten as:

$$(T_1 - T_{f1}) = [U_T(T_a - T_{f1}) + h_{r21}(T_2 - T_{f1})]/(U_T + h_1 + h_{r21}) \tag{5}$$

Also equation (3) can be expressed as:

$$(T_3 - T_{f2}) = \frac{[U_L(T_3 - T_{f2}) + h_{r21}(T_2 - T_{f2})]}{(U_L + h_1 + h_{r23})} \tag{6}$$

Introducing an equivalent radioactive heat transfer coefficient (hr) which can be expressed as:

$$h_r = \frac{4\sigma T_{av}^3}{\left(\frac{1}{\in_p} + \frac{1}{\in_b} - 1\right)} \tag{7}$$

Where $Tav = \frac{1}{2(Tp + Tb)}$

Equations 1 to 4 can be expressed as in one equation

$$(2h_2 + h_{r21} + h_{r23})(T_1 - T_f) = S + h_{r21}(T_1 - T_f) + h_{r23}(T_3 - T_f) \tag{8}$$

Also equation (7) can be written in this form with the help of equation (5) and (6).

$$(T_2 - T_f) = (SN + T_a - T_f G) \tag{9}$$

Where M, N and G are the functions of the heat transfer parameters for the collector and its values are

$$M = [(2h_2 + h_{r21} + h_{r23})(U_T + h_1 + h_{r21})((U_B + h_1 + h_{r23}) \\ -(h_{r21}^2 U_B + h_1 h_{r21}^2 + h_{r21}^2 h_{r23} + h_{r23}^2 U_T + h_1 h_{r23}^2 + h_{r21} h_{r23}^2]$$

$$N = (U_T + h_1 + h_{r21})(U_B + h_1 + h_{r23})$$

$$G = [U_T U_B + h_{r21} + U_T h_1 h_{r21} + U_T h_{r21} h_{r23} \\ + U_B U_T h_{r23} + U_B h_1 h_{r23} + U_B h_{21} h_{r23}]$$

Equation (4) can be written in this form by using equation (8)

$$Q_u = 1/N[H_r(T_2 - T_f) + K(T_a - T_f) \tag{10}$$

Where

$$H_r[2h_2N + h_1 h_{r21} U_B + h_1^2 h_{r21} + h_1 h_{r21} h_{r23} + h_1 h_{r23} U_T + h_1^2 h_{r23} + h_1 h_{r21} h_{r23}]$$

$$K = [h_1 U_T U_B + h_1^2 U_T + h_1 h_{r23} U_T + h_1 U_B U_T + h_1^2 U_B + h_1 U_B h_{r21}]$$

The general performance equation (10) can be rewritten as:

$$Q_u = \frac{H_r}{M}\left[S - \frac{W_i}{H_r}(T_f - T_a)\right] \tag{11}$$

$$= F\prime[S - U_L(T_f - T_a) \tag{12}$$

Where F' = collector efficiency factor = Hr/M

U_L = overall heat loss of coefficient = Wi /Hr

And $W = \frac{(GH+MK)}{N}$

Collector heat removal factor (FR)

It is convenient to define a quantity that relates the actual useful energy gained if the whole collector surface were at the fluid inlet temperature. This quantity is called the collector heat removal factor (FR) and can be written as:

$$F_R = \frac{\dot{m}C_p[T_{fo} - T_{fi})}{A_c[S - U_L(T_{fi} - T_a)]} \tag{13}$$

The collector heat removal factor FR can be expressed as (Karwa et al., 1999).

$$F_R = \frac{\dot{m}C_p}{A_c U_L}\left[1 - \exp\left(\frac{F' U_L A_c}{\dot{m}C_p}\right)\right] \tag{14}$$

The quantity FR is equivalent to a conventional heat exchanger effectiveness which is defined as the ratio of the actual heat transfer to the maximum possible heat transfer.

Collector efficiency

The efficiency of the flat plate collector with an alternative working fluid was calculated using the heat gained by air with respect to the actual solar energy received by the flat plate collector.

Overall efficiency of the system = (heat gained by air/input solar energy) (15)

Heat gained by the air = *m cp(to-ti)*

Input solar energy (solar energy falling on the collector) = *S A T*

Heat transfer coefficients

In order to calculate the performance of an expended metal mesh solar air heater, the co-relations are required for calculating the values of convective heat transfer coefficients between air and the roughened absorber plate (h_1) and between the air and bottom plate (h_3), respectively.

The following co-relation is used to calculate h_1, in turbulent flow conditions (Karwa et al., 1999).

$$Nu = 0.0158(Re)^{0.8}$$

The heat transfer co-efficient of artificially roughened absorber surface may be calculated using following correlation (Sharma et al., 1991):

$$N_u = 4x10^{-4}(Re)^{1.22}(e/D_h)^{0.625}(s/10e)^{2.22}(l/10e)^{2.66} \tag{16}$$

$$x[exp.\{-1.25(ln(s/10e))^2\}]x[exp.\{-0.824(ln(l/10e))^2] \tag{17}$$

Where

$$\mathrm{Re} = \frac{\dot{m} D_h}{\mu A_c} \qquad (18)$$

The characteristic length is the hydraulic diameter of the duct which can be expressed as

$$D_h = \frac{4 \text{ x cross-sectional area of the flow channel}}{\textit{wetted} \text{ perimeter of the flow channel}} = \frac{4(W \text{ x } H)}{2(W + H)} \qquad (19)$$

The collector overall loss co-efficient (UL)

The collector overall loss co-efficient, UL is the sum of the top, bottom and side loss coefficients

(UL = UB + UT + US) (20)

Bottom heat loss coefficient and side loss coefficient are considered as zero.

UL = 0+UT +0

UL = UT

An equation for UT is established by theoretical analysis in which constants are employed (Gatea, 2010).

$$Ut = \left(\frac{N}{\frac{C}{Tp}\left(\frac{TP - Ta}{N + f}\right)^{e}} + \frac{1}{hw} \right)^{-1} + \frac{\sigma(Tp^2 + Ta^2)(Tp + Ta)}{\frac{1}{d} + \frac{2N + f - 1}{\varepsilon g} - N} \qquad (21)$$

$$C = 520(1 - 0.000051\beta^2,$$

$$d = \varepsilon p + 0.0425N(1 - \varepsilon p)$$

$$f = (1 + 0.089hw - 0.1166hw\ \varepsilon p)(1 + 0.07866N),$$

$$hw = 5.7 + 3.8u$$

e = 0.252

Quantity of air needed for drying

The drying of any material involves migration of water from the interior of the material to its surface, followed by removal of the water from the surface. The rate of movement differs from one substance to another. These differences are greatest between hygroscopic and non- hygroscopic materials. For non-hygroscopic materials, drying can be carried out to zero moisture content, as for example, the textile materials being dried in a laundry. Hygroscopic material, such as grains, fruits, and foodstuffs in general, will have residual moisture content. There is then equilibrium between the vapor pressure of the air and that of the material being dried and the drying rate becomes zero. It may be necessary in drying to reduce the rate of drying to prevent cracking of the surface. In most drying operations, the heat comes from the air itself, which is cooled by the evaporation; this relationship (latent for evaporation heat given up by air) can be expressed by the following (Milan and Stakic, 2000):

$$Mw\ hfg = ma\ cp\ (T2 - T1) \qquad (22)$$

The quantity of water is calculated from the initial and desired final moisture content with the help of the following Equation (23) (Gunasekarm, 1986):

$$Mw = mg\left(\frac{mo - mf}{100 - mf}\right) \qquad (23)$$

Efficiency of solar dryers

The efficiency of solar drying systems can be evaluated either based on the thermal performance or drying rates of the products. The process based on drying rates is associated with a number of variables involved and is much complex and tedious for calculation 3. This study is an attempt to evaluate the thermal performance of different solar dryers.

The thermal efficiency of solar dryer can be defined as the thermal energy utilized for drying over the thermal energy available for drying.

The term ‘thermal energy utilized for drying’ includes the following parameters:

1) Sensible heat used to raise the temperature of the food, given by

$$wf\ cpf\ (t2 - t1) \qquad (24)$$

2) Sensible heat used to raise the temperature of the water in the food, given by

$$ww\ cpw(t2 - t1) \qquad (25)$$

3) Latent heat used to vaporize the water in the food, given by

$$Wv(hfgv)$$

Thus, the thermal efficiency of solar drying system is given by the relation 3 Heat

$$Efficiency\ SD = \frac{\text{Heat sensible(H\&F)input} + WvAv}{q\ \text{input}} \qquad (26)$$

RESULTS AND DISCUSSION

Solar collector performance

The performance of the solar air collector using three air flow rates has been tested. In this, Figures 3, 4 and 5 show temperatures of the atmosphere, air temperatures inlet and outlet of the solar collector absorber temperature and intensity of radiation per unit area. The highest temperature (71.4°C) of the outlet solar collector has been obtained at 11 am. At radiation intensity, 750 W/m² for air flow rate of 0.0401 kg/s was obtained, as seen in Figure 1. And minimum temperature (40.0°C) was obtained when air flow rate was 0.0675 kg/s. At radiation intensity 460 W/m² was obtained.

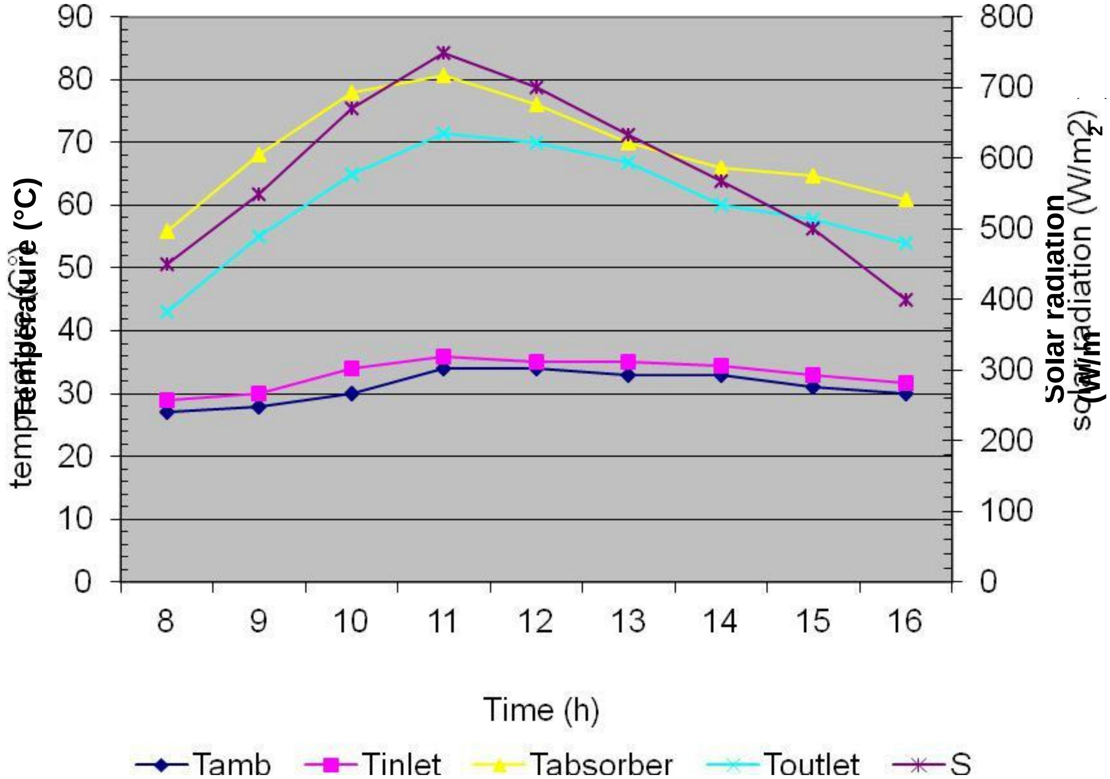

Figure 3. Variations of solar radiation, ambient temperature, absorber temperature, inlet air temperature and outlet air temperature with time a day for a typical experimental run during solar drying with air flow rate of 0.0405 kg/s (Date 1/3/2010).

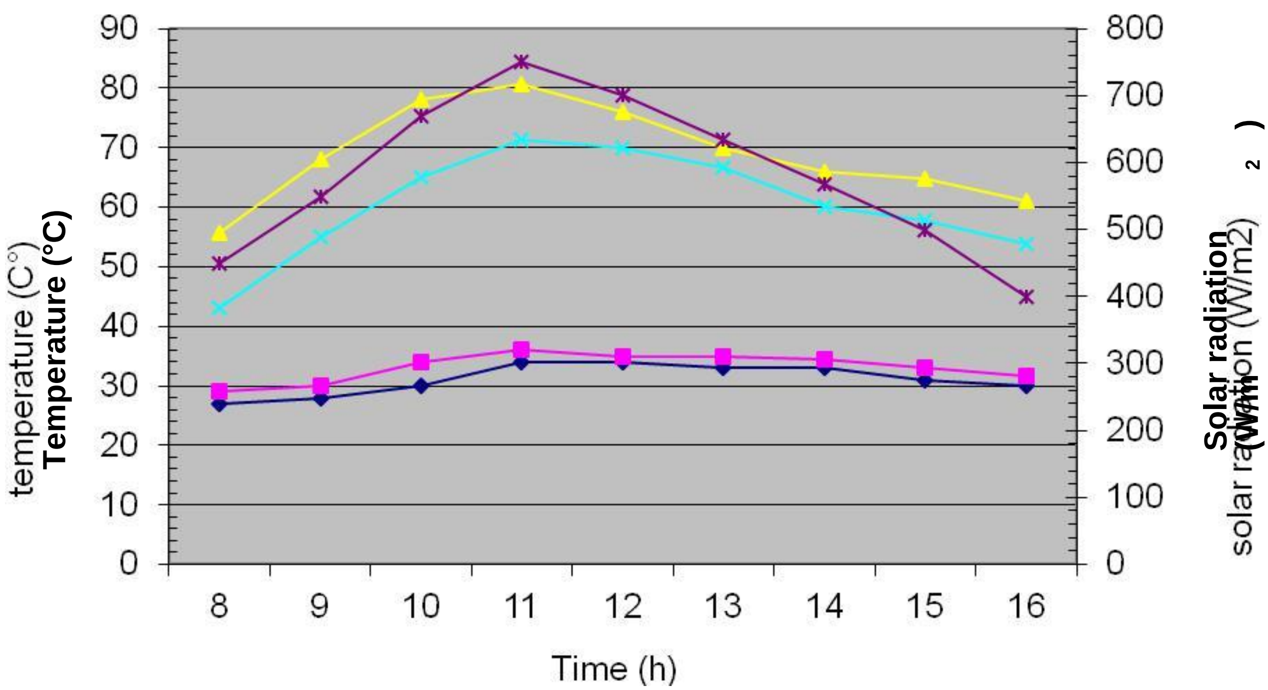

Figure 4. Variations of solar radiation, ambient temperature, absorber temperature, inlet air temperature and outlet air temperature with time a day for a typical experimental run during solar drying with air flow rate of 0.0540 kg/s (Date 2/3/2010).

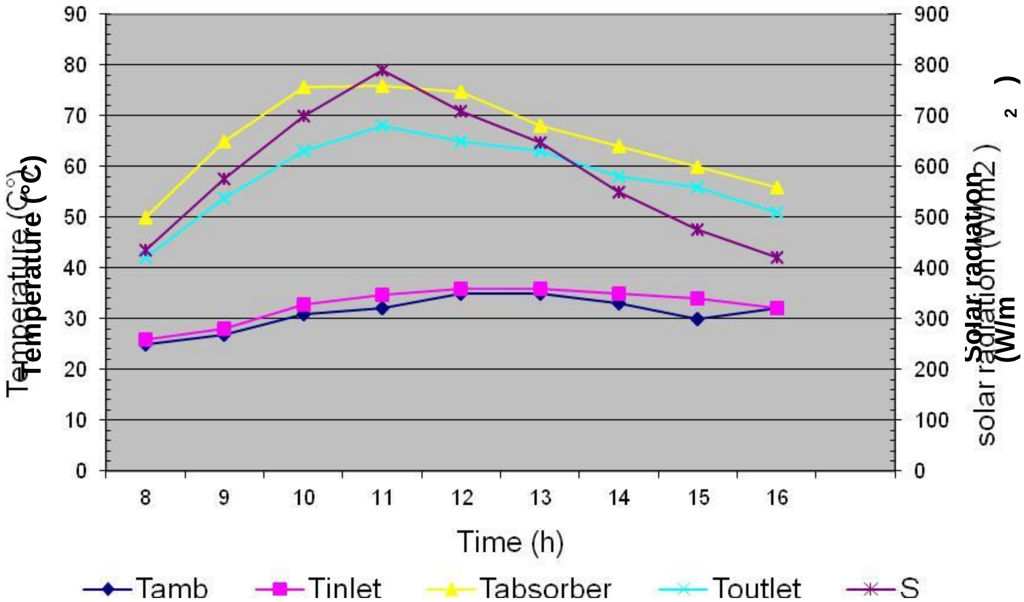

Figure 5. Variations of solar radiation, ambient temperature, absorber temperature, inlet air temperature and outlet air temperature with time a day for a typical experimental run during solar drying with air flow rate of 0.0675 kg/s (Date 3/3/2010).

Table 1. Statistical analysis of the thermal efficiency of solar air collector using SPSS program.

Descriptive

					95% confidence intervals for mean			
Efficiency	**N**	**Mean**	**Std deviation**	**Std error**	**Lower bound**	**Upper bound**	**Minimum**	**Maximum**
0.0405	9	18.6378	2.4149	0.8050	16.7816	20.4940	13.18	22.15
0.0540	9	20.7500	1.9882	0.6627	19.2217	22.2783	15.94	22.64
0.0675	9	25.6456	1.1091	0.3697	24.7931	26.4981	23.68	27.46
Total	27	21.6778	3.5122	0.6759	20.2884	23.0672	13.18	27.46

ANOVA

Efficiency	**Sum of squares**	**df**	**Mean square**	**F**	**Significance**
Between groups	232.611	2	116.305	31.678	0.000
Within groups	88.115	24	3.671		
Total	320.726	26			

Collector efficiency

The collector efficiency was found to increase with increasing mass flow rate and the differences in temperature (to-ti), thermal efficiency were analyzed statistically using the SPSS program (Table 1). The highest average thermal efficiency (25.64%) at air flow rate of 0.0675 kg/s, and minimum average thermal efficiency (18.63%) at air flow rate of 0.0405 kg/s were obtained. Agenda through analysis of variance indicated that there were significant differences between the average thermal efficiency of the solar collector F = 31.67. As seen in the chart of the scheme above, the rate of air flow (0.0675) accesses the highest thermal efficiency.

Daily drying efficiency of the solar drying system

The comparison of moisture content of beans, the use of three air flow rate with time and the moisture content before drying (primary 70%) are shown in Figure 6. As seen from Figure 6, moisture content of beans was

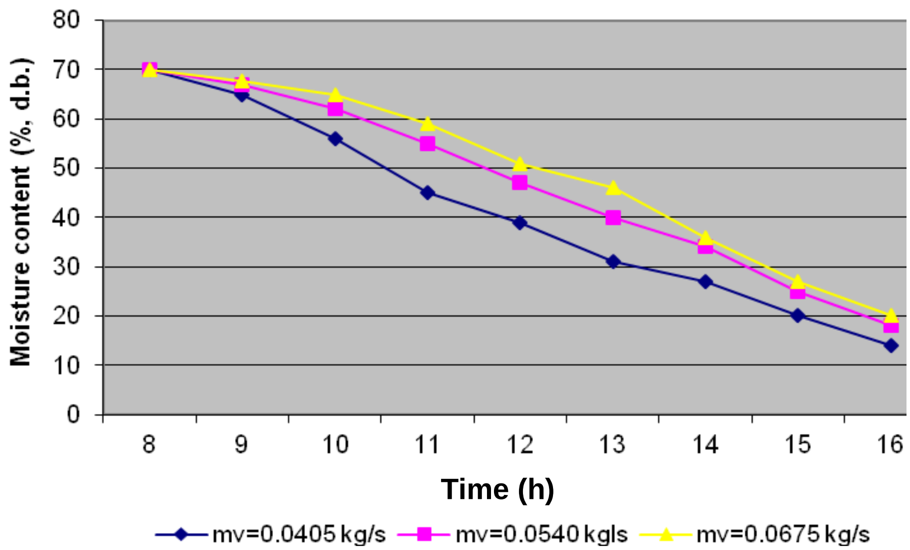

Figure 6. Comparison of moisture content of beans of solar dryer at different air flow rates.

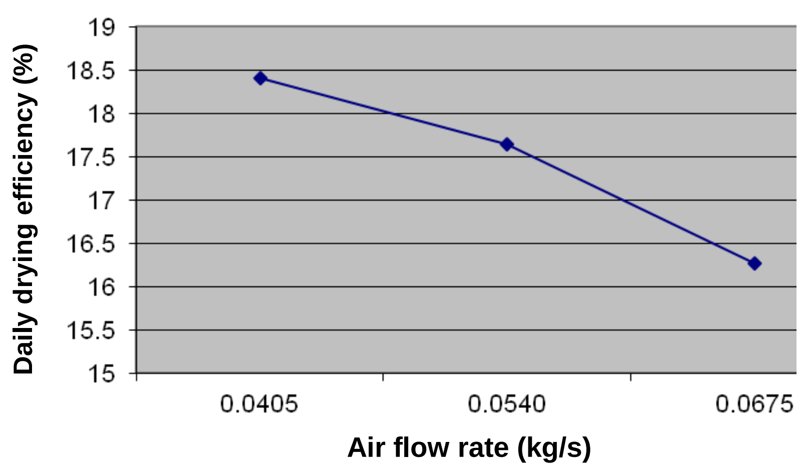

Figure 7. Variation of the daily drying efficiency for different air flow rates.

14% d.b at air flow rate of 0.0405 kg/s; 18% d.b at air flow rate of 0.0540 kg/s and 20% d.b at air flow rate of 0.0765 kg/s.

Variation of the daily drying efficiency of the solar drying system for different air flow rates is shown in Figure 7. The maximum daily efficiency was 18.41 % at air flow rate of 0.0405 kg/s and minimum was 16.27% at air flow rate of 0.0675 kg/s.

Conclusions

1) The maximum value of average thermal efficiency of the solar air collector obtained from the calculation is 25.64% at air flow rate of 0.0675 kg/s, and minimum average thermal efficiency is 18.63% at air flow rate of 0.0405 kg/s.

2) The moisture content of beans was 14% when the air flow rate was 0.0405 kg/s in nine hours.

3) The maximum daily efficiency drying system was 18.41 % at air flow rate of 0.0405 kg/s and minimum was 16.27% at air flow rate of 0.0675 kg/s.

4) The efficiency of solar drying system is affected by the properties of drying materials e.g. moisture content, size, shape and geometry as well as ambient conditions, which include solar radiation and temperature, relative humidity, velocity and atmospheric pressure of ambient air.

NOMENCLATURES

Ac: Surface area of collector (m^2)

š Angle tilted of solar collector (°)

ρ $^{-1 -1}$ C : Specific heat of air at constant pressure (kJkg K)

e: Height of roughness element (m)
FR: Friction factor for rough surface
H: Height of the duct (m)
Hr: Radiative heat transfer coefficient between two parallel plates
h1;h2: Convective heat transfer coefficient (W/m^2)
h 1,h 2: Convective heat transfer coefficient between air and roughness (W=m^2)
K: Thermal conductivity of air (Wm^{-1}K^{-1})
*H*w: Convection coefficient for the air between the top glass cover and environment W/m^2. k°
hfg: Latent heat of vaporization, (J/kg^{-1})
L: Length of collector (m)
I: Long way length of mesh (m)
m:Mass flow rate (kg/s)
mw: Water removed per kg of the air, (kg):
mg: Mass of grain in the dryer chamber,(kg):
M0: Initial moisture content, (%)
Mf: Final moisture content, (%)
Nu: Nusselt Number
N: Number of glass covers
Pm: Mechanical energy consumed for propelling air through collector, (W)
DP: Pressure drop across collector length (N/m^2)
Qu: Useful heat gain energy (W)
g: Product
r: Fraction of mass flow rate
Re: Reynolds number
S: Solar radiation (W/m^2)
s: Short way length of mesh (m)
T p: Absorber plate temperature (K)
Ta: Ambient temperature of air (K)
Tav: Average temperature of air (K)
Tin: Inlet temperature of air (K)
To: Outlet temperature of air (K)
Tf 1 and Tf 2: Fluid temperatures in upper and lower channel
T1, T2: The initial and final temperature of the given parameter, (K°)
UL: Overall loss of heat transfer coefficient
UT: Top loss heat transfer coefficient
UB: Bottom loss heat transfer coefficient
V: Velocity of air (m = s)
W: Width of collector (m)
Ŵ: Weight, kg
f: Food receiving radiant heat
τ : Transmittance of glass cover
α : Absorptivity of glass cover
ρ : Density of fluid (kgm^{-3})
μ: Dynamic viscosity of fluid
ε ρ : Emissivity of the absorbing plate
ε b: Emissivity of the bottom plate.
: Water
v: Water vapor

REFERENCES

Akinola AO (1999). Development and Performance Evaluation of a Mixed-Mode Solar Food Dryer. M. Eng. Thesis, Federal University of Technology, Akure, Nigeria.

Akinola AO, Fapetu OP (2006). Exergetic Analysis of a Mixed-Mode Solar Dryer. J. Eng. Appl. Sci., 1: 205-210.

Akinola, OA, Akinyemi AA, Bolaji BO (2006). Evaluation of traditional and solar fish drying systems towards enhancing fish storage and preservation in Nigeria. J. Fish. Int., Pak., 1(3-4): 44-49.

ASABE/ASAE S423 DEC 1991(R2007). Thermal performance testing of solar ambient air heaters. Am. Soc. Agric. Biological Eng.

Bassey MW (1989). Development and use of solar drying technologies, Niger. J. Solar Energy, 89: 133-64.

Ertekin C, Yaldiz O (2004). Drying of eggplant and selection of a suitable thin layer drying model, J. Food Eng., 63: 349-359.

Gatea AA (2010). Design, construction and performance evaluation of solar maize dryer. J. Agric. Biotechnol. Sustain. Dev., 2(3): 039-046.

Ikejiofor ID (1985). Passive solar cabinet dryer for drying agricultural products. *In:* O. Awe (Editor), African Union of Physics. Proc. Workshop Phys. Tech. Solar Energy Convers., Univ. Ibadan, Niger., pp. 157-165.

Karwa R, Solanki SC, Saini JS (1999). Heat transfer coefficient and friction factor co-relations for the transitional flow regime in rib-roughened rectangular ducts. Int. J.H. M.T., p. 42.

Kurtbas I, Turgut E (2006). Experimental investigation of solar air heater with free and fixed fins: efficiency and energy loss. Int. J. Sci. Technol., 1(1): 75-82.

Madhlopa A, Jones SA, Kalenga-Saka JD (2002). A solar air heater with composite absorber systems for food dehydration. Renewable Energy, 27: 27–37.

Milan B, Stakic (2000). Numerical Study on Hygroscopic Capillary-Porous Material Drying in packed bed. Thermal Sci., 4(2): 88-100.

Sharma SP, Saini JS, Verma HK (1991). Thermal performance of packed-bed solar air heaters, S.E., 47(2): 59-67.

Gunasekarm S (1986). Optimal Energy Management in Grain Drying, 25(1): 1-48.

Togrul IT, Pehlivan D (2004). Modelling of thin layer drying kinetics of some fruits under open-air sun drying process. J. Food Eng., 65: 413-425.

Waewsak J, Chindaruksa S, Punlek C (2006). A mathematical modeling study of hot air drying for some agricultural products. Thammasat Int. J. Sci. Technol., 11(1): 14-20.