ANALYSIS

55(284), August, 2019

ISSN 2278–5469 EISSN 2278–5450

DISCOVERY

Parametric study for Solar Dryer Design

Benjamin Ternenge Abur✉, Habou Dandakouta, Adisa AB, Ejilah RI

Department of Mechanical/Production Engineering, Abubakar Tafawa Balewa University, Bauchi.

✉**Correspondence author**
Department of Mechanical/Production Engineering, Abubakar Tafawa Balewa University,
Bauchi.
Email: engrbenjaminabur@gmail.com

Article History
Received: 30 May 2019
Reviewed: 02/June/2019 *to* 7/July/2019
Accepted: 10 July 2019
Prepared: 15 July 2019
Published: August 2019

Citation
Benjamin Ternenge Abur, Habou Dandakouta, Adisa AB, Ejilah RI. Parametric study for Solar Dryer Design. *Discovery*, 2019, 55(284), 430-440

ABSTRACT

Solar collector is a core sub-system in solar thermal energy systems with various applications including solar drying of agricultural products. This research paper studied the effect of input parameters that affect the collector size and efficiency of a solar cabinet dryer. For solar drying of tomatoes slices, design equations of collector top heat loss, bottom and edges heat losses, mass of moisture removed from crop, quantity of heat required for evaporating moisture from crop, drying chamber heat lost, estimation of average solar radiation for location using Angstron-Prescott equation, collector area and collector efficiency were coded in MATLAB programming language. The month of August was used as the design month bearing in mind that the month has the least solar radiation in Bauchi. Variables studied included single and double glazing, wind speed in the range of 1 to 5m/s, thermal conductivity of insulating materials between 0.01 to 0.1W/mK, collector tilt angle 0 to 60° and initial moisture content of crops in the range of 20 to 96%. The solar collector area and efficiency were calculated from the MATLAB programme codes. Result shows that wind speed of 1 and 2m/s have smaller collector areas and relatively higher collector efficiencies compared to wind speeds of 3, 4 and 5m/s for

both single and double glazing. Crops with low initial moisture content required small collector areas compared to those with high initial moisture content and optimum collector tilt angle of 13.01° for Bauchi. The use of insulating materials of low thermal conductivity minimizes heat lost and hence lower collector areas and higher collector efficiency. Collector air gap height of 50mm and optimize collector tilt angle of 13.01° in addition to materials of low thermal conductivity for thermal lagging maximizes solar energy input utilization, high thermal efficiency and small solar collector area. This design approach minimizes the use of materials, will maximized energy usage and give optimum returns on investment.

Keywords: Collector Area, Optimum Returns on Investment, Parametric Study, Solar Dryer Collector.

1. INTRODUCTION

Reduction of post harvest losses significantly contribute to the availability of food. A significant percentage of these losses are related to improper and/or untimely drying of foodstuffs such as cereals, grains, pulses, tubers, meat, fish, vegetables etc. Drying is a simple process of moisture removal from an agricultural product in order to attain the desired moisture content and can be achieved through several methods such as open-sun drying, solar drying and mechanized drying. Solar drying is an energy intensive operation and depends on climatic conditions (humidity, wind speed, solar radiation, cloud cover etc) and drying materials' properties (chemical composition, physical structure, size, shape etc) [10]. The removal of moisture prevents the growth and reproduction of microorganisms that causes decay, reduces the product weight and volume as well as the transportation cost. The drying of agricultural produce under the sun is a common practice most especially in developing countries. Traditional method of drying involves spreading the crops on concrete floor, mats, tarred surface roads and other forms of material and turned regularly until they are sufficiently dried to the desire moisture content. This method is characterized by a number of shortcomings including; lack of process control, non-uniformity in the dried product, soaking by rain, theft and vandalism, contamination by dust, rodents, and other domestic animals [9]. Also, prolonged open sun drying often causes deterioration of vital ingredients like vitamins, minerals and sensory characters of the dried product and thus, less market value (quality falls below local and international market standards). However, this method is more affordable to the farmers. Solar drying method converts sun's radiation into heat for drying of agricultural produce at a superior drying rate compared to the open sun drying method with improved quality of dried products at a relatively low cost. The solar collector supplies the product with hot airflow either generated naturally due to density changes resulting from temperature differences or through forced convection by incorporating a fan or blower in the drying system. Solar dryers can be constructed from locally available materials and are useful in areas where fuel or electricity is expensive, land for sun drying is insufficient or expensive, sunshine is plentiful but the air humidity is high. Besides, solar drying offers reduction to environmental risk compared to the mechanized drying method. Different types of solar dryers with varying sizes and designs have been developed and tested with some degree of efficiency. This work investigate the effects and sensitivity of parameters that affects collector size and efficiency and, provide a MATLAB programme code that could be used to design for minimum collector area required and maximum efficiency for drying.

2. METHODOLOGY

Parametric Design of Solar Dryer

Solar air heater is an integral part of the solar drying system and is basically made up of the glazing, thermal insulation, absorber plate, collector air inlet diameter for air flow into the channel and heat storage bed (optional) for improved performance. The above units including variables like collector tilt, depth of air passage in collector, wind speed, initial moisture content of crop and mass of crop to be dried are vital in determining the size and efficiency of the air collector. It is imperative to study the effect of these parameters in determining the size and efficiency of solar collector for improved performance.

Thermal Insulation

In solar collectors, the solar energy absorbed by the absorber plate is converted to useful energy gain while part of it is lost as thermal losses through the top, bottom and edges of the solar collector. With the assumption that all the losses are based on a common mean plate temperature T_p, the overall heat loss from the solar collector can be represented as given by [6]:

$$Q_{loss} = U_L A_c \left(T_p - T_a\right) \quad \text{... (1)}$$

Where;

U_L is solar collector overall heat loss coefficient, W/m^{2}K; T_a is the ambient temperature, K;

A_c is the collector area, m^2 .

The overall heat loss is the sum of the top losses, back losses and the edge losses given as: $Q_{loss} = Q_t + Q_b + Q_e$ ... (2)

Where;

subscripts t, b, and e represents the top, back and edge contributions of the heat losses respectively.

Collector Top Heat Loss for Single Glazing

The approximate overall top heat loss for the single glazing as proposed by [11]. It assumes that if the mean plate temperature T_p, the ambient temperature T_a, and convective heat transfer coefficient at the glass top h_w, are all known, then the overall heat transfer coefficient at the glass top cover U_t, which includes both the convection and the radiation effects is then computed using equation 3.

$$U_t^{-1} = \left[\left(h_{rpg} + h_{cpg}\right)^{-1} + \left(h_{rga} + h_w\right)^{-1} + \frac{l_g}{k_g} \right]^{-1} \quad \text{... (3)}$$

Where;

$$h_{rpg} = \frac{\sigma\left(T_p^{\ 2} + T_g^{\ 2}\right) + \left(T_p + T_g\right)}{\frac{1}{\varepsilon_p} + \frac{1}{\varepsilon_g} - 1} \quad \text{...(4)}$$

$$h_{cpg} = \frac{12.75\left[\left(T_p - T_g\right)\cos\beta\right]^{0.264}}{\left(T_p + T_g\right)^{0.46} L_{pg}^{\ \ 0.21}} \quad \text{... (5)}$$

$$h_{rga} = \sigma\varepsilon_g \left(T_g^{\ 2} + T_a^{\ 2}\right)\left(T_g + T_a\right) \quad \text{... (6)}$$

Where;

h_{rpg} is the radiation heat transfer coefficient between plate and glass, W/m^{2}k

h_{cpg} is the convection heat transfer coefficient between plate and glass, W/m^{2}k

h_{rga} is the radiation heat transfer coefficient between glass and the ambient W/m^{2}k

σ is Stefan–Boltzmann constant; T_g is glass cover temperature, °C

L_{pg} is air gap spacing between plate and glass, mm

β is collector tilt angle, °

ε_g is the emmitance of glass cover

l_g is the thickness of glass, m

k_g is the thermal conductivity of glass, W/mk.

Assuming the sky temperature to be same as that of the ambient temperature, the glass cover temperature, T_g, can be computed as given in equation 7 by [4]. However, for cases when the sky temperature is different from that of the ambient, the approximation proposed by [11] can be used.

$$T_g = \frac{f \times T_p + T_a}{1 + f} \quad \text{... (7)}$$

Where;

$$f=\frac{\left[12\times10^{-8}\left(T_a+0.2T_p\right)^3+h_w\right]^{-1}+0.3l_g}{\left[6\times10^{-8}\left(\varepsilon_p+0.028\right)\left(T_p+0.5T_a\right)^3+0.6l_{pg}^{-0.2}\left\{\left(T_p-T_a\right)Cos\beta\right\}^{0.25}\right]^{-1}} \qquad \text{... (8)}$$

ε_p is emmitance of plate 0.1≤ ε_p ≤0.95; T_a is ambient temperature, °C

l_{pg} air space between plate age glass cover, m

h_w is wind heat transfer coefficient (W/m^2 °C) 0≤ V ≤10m/s, if the wind velocity is in V (m/s), then: h_w = 5.7 + 3.8V ... (9)

Collector Top Heat Loss for Double Glazing

The top heat loss from a double glazing flat plate solar collector can be estimated based on the method also proposed by [11] as expressed in equation 10.

$$U_t=\left[\left(h_{rpg1}+h_{cpg1}\right)^{-1}+\left(h_{rg1g2}+h_{cg1g2}\right)^{-1}+\left(h_{rg2a}+h_w\right)^{-1}+\frac{l_{g1}}{k_{g1}}+\frac{l_{g2}}{k_{g2}}\right] \qquad \text{... (10)}$$

Where;

$$h_{rpg1}=\frac{\sigma\left(T_p^{\ 2}+T_{g1}^{\ 2}\right)+\left(T_p+T_{g1}\right)}{\frac{1}{\varepsilon_p}+\frac{1}{\varepsilon_{g1}}-1} \qquad \text{... (11)}$$

$$h_{cpg1}=\frac{12.75\left[\left(T_p-T_{g1}\right)Cos\beta\right]^{0.264}}{\left(T_p+T_{g1}\right)^{0.46}L_{pg1}^{\ 0.21}} \qquad \text{... (12)}$$

$$h_{rg1g2}=\frac{\sigma\left(T_{g1}^{\ 2}+T_{g2}^{\ 2}\right)+\left(T_{g1}+T_{g2}\right)}{\frac{1}{\varepsilon_{g1}}+\frac{1}{\varepsilon_{g2}}-1} \qquad \text{... (13)}$$

$$h_{cpg2}=\frac{12.75\left[\left(T_{g1}-T_{g2}\right)Cos\beta\right]^{0.264}}{\left(T_{g1}+T_{g2}\right)^{0.46}L_{g1g2}^{\ 0.21}} \qquad \text{... (14)}$$

$$h_{rg2a}=\sigma\varepsilon_{g2}\left(T_{g2}^{\ 2}+T_a^{\ 2}\right)\left(T_{g2}+T_a\right) \qquad \text{... (15)}$$

Where;

h_{rpg1} is the radiation heat transfer coefficient between plate and inner glass, W/m^{2}k

h_{cpg1} is the convection heat transfer coefficient between plate and inner glass, W/m^{2}k

h_{rg1g2} is the radiation heat transfer coefficient between inner and outer glass W/m^{2}k

h_{cpg2} is the convection heat transfer coefficient between plate and outer glass, W/m^{2}k

h_{rg2a} is the radiation heat transfer coefficient between glass and ambient, W/m^{2}k

T_{g1} and T_{g2} are the inner and the outer glass temperatures respectively, °C

L_{pg1} is air gap space between the absorber plate and the inner glass, m

L_{g1g2} is air gap space between the inner and outer glass, m.

The temperatures of the inner and the outer glasses T_{g1} and T_{g2}, can be approximated according to the method by [4] as expressed in equations 16 and 17.

$$T_{g1}=\frac{f_1\times T_p+T_{g2}}{1+f_1} \qquad \text{...(16)}$$

$$T_{g2} = \frac{f_2 \times T_p + T_a}{1 + f_2} \quad \text{... (17)}$$

Where; $$f_1 = \frac{\left[3.1\delta\left(T_{g2} + \frac{\left(1+\varepsilon_p\right)\left(T_p - T_{g2}\right)}{6}\right)^3 + \frac{0.8}{L_{g1g2}^{\ 0.2}}\left(\frac{\left(1+\varepsilon_p\right)\left(T_p - T_{g2}\right)Cos\beta}{6}\right)^{0.25} + 0.3L_{g1}\right]^{-1}}{\left[3.45\delta\varepsilon_p\left(T_p - \frac{\left(2-\varepsilon_p\right)\left(T_p - T_{g2}\right)}{6}\right)^3 \frac{0.8}{L_{pg1}^{\ 0.2}}\left(\frac{\left(2-\varepsilon_p\right)\left(T_p - T_{g2}\right)Cos\beta}{6}\right)^{0.25}\right]^{-1}} \quad \text{.(18)}$$

$$f_2 = \frac{\left[\left\{12\times10^{-8}\left(T_a + 0.2T_p\right)^3 + \right\}^{-1} + 0.3L_{g2}\right]\left(0.7 - 0.2\varepsilon_p\right)}{\left[6\times10^{-8}\left(\varepsilon_p + 0.028\right)\left(T_p + 0.5T_a\right)^3 + 0.6L_{pg1}^{\ -0.2}\left\{\left(T_p - T_a\right)Cos\beta\right\}^{0.25}\right]^{-1}} \quad \text{...(19)}$$

Where;

ε_p is the emmitance of plate

Bottom and Edges Heat Losses of the Solar Collector

The thermal losses coefficient for the bottom of a flat plate solar collector as [6]:

$$U_b = \frac{k_i}{x_i} \quad \text{.... (20)}$$

Where;

k_i is thermal conductivity of the insulating material, W/mk

x_i is thickness of the insulating material, m.

The edge loss coefficient depends on the area of the solar collector and is usually very small when compared to the losses of the top and bottom of the solar collector. The total heat loss coefficient can therefore be given as in the following expression by [6]:

$$U_L = U_t + U_b + U_e \quad \text{... (21).}$$

For uniform thickness of the thermal insulation for both edge and back of the collector, then equation 28 becomes $\left(U_e = U_b\right)$:

$$U_L = U_t + 2U_b \quad \text{(22)}$$

Amount of Moisture to be Removed from the Product, kg (m_w)

The moisture to be removed from the product to be dried is given by equation 23 as reported by [11].

$$m_w = m_p\left(\frac{m_i - m_f}{100 - m_f}\right) \quad \text{... (23)}$$

Where;

m_p is the dryer capacity per batch, kg

m_i is the initial moisture content of tomatoes, %

m_f is the maximum desired final moisture content of tomatoes, %.

Quantity of heat energy required for evaporating moisture from product, Q

The quantity of heat required to evaporate moisture from the product to be dried is given by [11] as:

$$Q = m_w \times h_{fg} \quad \text{... (24)}$$

Where;

Q is the amount of energy required for the drying process

m_w is mass of water vapour to be evaporated, kg

h_{fg} is latent heat of vaporization of water, kJ/kg

$$h_{fg} = 4186\left(597 - 0.56t_{pr}\right) \quad \text{... (25)}$$

Where;

t_{pr} is maximum allowable temperature of the dried product.

Area of the solar collector unit

According to [7], the model equation for calculating the area of a solar collector is given as:

$$A_c = \frac{Q}{F_R\left[(\alpha\tau)I_T - U_L \times t \times \left(T_f - T_a\right)\right]} \quad \text{... (26)}$$

Where; $(\tau\alpha)$ is the transmittance-absorptance product that represents the effective absorptance of the cover-plate system

I_T is total solar radiation incident on the collector, W/m^2

U_L is air heater overall heat loss coefficient, W/m^{2}K

t is the duration, s

T_f is the outlet collector temperature, K

T_a is the collector inlet temperature which is same as the ambient temperature, K

Q is amount of energy required for the drying process.

But in strict terms, Q should include the conductive heat lost from the drying chamber, Q_c

Drying Chamber Heat Lost

The conductive heat lost in the drying chamber is estimated using equation 27

$$Q_c = U_b \times t \times \left(T_f - T_a\right) \quad \text{... (27)}$$

Where; U_b is the back heat loss from the drying chamber

Total System Load

The system total load is given as in equation 28

$$Q_L = Q + Q_c \quad \text{... (28)}$$

Solar Collector Efficiency

The collector's efficiency, η_c is expressed as given by [9] as follows:

$$\eta_c = \frac{\rho V_a c_p \Delta T}{I_c A_c} \quad \text{... (29)}$$

Where;

c_p is the specific heat capacity of air, J/kgK

ΔT is the temperature elevation, K

I_c is the solar insolation, W/m^2

$\dot{V}_a$ is the volumetric flow rate, m^3/s

ρ is the density of air, kg/m^3

Estimation of Average Solar Radiation

The most convenient and widely used correlation for predicting solar radiation was developed by Angstrom and later modified by Prescott. The Angstrom formula is given by [12].

$$\frac{\bar{H}}{H_o} = a + b\frac{\bar{n}}{\bar{N}} \qquad ...(30)$$

Where;

$\bar{H}$ is the monthly average of the daily global radiation

H_o is the monthly average of the daily extraterrestrial radiation on a horizontal surface at the location

$\bar{n}$ is the monthly average of the sunshine hours per day

$\bar{N}$ is the monthly average of the maximum possible sunshine hours per day at the location

a and b are empirical constants for the location.

The monthly mean daily extraterrestrial irradiation and monthly mean day length can be derived from the equations 38 and 39 respectively [4].

$$H_o = \frac{24 \times 3600}{\pi} G_{sc}\left[1 + 0.033\left(360\frac{n}{365}\right)\right] \times \left[\sin\phi \sin\delta\left(\frac{2\pi\omega_s}{360}\right) + \cos\phi\cos\delta\sin\omega_s\right] \qquad ... (31)$$

$$\bar{N} = \frac{2}{15}\cos^{-1}(-\tan\phi\tan\delta) \qquad ... (32)$$

Where;

the hour angle ω_s, latitude ϕ and declination δ are related by:

$$\cos\omega_s = -\tan\phi\tan\delta \qquad ... (33)$$

Where;

δ is the solar angle of declination and is approximately given as:

$$\delta = 23.45\sin\left[\frac{360}{365}(284 + n)\right] \qquad ... (34).$$

The value 1367W/m^2 has been recommended for the solar constant ([1].

Where;

n is the day of the year from January 1 to December 31.

Solution Scheme of the Solar Dryer Design

The study of the solar dryer parameters for sizing and performance was carried out using MATLAB programming language. Design equations for collector top heat loss for single and double glazing, bottom and edges heat losses, amount of moisture to be removed from crop, quantity of heat required for evaporating moisture from crop, collector area, drying chamber heat lost, collector efficiency and estimation of average solar radiation were coded in MATLAB programming language. Input data of number of glass cover, wind speed, solar radiation available for Bauchi town in the design month of August, thermal conductivity of lagging materials, collector tilt angle, collector air depth passage, product design and ambient temperature, crop initial moisture content, latitude location and other input data's were used for the study. To facilitate the parametric design study of the solar dryer, refer to Table 1 for flat plate air heating system.

Table 1 Flat Plate Air Heater System Design Parameters

Parameter	Design value
Crop	Tomatoes
Mass of crop, kg	6
Initial moisture content, %	94
Final moisture content, %	4.5
Latitude of Bauchi, degree	7.7306
Thickness of glazing, mm	4
Transmissivity of glazing.	0.88
Thermal conductivity of glazing, W/mk	1.05
Air gap space between plate and inner glass, mm	10
Air gap space between inner and outer glasses, mm	10
Absorber plate (Aluminum)	
Emittance	0.09
Absorptance	0.91
Thickness of absorber plate, m	0.05
Thermal conductivity of plate, W/mk	205
Specific heat capacity of absorber plate, J/kgK	900
Back insulation thickness, mm	50

3. RESULTS AND FINDINGS

Glazing Number and Wind Speed

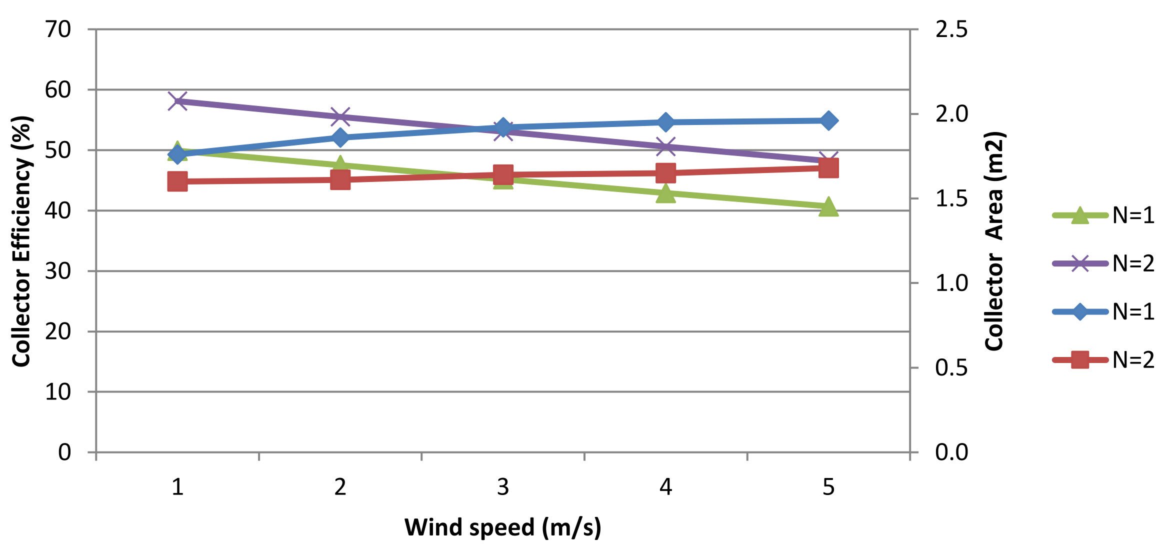

Figure 1 Effect of glazing and wind speed on collector area and efficiency

Fig. 1 shows the effect of glazing number and wind speed on collector area and efficiency. Double glazing minimizes collector top heat loss and thus, decreased collector area required compared to single glazing signifying more heat retention and utilization (efficiency). Comparatively, wind speed of 1 and 2m/s have low collector areas as against 3, 4 and 5m/s especially for single glazing indicating that heat is carried away more faster at high speeds (air flow rate). Thus, more collector area is required to complement for the lost heat in the collector for single glazing which transcends to relatively low collector efficiency. For average wind speed of 2.5m/s obtainable for Bauchi, collector area of 1.88 and 1.63m^2 are needed for single and double glazing. No significant difference in the collector efficiencies for wind speed of 3, 4 and 5m/s for double glazing.

Glazing Number and Thermal Conductivity of Insulating Materials

Substantial amount of heat could be loss through the back and edge of absorber plate if not properly or poor insulator is used. Lagging of collector systems with materials of synthetic origin of low thermal conductivity are not only expensive (rock wool, fibre glass, polystyrene and the polyurethane) but may pose health risk to those handling them in fibrous form. The thermal conductivity of insulating materials is proportional to collector area and inversely proportional to the collector efficiency for both single and double glazing (see Fig. 2). For this research, collector areas of 1.88 and 1.62m^2 are required for single and double glazing while, efficiency of 45.95 and 53.63% are obtained respectively using saw-dust which is readily available and relatively cheap with thermal conductivity of 0.08W/mK. For any application, the economy viability of using such an insulator is needed.

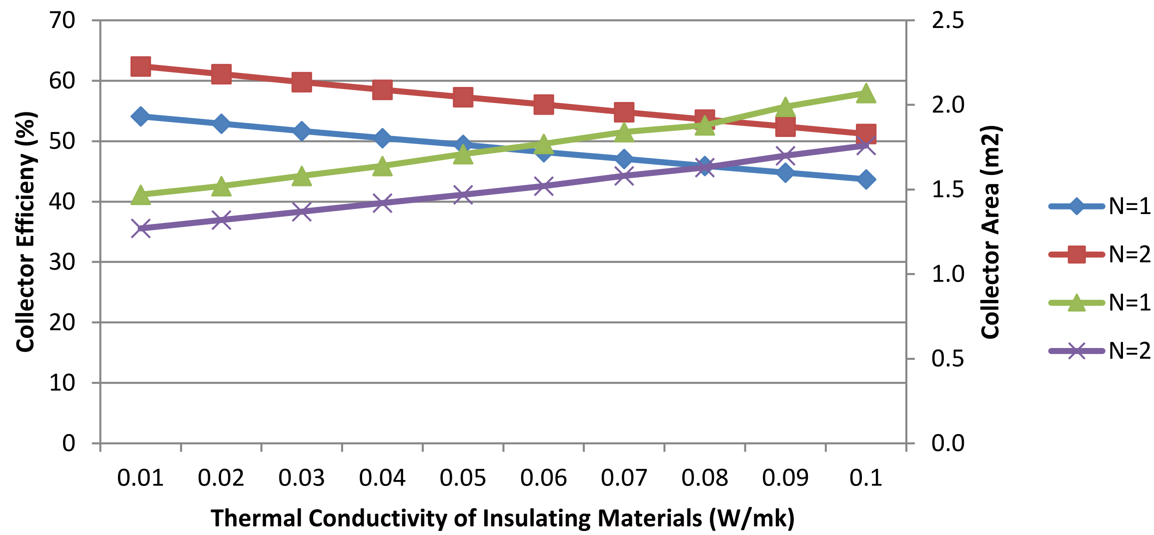

Figure 2 Effect of thermal conductivity of insualting material on collector aea and efficiency

Collector Tilt

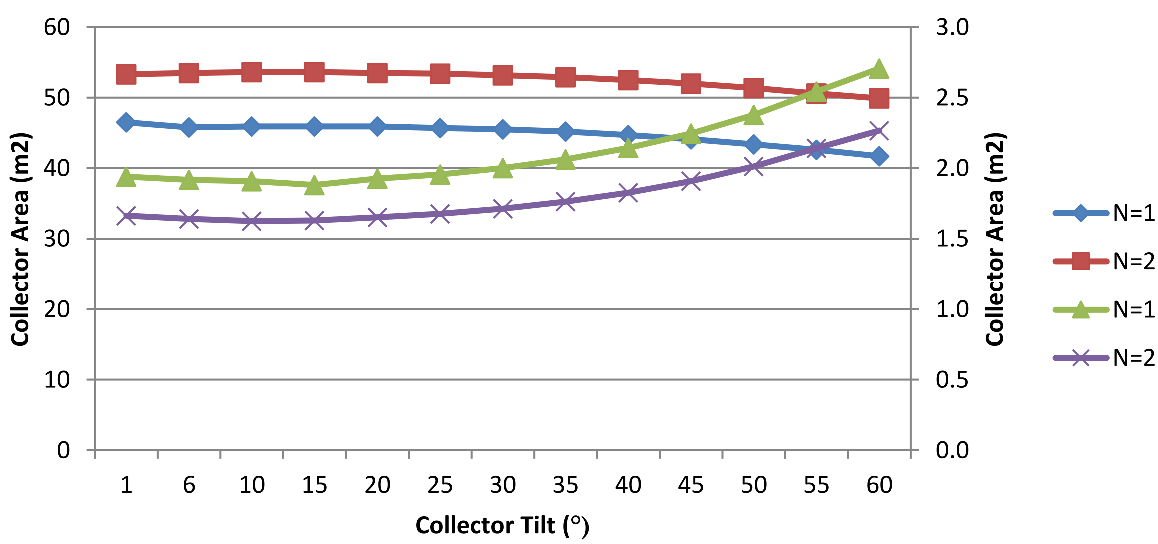

Figure 3 Effect of collector tilt on collector area and efficiency

One of the parameters that affect the performance of a flat plate solar collector is its angle of tilt with the horizontal as the variation of tilt angle affects the positioning of the collector to receive maximum amount of solar radiation. As the collector tilt increases from 0°, the collector area decreases and reaches a minimum at 1.88 and 1.63m^2 for single and double glazing corresponding to optimum tilt angle of 13.01° and system efficiency of 45.96% and 53.64% respectively (see figure 3). It is worth to note that solar collector using monthly average optimum tilt angle will received more maximum solar radiation than that using optimum annual tilt angle.

Initial moisture content of crop

Most commonly dried fruits and vegetables have initial moisture content that ranges from 72 to 94% which need to be dried to equilibrium moisture content of 7 to 4.5%, while cereals like maize, sorghum have about 30% initial moisture content and needs to be dried to a safe storage moisture content of about 14%. From Figure 4, it can be seen that the required collector area is proportional to the percentage of initial moisture content of the crop for both single and double glazed collectors. However, double glazed collectors require much smaller areas due to the reduction in the top heat loss coefficient as against single glazed collectors. At 94% initial moisture content for tomatoes, the collector areas required were found to be 1.88 and 1.62m^2 for single and double glazing, and efficiency of 45.96% and 53.64% respectively (see Fig. 4).

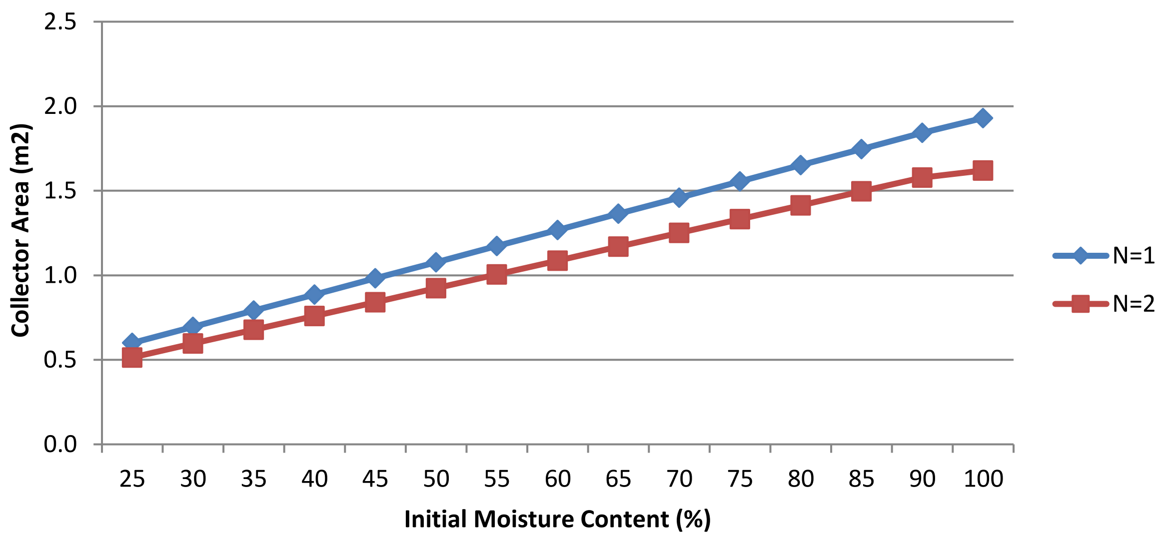

Figure 4 Effect of initial moisture content of crop on collector area and efficieny

4. CONCLUSION

For development of an efficient and effective solar drying system, studying some vital parameters that affect the size and efficiency of the integral part of the drying system (solar air heater) and the auxiliary heating system for continuation drying operation (if any) are essential. The MATLAB programme language could serve as a design format to be use for all crops and locations provided the changing variables are known and substituted into the programme.

REFERENCE

1. Abdul Azeez M.A, "Artificial Neural Network Estimation of Global Solar Radiation Using Meteorological Parameters in Gusau, Nigeria", Archives of Applied Science Research, Vol. 3, No. 2, pp. 586-595, 2011.
2. Ajadi, D. A., Sanusi, Y. K., Abiodun, A.O., Bioku J.O and Adeyemo I,, "Effect of Distance Between the Glass and the Absorber on Glass and Oven Temperatures of a Locally Designed Solar Cabinet Dryer", Journal of Scientific Research, Vol. 2, No. 1, pp. 5-10, 2014.
3. Akhtar N. and Mullick S.C, "Approximate Method for Computation of Glass Cover Temperature and Top Heat-loss Coefficient of Solar Collectors with Single Glazing" Solar Energy, Vol. 66, No. 5, pp. 349–354, 1999.
4. Duffie J. and Beckman W. (1991). Solar Engineering of Thermal Processes. John Wiley and Sons Inc., Second

Edition.

5. Duffie J. A. and Beckman W. A. (2013). Solar Engineering of Thermal Processes. John Wiley and Sons, Inc., Fourth Edition, Hoboken, New Jersey.
6. Ezekoye B. A. and Enebe O. M, “Development and performance evaluation of modified integrated passive solar grain dryer”, The pacific journal of science and technology. Vol. 7, pp. 185-190, 2006.
7. Garg H.P. and Prakash J, “Solar Energy: Fundamentals and Applications. Tata McGraw Hill Education Private Limited”, New Delhi, First Revised Edition, pp. 54, 2010.
8. Gutti B., Kiman S. and Murtala A. M, “Solar Dryer-An Effective Tool for Agricultural Products Preservation”, Journal of Applied Technology in Environmental Sanitation, Vol. 2, No. 1, pp. 31-38, 2012.
9. Medugu D. W and Yakubu D (2011). Estimation of mean monthly global solar radiation in Yola- Nigeria using angstrom model. Advances in Applied Science Research. 2(2): 414-421.
10. Michael A. O and Oluseun A.S, “Thermal diffusivity, Thermal effusivity and Specific heat capacity of Soils in Olorunsogo Power Plant, Southwestern Nigeria”, International Journal of Recent Research and Applied Studies, IJRRAS, Vol. 13, No. 2, pp. 501-521, 2012.
11. Samdarshi S.K, Mullick S.C, “Analysis of the top heat loss factor of flat plate solar collectors with single and double glazing”, International Journal of Energy Resources, Vol. 14, No. 9, pp. 975–990, 1990.