DEVELOPMENT OF A WIND-VENTILATED BOX-TYPE ABSORBER SOLAR AIR COLLECTOR

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ABSTRACT

This paper presents the design, construction and performance evaluation of a wind-ventilated box-type absorber solar air collector. A box absorber was used to transfer heat to the air stream via a larger surface area than ordinary flat-plate absorber and a rotary wind ventilator was incorporated to exploit the benefits of using forced convection solar dryers in the remote areas where electricity and other power source are non-existent. During the tests, maximum temperatures of 62.5 and 58.5^{0}C were obtained inside the collector and drying chamber, respectively, while the maximum ambient temperature obtained was 33.0^{0}C. The system drying efficiency increases as the air velocity through the system increases. The average air velocity obtained through the solar dryer was 1.76 ms^{-1} and the average drying efficiency of the system was 40.9%. The dryer performance was optimum from 12.00 hour to 15.00 hour. The maximum collector efficiency and average collector daily efficiency obtained were 55.2 and 41.0%, respectively.

Keywords: solar energy, wind-ventilator, collector, box-type absorber, dryer.

INTRODUCTION

The earth and its atmosphere receive continuously 1.7 x 10^{17} W solar radiation per year. A world population with a total power need of 10 kW per person would require about 10^{11} kW of energy per year. It is thus apparent that if irradiant on only one percent of the earth's surface could be converted into useful energy, then ten percent efficiency solar energy could provide all the energy needs of the people on earth. This figure is often quoted by solar energy enthusiasts, but unfortunately the nature of this energy source has technical problems and economical limitations that are not apparent from this microscopic view of the energy budget. In spite of these limitations, solar energy is essentially inexhaustible and potentially capable of meeting a significant portion of the nation's future energy needs with minimum of adverse environmental consequence (Bolaji, 2003; Nasir, 2004).

Solar energy is most promising of the unconventional energy sources and the use of the sun's energy for drying has been in practice from prehistoric times. However, the old practice of simply spreading the items to be dried in the open had proved very unsatisfactory, hence the search for alternative methods. Artificial dryers have

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long been in existence; some of them powered electrically or by fossil fuels. The ever-rising cost of electricity and fossil fuels have kept the operating cost of these dryers rising and indication is that the trend will continue (Liu et al., 1994; Itodo et al., 2002). Over the years, interest has developed on solar powered dryers to discover their potential for drying purposes. However, because of the diffuse nature of solar radiation, the efficiency of simple solar air collectors is not very high thus limiting their use (Bolaji, 2005).

The design of the flow duct and heat transfer surfaces of solar air collectors should therefore be executed with the objective of achieving a high efficiency with low friction losses. A number of investigators have studied air collectors with different configurations. Hamid and Beckman (1991) and Neeper (1999) have studied the matrix –type air collector. A detailed theoretical parametric analysis of a single-pass corrugated bare plate solar air collector has been made by Han (1994). Ligrani and Moffat (1996) have investigated the effect of different geometries of artificial roughness on the heat transfer and friction factor in duct flows. Their analyses and correlations were based on the correlation development by Webb et al. (1991), who related the friction coefficient to the wall similarity and the heat transfer coefficient to the heat-momentum transfer analogy of flow over a rough surface.

The finned absorber for the air collector was also studied by Liu et al. (1994), who demonstrated that extended surfaces on the absorber plate caused an increase in the heat transfer coefficient. Ong (1995)

presented a simple matrix inversion method of solution that obviated the need for complex algebraic manipulation of energy equations for different designs of air collectors. The designs studied were distinguished by air flow between the top glass and the absorbed plate, and by air flow between the absorber and the bottom plate.

Several other researchers have reported in their studies, that to achieve better drying rates, a proper circulation of heated air through the dryer is required. In the developing countries, the agricultural practice of drying is mostly undertaken in the rural areas where electricity does not exist to power the fan. Therefore, in this study a box-type absorber solar air collector for crop drying is designed with a rotary wind ventilator incorporated in to the dryer, which will increase the rate of air circulation through the dryer. The box-type absorber is used with the aim of increasing the heat transfer area exposed to the flowing air through the collector in order to achieve high temperature and high efficiency.

MATERIALS AND METHODS
Theoretical Considerations
A schematic diagram of a box-type absorber solar air collector is shown Fig. 1, in which the collector absorbing-surface receives the solar energy transmitted through the glazing system. The absorbed energy is transferred by convection to the air flowing through the channels; by radiation to the glazing and back plate; and by conduction to the metal vanes. The energy balance equations for the collector components are as follows:
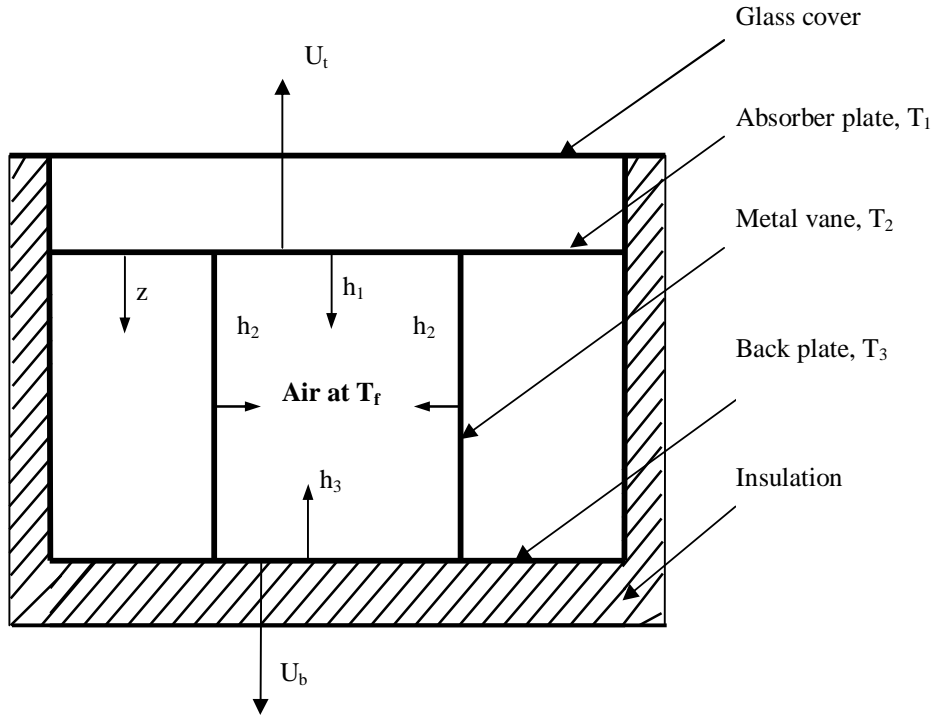
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Fig. 1: A schematic diagram of a box-type absorber solar air collector

For absorber plate (Bolaji, 1997):

$$U_t(T_1 - T_a) + h_1(T_1 - T_f) + h_r(T_1 - T_3) + \left(\frac{N}{A_c}\right)\left[-KA\left(\frac{dT_2}{dz}\right)\right]_{z=0} = S$$

(1)

where, A = cross section area of the metal vane (m^2); A_c = area of the collector (m^2); h_1 = heat transfer coefficient, as shown in Fig. 1 (Wm^{-2}k^{-1}); h_r = heat transfer coefficient as shown in Fig. 1 (Wm^{-2}k^{-1}); k = thermal conductivity (Wm^{-1}k^{-1}); N = the number of metal vanes per meter of width; S = absorbed radiation (Wm^{-2}); T_1 = absorber plate temperature (K); T_2 = metal vane temperature (K); T_3 = back plate temperature (K); T_a = ambient temperature (K); T_f = fluid temperature (K); U_t = top loss coefficient (Wm^{-2}k^{-1}); and z = distance of air in the duct from absorber plate (m).

For back plate (Bolaji, 1997):

$$h_r(T_1 - T_3) + \left(\frac{N}{A_c}\right)\left[-KA\left(\frac{dT_2}{dz}\right)\right]_{z=D} = h_3(T_3 - T_f) + U_b(T_3 - T_a)$$

(2)

where, D = air duct depth, (m); h_r = heat transfer coefficient as shown in Fig. 1 (Wm^{-2}k^{-1}); and U_b = bottom loss coefficient (Wm^{-2}k^{-1}).

For metal vanes:

$$\int_{z=0}^{z=D} 2h_2 l_c (T_2 - T_f)dz = \left[-KA\left(\frac{dT_2}{dz}\right) \right]_{z=0} - \left[-KA\left(\frac{dT_2}{dz}\right) \right]_{z=D}$$

(3)

where, l_c = length of the collector (m).
For air flow passage:

$$h_1(T_1 - T_f) + \frac{N}{A_c} \int_{s=0}^{s=D} 2h_2 l_2 (T_2 - T_f)dz + h_3(T_3 - T_f) = q_u$$

(4)

where, q_u = useful heat per unit area (Wm^{-2}).
Solving the differential part of Eq. (1) yields:
$T_1 - T_f = F_2^{-1}[\,F_3(T_3 - T_f) - U_t(T_f - T_a) + S]$ (5)
where the factors F_1, F_2 and F_3 are given as follows

$$F_1 = \left(\frac{N}{A_c}\right)\left[\frac{(2h_2 l_c kA)^{1/2}}{\sinh mD}\right] = \left(\frac{N}{A_c}\right)C_1$$

(6)

$F_2 = U_t + h_1 + h_r + F_1(\cosh mD)$ (7)
$F_3 = h_r + F_1$ (8)

$$C_1 = \frac{(2h_2 l_c kA)^{1/2}}{\sinh mD}$$

where,   (9)

Similarly, solving the deferential part of Eq. (2) yields
$F_3(T_1 - T_f) - F_4(T_3 - T_f) - U_b(T_f - T_a) = 0$ (10)
where, $F_4 = h_r + F_1(\cosh mD) + h_3 + U_b$ (11)

Eqs. (5) and (10) can be solved algebraically and the final results are given as follows:

$$(T_1 - T_f) = \left(\frac{F_3 U_b + U_4 F_4}{F_3^2 - F_2 F_4}\right)(T_f - T_a) - \left(\frac{F_4}{F_3^2 - F_2 F_4}\right)S$$

(12)

$$(T_3 - T_f) = \left(\frac{F_2 U_b + U_4 F_3}{F_3^2 - F_2 F_4}\right)(T_f - T_a) - \left(\frac{F_3}{F_3^2 - F_2 F_4}\right)S$$

(13)

Substituting Eq. (3) into Eq. (4) and rearranging of terms gives the useful heat which can be taken from the collector as
$q_u = F_5(T_1 - T_f) + F_6(T_3 - T_f)$ (14)
where, $F_5 = h_1 + F_1[(\cosh mD) - 1]$ (15)
and $F_6 = h_3 + F_1[(\cosh mD) - 1]$ (16)

Finally, substituting Eqs. (12) and (13) into Eq. (14) yields

$$q_u = \left(\frac{F_4 F_5 + F_3 F_6}{F_2 F_4 - F_3^2}\right)\left\{S - \left[F_5(U_b F_3 + U_4 F_4) + \frac{F_6(U_b F_2 + U_4 F_3)}{(F_4 F_5 + F_3 F_6)}\right](T_f - T_a)\right\}$$

(17)

Eq. (17) is similar to the general expression for the useful heat, Q_u, from the solar collector, given by Duffie and Beckman (1991):

$Q_u = A_c F'[atI - U_L(T_f - T_a)]$

or $q_u = F'[atI - U_L(T_f - T_a)]$ (18)

where,
$$F' = \frac{F_4 F_5 + F_3 F_6}{F_2 F_4 - F_3^2}$$

(19)

$$U_L = F_5(U_b F_3 + U_4 F_4) + \frac{F_6(U_b F_2 + U_4 F_3)}{(F_4 F_5 + F_3 F_6)}$$

(20)

atI = S,
a = solar absorptance
t = transmittance
I = incident solar radiation (W/m^2)

The collector efficiency is obtained by using the relation given in Eq.   (21)

$$\eta_c = \frac{q_u}{I} = F'\left[\alpha\tau - U_L \frac{(T_f - T_a)}{}\right]$$

(21)

The useful heat gained by the air, Q_g, is given as (Youcef-Ali et al., 2001):
$$Q_g = \dot{m}_a C_{pa}(T_o - T_i)$$

(22)

Therefore, the drying efficiency (h_d) is expressed as:
$$\eta_d = \dot{m}_a C_{pa} \frac{(T_o - T_i)}{I}$$

(23)

where, C_{pa} = Specific heat capacity of air (kJ/kg.K); $\dot{m}_a$ = mass flow rate of air (kg/s); T_o = temperature of outgoing air (K); and T_i = temperature of incoming air (K).

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METHODOLOGY

Experimental Set -Up

Figure 2 shows the diagram of the solar dryer. The dryer has five main features viz: the box-type absorber solar air collector, the drying chamber, the drying trays, the chimney and the wind ventilator.

Box-type Absorber Air Collector

The frame of the absorber is made of aluminium 2 mm thick, painted black and it consists of air flow channels enclosed by an upper absorbing plate, a back plate and metal vanes. The effective area of the collector is 0.412 m^2. It has heat insulation lining made from foam material of about 40 mm thick. The window is covered with transparent glass of thickness 4 mm. One end of the collector has an air inlet vent of area 0.084 m^2, the other end opens to the drying chamber. The collector is south-facing and tilted 17.5^0 to the horizontal which is approximately 10^0 more than the local geographical latitude (Ado-Ekiti, Nigeria, location which is on latitude 7.5^0N).

The Drying Chamber

The drying chamber together with the structural frame of the dryer was built from well-seasoned woods which could withstand termite and atmospheric attacks.

It houses drying trays. Access door to the drying chamber was also provided at the back of the cabinet. This consists of three removable wooden panels made of 13 mm plywood, which overlapped each other to prevent air leakages when closed. The drying chamber is also lined with foam insulation material 4 mm thick to reduce heat loss.

Drying Trays

The drying trays are contained inside the drying chamber and were constructed from a double layer of fine chicken wire mesh with a fairly open structure to allow drying air to pass through the food items.

The Chimney

It is located to the top of the drying chamber. It is a round ducts with diameter and height of 100 and 350 mm, respectively. It acts as air passage from the drying chamber to the wind ventilator.

Wind Ventilator

As shown in Fig. 2, a rotary wind ventilator is located at the top for effective circulation of heated air through the cabinet dryer. The wind ventilator is a corrugated vane rotor, which sucks air from the ventilator stack as it spins.
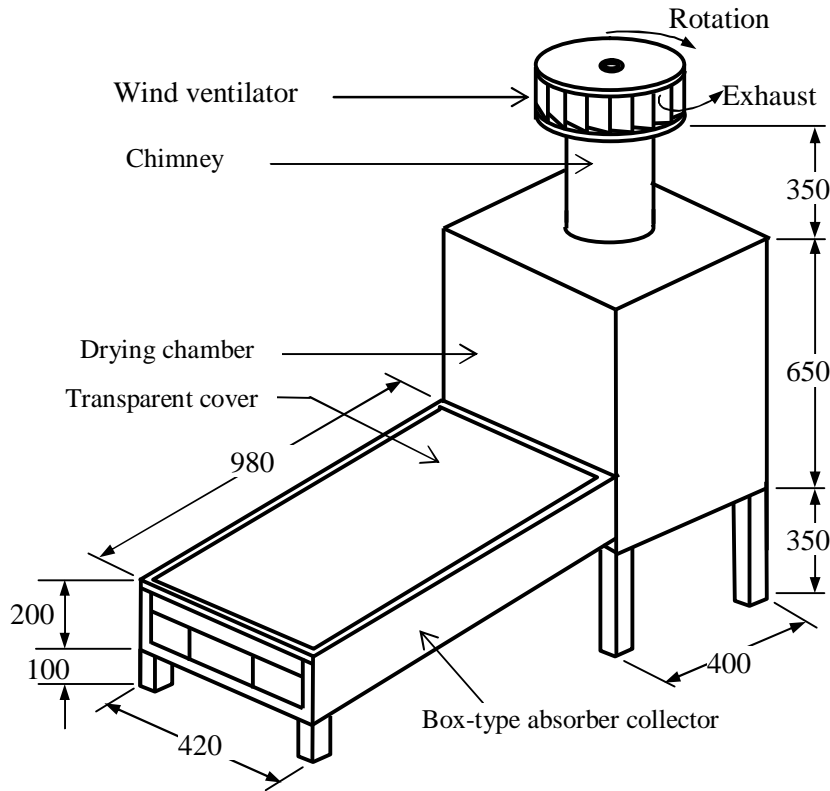
Fig. 2: Solar crop Dryer with box-type absorber Collector.

RESULTS AND DISCUSSION

The wind ventilated box-type absorber solar dryer was tested in the month of September 2003. It was used to dry maize and during the drying period, the incident solar radiation intensity was measured using pyranometer. The temperatures of collector inlet air, collector outlet air, drying chamber, chimney exhaust and ambient air were measured hourly with mercury-in-glass thermometer between the hour of 08.00h and 18.00h local time. The air velocity through the dryer was measured with a portable hand anemometer. The useful energy gain per unit area, the collector thermal efficiency and drying efficiency were calculated using Eqs. (18), (21) and (23), respectively.

Figure 3 shows a typical day hourly variation of temperatures in the dryer and solar radiation. The maximum average temperature obtained (during the daylight) inside the collector and drying chamber were 62.5 and 58.5^0C, respectively, while the maximum ambient temperature observed was 33.0^0C. The wide gap between the curve of drying chamber temperature and ambient temperature (Fig. 3) shows that the temperature inside the dryer was much higher than ambient temperature during most hours of the day-light and the dryer was hottest about mid-day when the sun is usually overhead. As also shown in Fig. 3, the high temperatures in the dryer correspond to the period of high solar radiation.
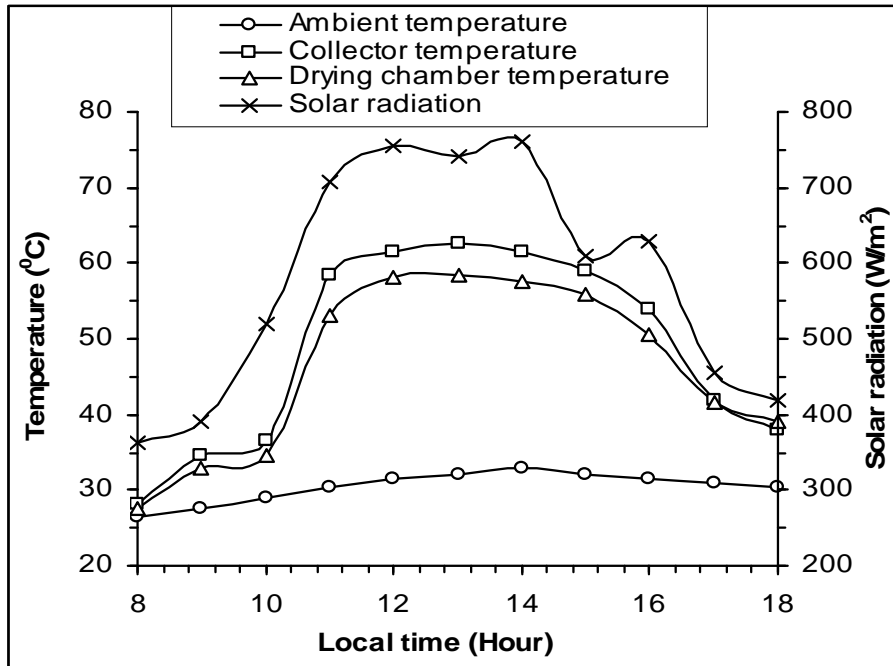
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Fig. 3. A typical day hourly variation of temperatures and solar radiation in the dryer
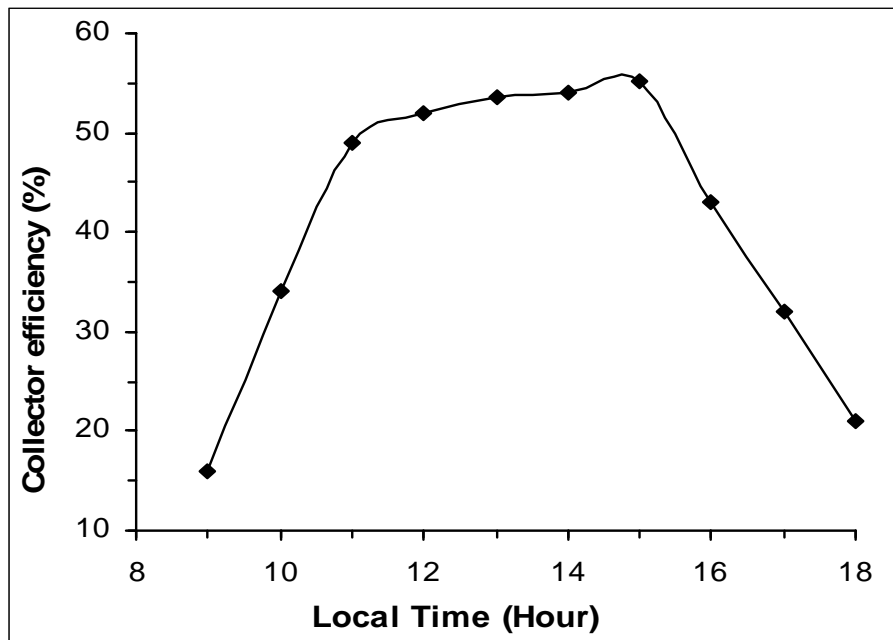

Fig. 4. Variation of collector efficiency with time

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The diurnal variation of the relative humidity of the ambient air and drying chamber are presented in Fig. 4 and when compared with Fig. 3 shows that the drying processes were enhanced by the heated air at very low humidity. Figure 5 shows the effect of air velocity on the drying efficiency. The system drying efficiency increases as the air velocity increases, which shows the dependence of the dryer performance on the air velocity. The average air velocity through the solar dryer during the period of test was 1.76 ms^{-1} and the average day-light efficiency of the system was 40.9%.

Figure 6 shows the graph of collector efficiency against local time. As shown in this figure the collector efficiency increases steadily with time between 8.00 hour and 11.00 hour and attained peak values between the hours of 12.00 to 15.00, and decreases steadily with time as the sun goes down in the late afternoon. This shows that the dryer performance was optimum in the afternoon from 12.00 hour to 15.00 hour. The maximum collector efficiency and average daily collector efficiency obtained were 55.2 and 41.0%, respectively.
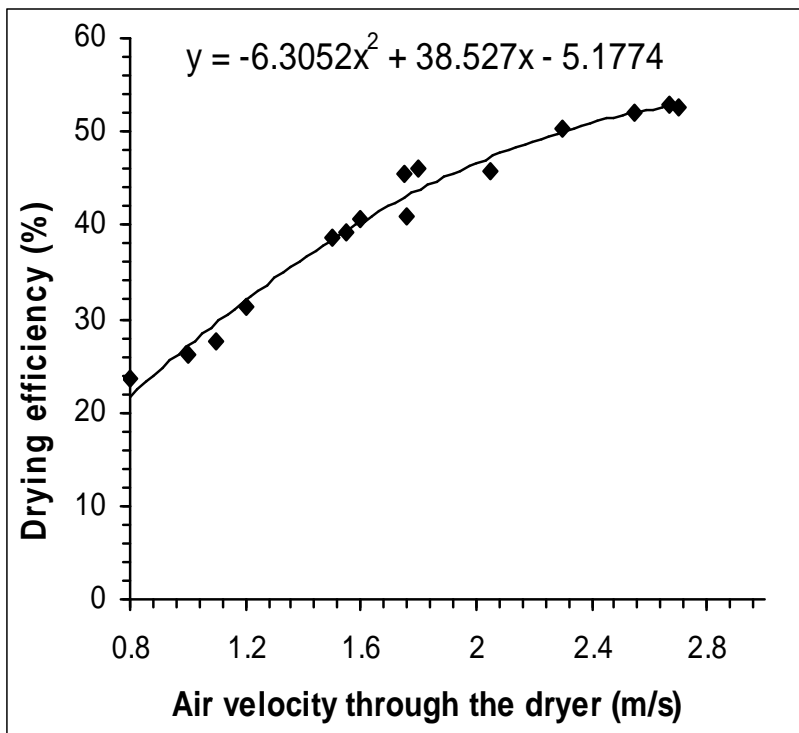

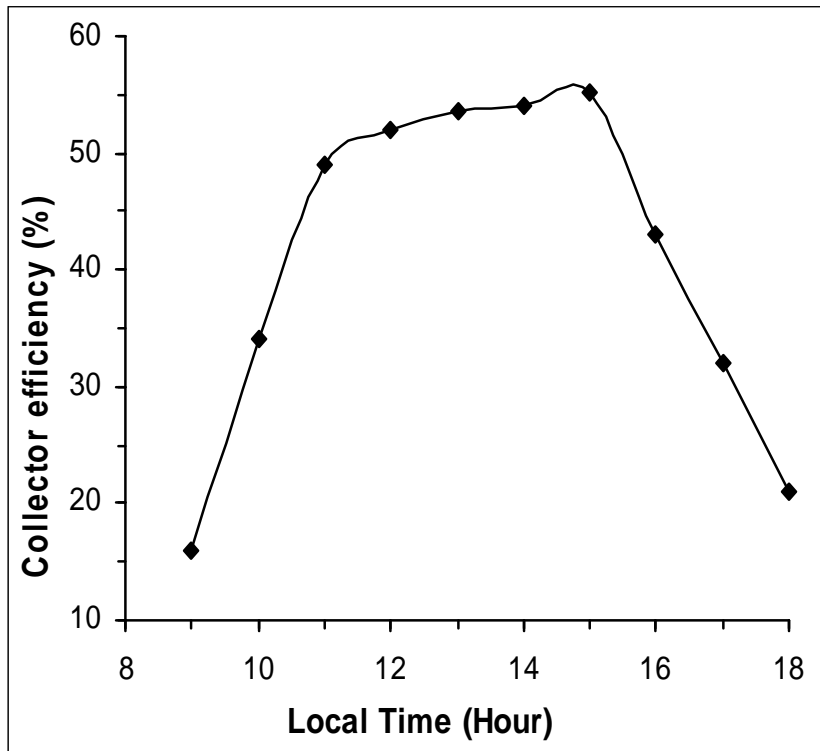**Fig. 5. Variation of drying efficiency with local time**

Fig. 6. Variation of collector efficiency with time

These results verified the high efficiency of the air collector with a box-type absorber when compare with the work of Awachie (1985) who worked on solar dryer with flat-plate absorber air collector and Matrawy (1998) who worked on solar dryer with fin-type absorber air collector. The maximum efficiencies obtained from the two types of collectors are 21 and 36%, respectively.

CONCLUSION

A wind-ventilated box-type absorber solar air collector for crop drying was designed and constructed using locally available materials. The test results show that high temperature of about 50^0C can be obtained in the solar dryer between the hours of 11.00 and 15.00 local time. The dryer temperature was much higher than ambient temperature during most hours of the daylight and the dryer was hottest about midday when the sun is usually overhead. On the basis of this study, it can be concluded that the temperature of the dryer with box-type absorber collector is always higher than the ambient temperature and drying can take place over a long time. The drying efficiency increases as the air velocity increases, which show that the rotary wind ventilator installed in the system has really improved the performance of the system.

The efficiency of the solar air collector can be enhanced by using box-type absorber. Comparison between the box-type absorber

solar air collector and other types of air collector used by other authors shows that the box-type is more efficient. The maximum efficiency obtained in the box-type absorber system was 55.2% while values obtained by other authors for both flat-plate absorber and fin-type absorber were 21 and 36%, respectively. The high efficiency of the present work is expected, because in the box-type absorber collector the heat is transferred to the flowing air via a larger area than in the other cases and the proper circulation of the heated air with the aid of wind ventilator through the dryer improved its performance. The fabrication of the dryer does not require high technology and once installed the maintenance cost is minimal.

NOMENCLATURE

A   cross section area of the metal vane (m^2)
A_c area of the collector (m^2) C_1 constant
D   air duct depth, diameter (m)F_1, F_2, F_3, F_4, F_5, F_6 collector factors
F'   collector efficiency factor
h   heat transfer coefficient ($Wm^{-2}k^{-1}$)
I   incident radiation (Wm^{-2})
k   thermal conductivity ($Wm^{-1}k^{-1}$)
l_c length of the collector (m)
N   the number of metal vanes per meter of width
q_u Useful heat per unit area (Wm^{-2})
Q_u Useful heat (W)
T   temperature
U   loss coefficient ($Wm^{-2}k^{-1}$)
S   absorbed radiation (Wm^{-2})
W   collector width (m)
z   distance from absorber plate (m)
a   solar absorptance
t   transmittance

Subscripts
1, 2, 3 absorber plate, metal vane and back plate, respectively,
a   ambient or air
b   bottom
c   collector
i   incoming
L   overall
o   outgoing
r   radiation
t   top

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