Thermodynamic Analysis of a Mixed-Mode Solar Dryer

I.A. Ikem[1,*], I.H. Aondover[2], O.E. Nyong[3], A.J. Adie[4]
[1](Department of Mechanical Engineering, Faculty of Engineering, Cross River University of Technology, Nigeria)
[2](Department of Mechanical Engineering, Faculty of Engineering, University of Agriculture, Nigeria)
[3](Department of Mechanical Engineering, Faculty of Engineering, Cross River University of Technology, Nigeria)
[4](Department of Mechanical Engineering, Faculty of Engineering, Cross River University of Technology, Nigeria)
E-mail address: azors9kee@yahoo.com
[*]Corresponding author: Tel: +79615819083

Abstract

Thermodynamic analysis of a mixed-mode solar dryer used for restaurant wastes is presented. This dryer is made up of up of flat-plat collector and a drying chamber. Thermodynamic properties of the working fluid were measured at input and exit points. Energy and exergy balance equations at the various segments of the dryer were formulated based on the balances in the solar grains dryer. The results show that energy and exergy efficiency are approximately in direct relation to the energy loss and exergy destruction. The dried product was tested suitable for animal consumption. These results are useful for design of solar grain dryer.

Keywords: flat-plate collectors; energy and exergy analysis; solar grain dryers; restaurant wastes; renewable energy.

Introduction

Drying is widely used in a variety of applications ranging from food drying to wood drying. Dryer supplies the product with more heat than is available under ambient conditions thus sufficiently increasing the vapor pressure of the moisture held within the product to enhance moisture migration from within the product and significantly decreasing the relative humidity of the drying air to increase its moisture carrying capability and to ensure sufficiently low equilibrium moisture content.

Energy from the thermodynamic point of view is made up of available and unavailable forms. Work done by a system is obtained from the available energy, while the unavailable form of energy remains unexploited (Ozegerner and Ozegerner, 2006; Kotas, 1995). The available form of energy of a system is convertible to maximum useful work, otherwise known as exergy as it comes to equilibrium with its environment from its original state (Coskun *et al*., 2009; Hou *et al*., 2007). This exergy is dependent on the thermodynamic properties of the working fluid – temperature and pressure (Coskun *et al*., 2009).

The thermodynamic analysis – energy and exergy of a thermal system is very important to engineers in order to optimize the efficiency of the system, minimize losses, reduce the operational and capital investment costs, and improvement of productivity of the thermal system (Riviere *et al*., 2009).

This paper presents a systematic energy and exergy analysis for a mixed-mode flat-plate solar dryer (MSD) employed in drying restaurant wastes, to ascertain its performance characteristics and quality of the dried product. This work was carried out in the department of mechanical engineering, University of Agriculture, Makurdi Nigeria, 2011. The necessity of this research stemmed from the fact that Makurdi, a town in the middle belt of Nigeria has abundant sun light and very rich in various agricultural products and animal husbandry. A lot of restaurant wastes which are normally thrown away never to be directly used again can be dried and given to the animals. As a lot of researches on solar food dryers concentrate on drying single products like pepper, okra, ground nut etc., it was necessary to apply it to dry restaurant wastes.

1.1 System Process Description

The mixed-mode solar dryer (MSD) in Figure 1 consists of an air-heater (AH), and a drying chamber (DC). The initial thermodynamic properties of air temperature, relative humidity and specific humidity (T_0, Φ_0, ω_0) are recorded as it flows into the inlet of AH through point (0), is heated by solar energy absorbed in the AH. These properties are changed from (T_0, Φ_0, ω_0) to (T_1, Φ_1, ω_1) at the collector outlet (point1). Here ($T_0 < T_1$), ($\Phi_0 < \Phi_1$) and ($\omega_0 = \omega_1$). The relative humidity of the air is lowered as its temperature increases to be able to absorb more moisture from the wet solid materials in the DC. The working fluid at (T_1, Φ_1, ω_1) enters the DC and comes in contact with the wet solid materials at the initial temperature and moisture contents (T_{s2}, X_2).

The wet solid is heated to (T_s, X) in the CD. The working fluid exits the CD at point (3) at lower temperature, higher relative humidity and specific humidity having (T_3, Φ_3, ω_3) with ($T_1 > T_3$), ($\Phi_3 > \Phi_1$) and ($\omega_3 > \omega_1$).

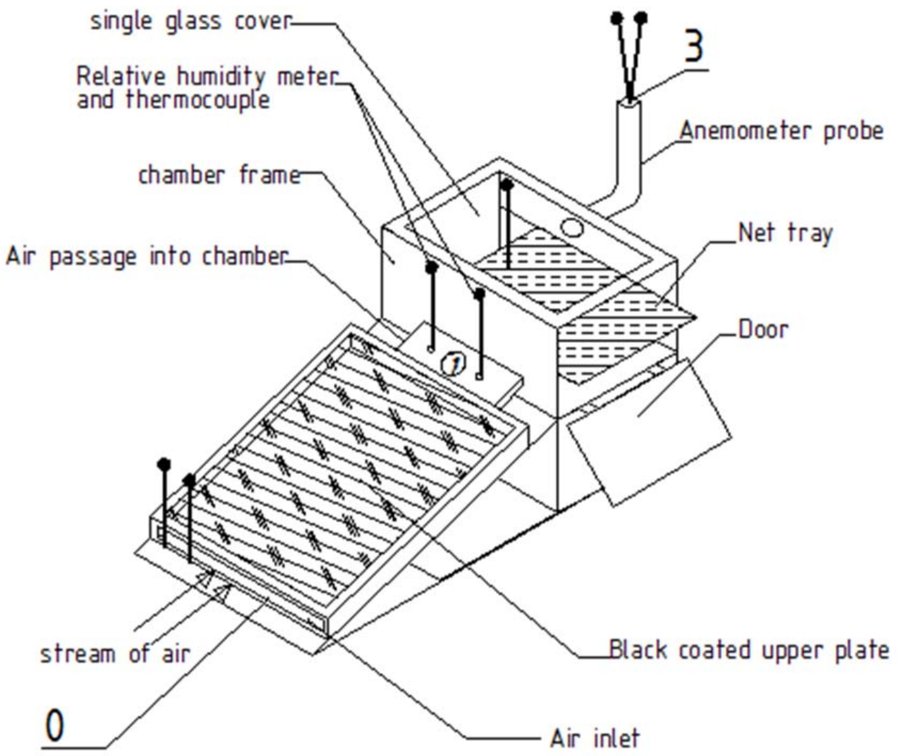

Figure 1(a) Schematic of Mixed-Mode Solar (MSD) Dryer

2.1 Analysis

This section presents energy and exergy analysis of drying processes. The system is illustrated with input and output terms in Figure 2, where there are four major interactions:

1. Input of drying air to the drying chamber to dry the products.
2. Input of moist products to be dried in the chamber.
3. Output of moist air after containing the evaporated moisture removed from the products.
4. Output of the dried products, with moisture content reduced to the desired level.

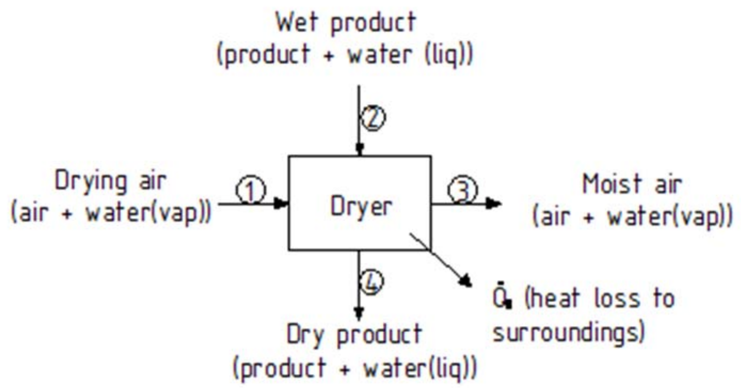

Figure 1(b) Schematic of a Drying Process Showing Input and Output Terms

1.2 Balances

Mass, energy and energy balances can be written for the above system, treated as a control volume.

1.3 Mass balances

We can write mass balance equations for the dryer given above for three flows: product, dry air and water.

Product: $\dot{m}_{p,2} = \dot{m}_{p,4} = \dot{m}_p$ (1)

Air: $\dot{m}_{a,1} = \dot{m}_{a,3} = \dot{m}_a$ (2)

Water: $\omega_1 \dot{m}_a + \dot{m}_{w,2} = \omega_3 \dot{m}_a + \dot{m}_{w,4}$ (3)

$\omega_i = \frac{0.622}{\frac{P}{\Phi_1 P_{sat} T_i - 1}}$; i = 0, 1 (4)

1.4 Energy balance

Energy balance for the MSD

An energy balance can be written for the entire system, by equating input and output terms:

$$\dot{m}_a h_1 + \dot{m}_p h_{p,2} + \dot{m}_{w,2} h_{w,2} = \dot{m}_a h_3 + \dot{m}_p h_{p,4} + \dot{m}_{w,4} h_{w,4} + \dot{Q} \quad (5)$$

where

$$h_1 = h_{a,1} + \omega_1 h_{v,1} = h_{a,1} + \omega_1 h_{g,1} \quad (6)$$

$$h_3 = h_{a,3} + \omega_1 h_{v,3} \quad (7)$$

The values of h_1 and h_2 can be obtained from a psychrometric chart. The heat loss rate from the chamber can be expressed as

$$\dot{Q} = \dot{m}_a q_1 \quad (8)$$

where q_1 is insolation on the plate, [kw/m^2].

1.5 Exergy balance

Exergy balance for the MSD ω_3

An exergy balance for the entire system can be written analogously to the energy balance as follows:

$$\dot{m}_a ex_1 + \dot{m}_p ex_{p,2} + \dot{m}_{w,2} ex_2 = \dot{m}_a ex_3 + \dot{m}_p ex_{p,4} + \dot{m}_{w,4} ex_{w,4} + \dot{E} x_q + \dot{E} x_d \quad (9)$$

The specific exergy for the flow, [kJ/kg] at point 1, (AH) can be expressed as

$$ex_1 = [C_{p,a} + \omega_1 C_{p,v}](T_1 - T_0) - T_0\{[C_{p,a} + \omega_1 C_{p,v}] \ln\left(\frac{T_1}{T_0}\right) - (R_a + \omega_1 R_v) \ln\left(\frac{P_1}{P_0}\right)\}$$
$$+ T_0\{(R_a + \omega_1 R_v) \ln\left(\frac{1+1.6078\,\omega_0}{1+1.6078\,\omega_1}\right) + 1.6078\,\omega_1 R_a \ln\left(\frac{\omega_1}{\omega_0}\right)\} \quad (10)$$

and the specific exergy, [kJ/kg] at point 3, (DC) as

$$ex_3 = [C_{p,a} + \omega_3 C_{p,v}](T_3 - T_0) - T_0\{[C_{p,a} + \omega_3 C_{p,v}] \ln\left(\frac{T_3}{T_0}\right) - (R_a + \omega_1 R_v) \ln\left(\frac{P_3}{P_0}\right)\} +$$
$$T_0\{(R_a + \omega_3 R_v) \ln\left(\frac{1+1.6078\,\omega_0}{1+1.6078\,\omega_3}\right) + 1.6078\,\omega_3 Raln\left(\frac{\omega_3}{\omega_0}\right)\} \quad (11)$$

the specific exergy, [kJ/kg] for the moist product can be written as

$$ex_p = [h_p(T, P) - h_p(T_0, P_0) - T_0[s_p(T, P) - s_p(T_0, P_0)] \quad (12)$$

and the specific exergy, [kJ/kg] for the water content as

$$ex_w = [h_f(T) - h_g(T_0)] + v_f[P - P_g(T)] - T_0[s_f(T) - s_g(T_0)] + T_0 R_v \ln\left[\frac{P_g(T_0)}{X_{v,0} P_0}\right] \quad (13)$$

the exergy flow rate, [kJ/kg] due to heat loss can be, expressed as follows:

$$Ex_q = \dot{m}_a ex_q = \dot{m}_a\left(1 - \frac{T_0}{T_{av}}\right) q_1 = \left(1 - \frac{T_0}{T_{av}}\right) Q \quad (14)$$

where T_{av} is the average outer surface temperature of the dryer.

1.6 Exergy efficiency

We define the exergy efficiency for the drying process as the ratio of exergy use (investment) in the drying of the product to the exergy of the drying air supplied to the system. That is,

$$\psi = \frac{\text{Exergy input for evaporation of moisture in product}}{\text{Exergy of drying air supplied}}$$

$$\psi = \frac{(\dot{m}_w)ev[(ex_w)_3 - (ex_w)_2]}{\dot{m}_a ex_1} \text{ or } \frac{(ex_w)_3}{\omega_{3(}ex_w)_2 + (ex_w)_3/\dot{m}_a\left(1 - \frac{T_0}{T_{av}}\right)} \quad (15)$$

where

$$(\dot{m}_w)_{ev} = (\dot{m}_w)_2 - (\dot{m}_w)_4 \quad (16)$$

$$(ex_w)_3 = [h(T_3, P_{v3}) - h_g(T_0)] - T_0 [s(T_3, P_{v3}) - s_g(T_0)] + T_0 R_v \ln \left[\frac{P_g(T_0)}{X_{v,0} P_0}\right] \quad (17)$$

$$(P_{v3} = x_v)_3 P_3 \quad (18)$$

1.7 Energy Equations

The steady state energy balance equations for the AH, DC and MSD as follow:

1.7.1 Energy equation in the AH

The steady state energy balance in the AH is given as

$$A_C q_1 = \dot{Q}_{AH} = \dot{m}_a \Delta h_{a,1} + \dot{m}_{v,1} \Delta h_{v,1} \quad (19)$$

where heat loss in the AH, $\dot{Q}_{AH}$ [KW] is given as

$$\dot{Q}_{AH} = A_C q_1 - \dot{m}_a (\Delta h_{a,1} + \omega_1 \Delta h_{v,1}) \quad (20)$$

$$\Delta h_{a,1} = C_{p,a}(T_{a,1} - T_{a,0}); \ \Delta h_{v,1} = C_{p,v}(T_{v,1} - T_{v,0});$$

where $\dot{m}_a$ is the air flow rate [kg/s]; according to Duffie and Beckman (1974) is given as

$$\dot{m}_a \approx A_C U_L F' / C_{p,a} \quad (21)$$

Where q_1 is the insolation on the flat plate, [kw/m^2], which was measured directly with a pyrometer.

There are various sources of heat loss from the DH collectors as observed by (Kreith and Black, 1980). Hence; heat loss coefficient of the collector due to wind,

$$h_{wind} = 5.7 + 3.8v \quad (22)$$

where, v is the average wind velocity.

The radiative heat loss coefficient, $h_r = 4\sigma_g T_g^3$ (23)

(Duffie and Beckman, 1974), where g is the emissivity of glass, σ is Stefan-Boltzmann constant and T_g is the temperature of glass above ambient.

The overall heat loss coefficient, $U_L = h_r + h_{wind}$ (24)

1.7.2 Energy equations in the DC

The steady state energy balance in the DC yields

$$\dot{m}_a \Delta h_{a,1} + \dot{m}_{v,1} \Delta h_{v,1} + (\dot{m}_{v,3} - \dot{m}_{v,1}) \Delta h_w = \dot{m}_a \Delta h_{a,3} + \dot{m}_{v,3} \Delta h_{v,3} + \dot{Q}_{DC}$$

or

$$\Delta h_{a,1} + \omega_1 \Delta h_{v,1} + (\omega_3 - \omega_1) \Delta h_w = \Delta h_{a,3} + \omega_3 \Delta h_{v,3} + \dot{Q}_{DC} / \dot{m}_a \quad (25)$$

Where heat loss in the drying chamber, + $\dot{Q}_{DC}$ [kw] is given as

$$\dot{Q}_{DC} = \dot{m}_a [(\Delta h_{a,1} - \Delta h_{a,3}) + \omega_1 (\Delta h_{v,1} - \Delta h_w) + \omega_3 (\Delta h_w - \Delta h_{v,3})]; \quad (26)$$

$$\Delta h_{a,1} - \Delta h_{a,3} = C_{p,a}(T_{a,1} - T_{a,3}); \ \Delta h_{v,1} - \Delta h_w = h_g(T_{a,1}) - h_f(T_w);$$

$$\Delta h_w - \Delta h_{v,3} = h_f(T_w); - h_g(T_{a,3}); \quad (27)$$

and

$$\omega_3 = \frac{C_{p,a}(T_{a,1} - T_{a,3}) + \omega_1 [h_g(T_{a,1}) - h_f(T_w)]}{h_g(T_{a,3}) - h_f(T_w)} \quad (28)$$

1.7.3 Overall energy balance for the dryer

The overall steady state energy balance is given as

$$(\dot{m}_{v,3} - \dot{m}_{v,0}) \Delta h_w + A_C q_1 = \dot{m}_a \Delta h_{a,3} + \dot{m}_{v,3} \Delta h_{v,3} + \dot{Q}_{AH} + \dot{Q}_{DC}$$

Or

$$(\omega_3 - \omega_0)\,\Delta h_w + A_C q_1/\dot{m}_a = \Delta h_{a,3} + \omega_3 \Delta h_{v,3} + (\dot{Q}_{AH} + \dot{Q}_{DC})/\dot{m}_a \tag{29}$$

Where overall heat loss in the dryer, $\dot{Q}_{MSD}$ [KJ/s] is given as

$$\dot{Q}_{MSD} = \dot{Q}_{AH} + \dot{Q}_{DC} = A_C q_1 + \dot{m}_a[(\omega_3 - \omega_0)\,\Delta h_w - (\Delta h_{a,3} + \omega_3 \Delta h_{v,3})]; \tag{30}$$

$$\Delta h_{a,3} = C_{p,a}(T_{a,3} - T_{a,0});\ \Delta h_w = h_f(T_w);\ \Delta h_{v,3} = h_g(T_{a,3}) \tag{31}$$

1.8 The overall energy (η_{MSD}) and exergy (Ψ_{MSD}) efficiencies of the MSD

1 $$\eta_{MSD} = \frac{\Delta ha,3 + \omega_3 \Delta hv,3}{(\omega_3 - \omega_0)\Delta hw + A_C q_1/\dot{m}_a} \tag{32}$$

2 $$\Psi_{MSD} = \frac{(\dot{m}_w)_{ev}[(ex_w)_3 - (ex_w)_2]}{\dot{m}_a ex_1} \tag{33}$$

2 .0 Measurement and Experimentation

2.1 Solid material

The waste food used in the present work was obtained from local domestic kitchens or restaurants and was manually separated from any non-organic materials within it, such as glass, paper, plastic, foil, etc. The waste was then well mixed and ground using local pestle and mortar, and this was done to ensure the highest possible homogeneity of the constituents. The waste at this stage was in the state of a paste, and to determine the moisture content of the waste, a weighed sample was heated gently in a crucible and re-weighed until a constant weight was obtained. By simple ratio the weight of water in the sample to be dried was calculated and recorded. The final stage in preparation of the food involved manually forming the paste into spheres of approximately 6.0 cm diameter and putting them on the net- tray, which was then inserted into the dryer. This was done in the morning before the scheduled time of drying.

2.2 Equipment

The MSD consists of an AH attached to a fixed drying chamber. The air heater is similar to a solar collector of 2.0 m length and 1.0 m width with an air channel depth of 0.02 m or 0.04 [m^3] in capacity. It is made of galvanized steel sheets of 0.7 mm thick and pre-heats the air before entering the DC. The bottom of the air heater is insulated using polystyrene sheets of 10 mm. The drying chamber is designed to accommodate the tray measuring 1.0 m x 1.0 m holding the products and an inlet and outlet for the air passing through the product to be dried. It is made out of galvanized steel sheet metal of 0.7 mm thick with a height of 0.44 m or 0.44 [m^3] in capacity. The chamber is insulated with 50 mm polystyrene at the bottom and 30 mm at the sides.

The air heater was fitted with thermocouples and relative humidity meter at its inlet and exit to the chamber to measure the drying air properties. The drying chamber was fitted with thermocouples to measure the solid temperature. The amount of air leaving has to be measured to give an indication of the actual amount of water lost to the air, and the heat loss from the surface of the collector. The whole setup was directed due south at an inclination of 23^0, which was found to give the maximum energy gain throughout the day, in Makurdi, Nigeria at the month of testing, March 2011.

The solar radiation was measured directly using an SL200 solarimeter, which was controlled to give readings at one-hour intervals. The solarimeter was placed at an angle of 23^0, which is the same angle as the dryer.

Having fixed the measuring instruments at the various points of the interaction, the dryer was left onto the open sky void of any obstructing shadows.

2.3 Experimentation

5.0 kg of the paste was placed on the net tray and put in the drying chamber. A sample was weighed at the beginning and placed in the chamber which was withdrawn at every 60 minutes interval and its new weight recorded for 11 hours. The air temperature and relative humidity were recorded in the collector unit; the temperature of the solid in the drying chamber was also recorded hourly throughout the duration of drying. The moisture content of the solid was determined by Standard Test Method (ASTMD2216).

However, samples from both the dryer and control were weighed at an hourly interval for comparison purpose. The weighing was done fast in order to avoid error due to moisture loss during the process.

The purpose of this experiment was to determine the dryer overall thermal (η_{MSD}), exergy (ψ_{MSD}) efficiencies and the suitability of the dried products as animal feed.

3.0 Results and Discussion

The input data were made up of the thermodynamic properties of air and water vapor obtained from steam tables fitted in tables 1. The mean results obtained from section 3 and psychrometric chart are shown in table 2.

The results in Figures 2 to 4 show that the energy (thermodynamic) efficiency was higher than the exergy efficiency because the energy loss was lower than the intrinsic irreversibility (exergy destruction) in the AH, CD and MSD, respectively. Comparing the magnitude of energy and exergy efficiencies in Figures 2 and 3, energy efficiency in DC was more than that in the AH, this indicates that more energy loss occurred in the AH than in the DC. Similarly, exergy efficiency in the DC was more than that in the AH. The drop in the energy and exergy efficiencies in the MSD, that is the overall energy and exergy efficiencies of the system was due to the energy loss and exergy destruction in the AH.

Figures 2 to 5 show that between the periods: 12:00 pm to 14:00 pm, the energy efficiency was at its peak value with corresponding lowest values of the specific humidity (humidity ratios) since the maximum insolation was produced within these periods. The non smooth profile nature of the figures 2 to 5 is attributed to the fluctuating atmospheric conditions (T, Φ, ω).

In the AH, the mean inlet and outlet: air temperatures were 303.9 and 321 [K] respectively, humidity ratio was 0.022, air mass flow rate was 0.0482 [kg/s], the mean values of energy efficiency, exergy efficiency and exergy destruction were 0.43 [%], 0.18 [%] and 6.77 [kg/kg] respectively.

In the DC, the mean inlet and outlet: air temperatures were 321 and 315.5 [K] respectively, humidity ratio was 0.0204 and 0.0227 respectively, and air mass flow rate was 0.0432 [kg/s], the mean values of energy efficiency, exergy efficiency and exergy destruction were 0.54 [%], 0.35 [%] and 1.78 [kg/kg] respectively.

Table 1 Input Constant

S/№	Symbol	Units	Value
1	g	m/s^2	9.81
2	A_C	m^2	2.0
3	б	W/m^2K^4	5.6 x 10^{-8}
4	P_0	kPa	101.3
5	w	m	1.0
6	m_s	kg	5.0
7	$C_{p,a}$	kJ/kg^0C	1.004
8	$C_{p,v}$	kJ/kg^0C	1.872
9	R_a	kJ/kg^0C	0.287
10	R_v	kJ/kg^0C	0.4615

Table 2 Mean Input Data

				Temperature						Relative Humidity			Humidity Ratio			
Time	Insolation	Power on Plate	Solid Moisture	Point (0)	Point (1)	Point (3)	Solid	Cover	Plate	Point (0)	Point (1)	Point (3)	Point (1)	Point (3)	Wind speed	Mass flow rate
t	I	P	X	T_0	T_1	T_3	T_s	T_c	T_p	Φ_0	Φ_1	Φ_3	ω_1	ω_3	S_{wind}	$\dot{m}_a$
[hr]	[w/m^2]	[kW]	[kg/kg]	[K]	[K]	[K]	[K]	[K]	[K]	[-]	[-]	[-]	[-]	[-]	[m/s]	[kg/s]
8:00	74	0.148	0.378	303	309	307	311	313	319	0.33	0.25	0.31	0.022	0.022	1.00	0.0482
9:00	180	0.360	0.368	305	311	309	311	315	320	0.31	0.24	0.30	0.022	0.023	1.00	0.0482
10:00	290	0.580	0.355	307	317	313	317	319	327	0.30	0.23	0.28	0.022	0.024	1.00	0.0482
11:00	360	0.720	0.332	305	319	315	321	327	331	0.29	0.22	0.26	0.021	0.023	0.99	0.0477
12:00	460	0.920	0.308	303	329	319	333	337	339	0.29	0.22	0.26	0.020	0.025	0.89	0.0429
13:00	450	0.900	0.273	305	333	323	337	339	341	0.28	0.20	0.25	0.019	0.023	0.99	0.0477
14:00	350	0.700	0.245	305	331	321	333	339	343	0.28	0.20	0.25	0.020	0.024	1.00	0.0482
15:00	300	0.600	0.222	303	327	321	329	331	339	0.29	0.21	0.26	0.019	0.022	1.00	0.0482
16:00	250	0.500	0.200	303	325	317	327	329	337	0.30	0.22	0.28	0.020	0.023	0.99	0.0477
17:00	93	0.186	0.192	303	317	315	319	317	329	0.31	0.23	0.28	0.019	0.020	0.99	0.0477
18:00	70	0.140	0.181	301	313	311	315	317	323	0.32	0.24	0.30	0.020	0.021	1.00	0.0482

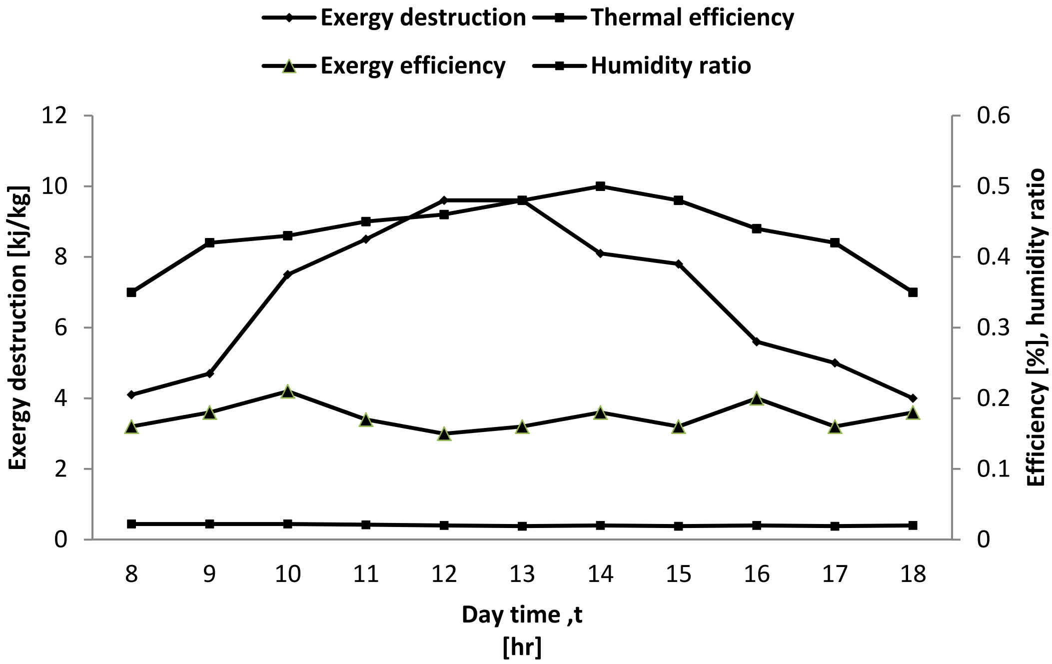

Figure 2 Curves of Efficiency, Exergy Destruction and Humidity Ratio in AH

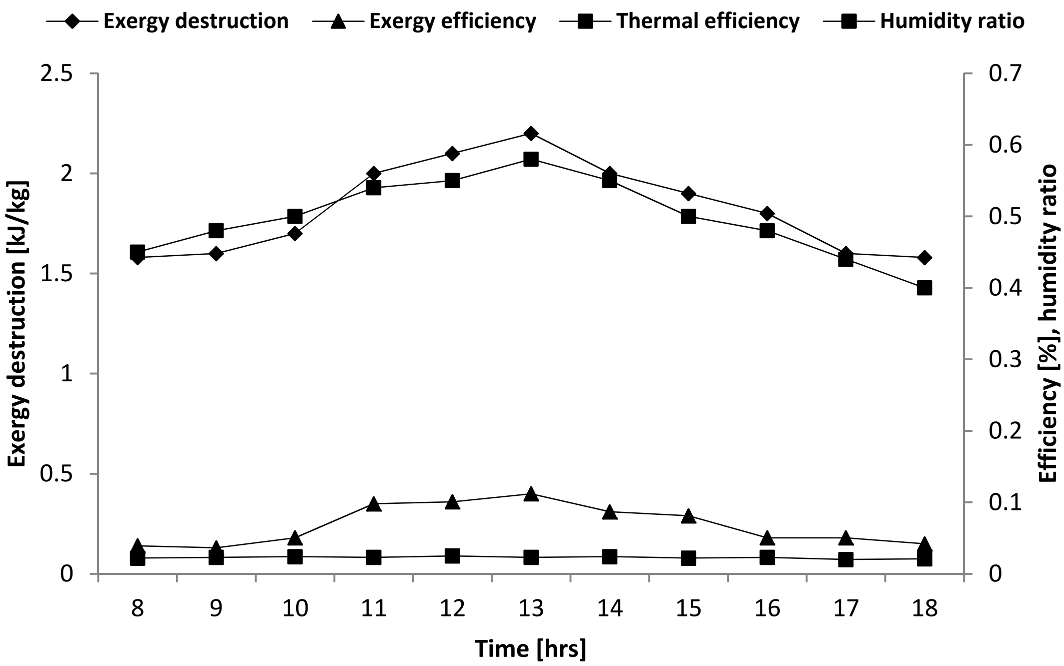

Figure 3 Curves of Efficiency, Exergy Destruction and Humidity Ratio in DC

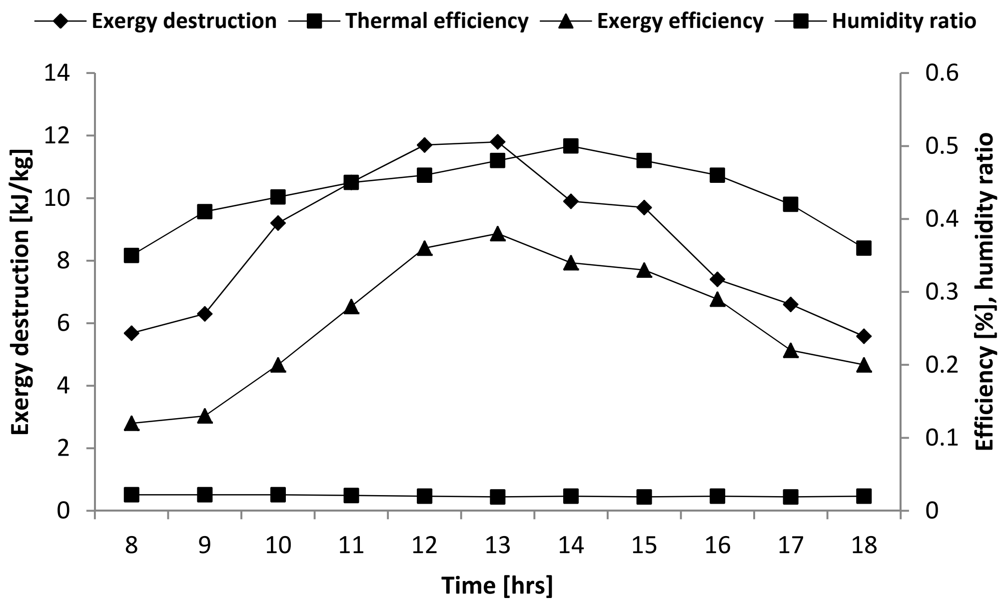

Figure 4 Curves of Overall Efficiency, Exergy Destruction and Humidity Ratio in MSD

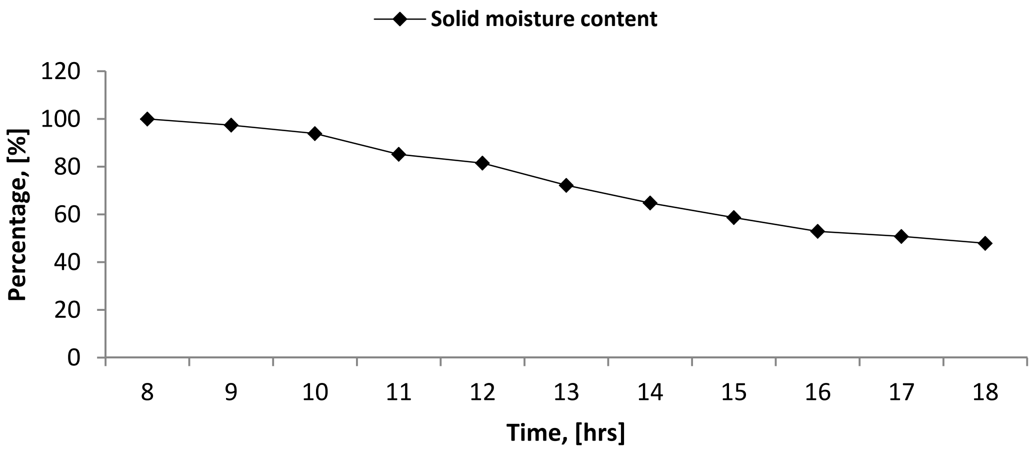

Figure 5 Curve of overall solid moisture content in MSD

The overall mean values of energy efficiency, exergy efficiency and exergy destruction were 0.44 [%], 0.34 [%] and 8.55 [kg/kg] respectively. From these results, minimizing exergy destruction in the entire system will appreciate the exergy efficiency by lowering the temperature of the drying air or alternatively lowering the air flow rate thereby increasing the power available for the drying. Proper lagging (applying critical insulation thickness) will minimize heat loss from the system will also increase the system efficiency.

Figure 5 shows the overall rate of solid moisture content loss from the system throughout the duration of drying. At the end of the experiment, 52.1% of the moisture was lost from the solid waste which was tested by the department of animal nutrition within the University for Suitability as animal feeds and was given a pass mark.

4.0 Conclusions

Thermodynamic analysis of a mixed-mode solar dryer (MSD) has been carried out. The energy and exergy efficiencies in the air heater (AH) were in comparison lower than in the drying chamber (DC) since the heat loss in the AH was more than in the DC. The overall analysis showed that energy and exergy efficiencies in the MSD was slightly higher than those in the AH but slightly lower than those in the DC due to the cumulative effect of energy losses and exergy destruction in the AH and the length of time that was given to the CD utilize the stored energy in the solid to expel moisture. The solid waste was tested and fit for animal consumption. Thus the MSD can be used to dry restaurant wastes for animal consumption.

References

[1] Akhtar, N. and S. C., Mulluick, (1999). Method for Computation of Glass Cover Temperature and Top Heat –Loss Coefficient of solar Collectors with Single Glazing, Solar Energy Journal Vol. 66, No. 5, pp. 349-354.

[2] Anderson, E.E.(1983) Fundamentals of solar energy conversion, Addison Wiley publishing company reading, pp 85, 140.

[3] Arata A. and V.K. Sharma, (1991) Performance evaluation of solar assisted dryers for low temperature drying applications. I. Plants description, Renewable Energy 1 (5/6) pp. 729-735

[4] Ayensu, A. (1997) Dehydration of food crops using a solar dryer with convective heat flow, Solar energy 59 pp. 121–126

[5] Bansal, N. K., (1999). Solar Air Heater Applications in India. Renewable Energy Vol. 16 pp.618-623

[6] Bhargara, A. K, H. P., Garg, (1982). Evaluation of the Performances of Air Heater of Conventional Design, Solar Energy Vol. 41, No. 1, pp. 101-107.

[7] Biond, P. and L. Cicala, (1988). Performance Analysis of Solar Air Heater of Conventional Design, Solar Energy Vol. 41, No. 1 pp. 101-10.

[8] Coskun, C., Bayraktar, M., Oktay, Z. and Dincer, I. (2009) 'Energy and exergy analyses of an Industrial wood chips drying processes', Int. J. Low-carbon Tech., Vol. 4, pp.224-229

[9] Duffie, J.A. and Beckman, W.A. (1974) Solar Energy Thermal Processes, John Wiley and Sons. Inc.

[10] Hou, S., Zang, D., Ye, S. and Zhang, H. (2007) 'Energy analysis of the solar multi-effect Humidification – dehumidification desalination processes, Desalination (The Int. J. Scs. & Tech. of Desalting and water purification), Vol. 203, №s 1-3, pp.403-409.

[11] Ibrahim, D. and Mark, A.R, (2007) 'Energy, Environment and Sustainable Development' pp 120-126 Kotas, T.J. (1995) Energy Method of Thermal Plant Analysis, Krieger Publishing, Malabar

[12] Oko, C.O.C., Nnanchi, S.N. (2012). 'Energy and exergy audit of a solar grain dryer', Int. J. Renewable Energy Technology, Vol.3, № 1, pp. 79-93

[13] Ozegerner, L. and Ozegerner, O. (2006) 'Energy analysis of industrial pasta drying processes', Int. J. of Energy Research, Vol. 30, № 15, pp 1323-1335.

[14] Riviere, A., Khalil, B.A. and Berthou, M. (2009) 'An energy efficiency assessment, tool base on Process energy and exergy analysis method', paper presented at the proceedings of the European Council for Energy Efficient Economy (ECEEE) Summer Study, Panel 5, June9-10, Stockholm, Sweden.

Nomenclature

g	Gravitational constant, [m^2]	t	Time, [s]
A_C	Cover surface area [m^2]	T_0	Ambient temperature, [K]
A_p	Area of absorber plate [m^2]	T_a,i,i = 0, 1, 2	Humid air temperature, [K]
б	Stefann-Bolzmann constant	T_C	Cover tempereature, [K]
P	System pressure, [kN/m^2]	T_p	Plate teperature, [K]
P_{sat}	Saturation pressure, [kN/m^2]	T_s	Solid temperature, [K]
w	Width of plate, [m]	T_w	Water temperature, [K]
m_s	Mass of solid, [kg]	$Ʈ_C$	Transmittance of cover, [-]
$C_{p,a}$	Air specific capacity, [kJ/kgK]	S_g	Gas entropy, [kJ/kgK]
$C_{p,v}$	Vapor specific heat capacity, [kJ/kg]	S_f	Liquid entropy, [kJ/kgK]
R_a	Air constant, [kJ/kgK]	U_L	Overall heat transfer
R_v	Water vapor gas const., [kJ/kgK]	v_f	coefficient, [kW/m^2k]
ex, i = 1, 2	Exergy of humid air, [kJ/kg]	X	
ex_w,i = 1, 2	Exergy of water, [kJ/kg]		Liquid specific vol., [m^3/kg]
ex_p	Exergy of product, [kJ/kg]		Solid moisture content, [kg/kg]
h_v, h_g	Vapor enthalpy, [kJ/kg]	Superscript	
h_f, h_w	Liquid enthalpy, [kJ/kg]	0	Angle, [^{0}]
ha,i, i = 1, 2	Air enthalpy, [kJ/kg]	Subscript	
ω,i, i =1,2	Humidity ratio	a	Air
I_{sc}	Insolation constant	p	Collector plate or
$\dot{m}_a$	Air flowrate, [kg/s]		isobaric condition
		w	Water
		i = 0,1,2	Air states