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Drying chamber performance of V-groove forced convective solar dryer

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Abstract

The performance of the drying chamber of a solar assisted drying system was analyzed. The solar assisted drying system consists of drying chamber, V-groove collector of 13.8 mð area, the auxiliary heater and two variable speed centrifugal fans. K-type thermocouples were used for temperature measurement, while solar radiation was measured by Eppley Pyranometer. Drying temperature is considered the most important factor in the drying operation. The accuracy of the temperature measurement is important in this analysis. The procedure for calculating the heat losses from drying chamber may be quite complex and often difficult to accurately determine the numerous variables. In this work standard equations under steady state condition were used to calculate the drying chamber efficiency and the heat losses from the chamber room.

Keywords: Temperature; Drying chamber; Performance; Heat transfer coefficient

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1. Objective

The main purpose of this paper is to report an analytical study of the energy performance of the drying chamber. Although a numerical analysis can be take into account more details related to the energy balance. The analytical study will gives a better feeling of the influence of the operational and design parameters, such as ambient temperature, drying chamber temperature, ambient humidity, solar radiation and airflow.

2. Introduction

Solar energy can be used as an important and environmental compatible source of renewable energy. The use of solar energy for drying effectively reduces the problems arising from generating energy by convention method. This is because the use of the conventional energy source for drying purposes is costly and hazardous to environment. The maximum temperature of 60°C was recorded during an operation without heater system at min daily solar radiation of 700–800 $\mathrm{Wm^{-2}}$ and at 6–17 $\mathrm{m^3\ min^{-1}}$ of air flow

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rate. With the electric heater off the dryer generates 4.0–6.5 KW of heat depending on airflow rate and solar radiation intensity. For constant temperature drying, both the solar collector and electric heater were simultaneously used. Drying chamber is rated in terms of thermal efficiency as a function of specific insulation and mass flow of the heat transfer.

In this project temperature measurements were taken by means of thermocouples fixed at different points in the drying chamber. The average temperature outside the drying chamber ranged from 28–34°C while the drying chamber temperature maintained at 55°C in our experiments using the auxiliary heater. The radiation measurement arranged between 400–800 W/m^2.

3. System description

The design and fabrication of the solar dryer has been reported elsewhere (Ruslan et al. 2001). The components of the drying system consist of solar collector, blower, drying chamber and electrical heater. An array of V-groove back-pass solar collector with a total area of 13.8 m^2 was used to supply the heat. The blower with variable speed was used to circulate the air at the range of 6–17 m^2 min^{-1} to the drying chambe. The drying chamber measuring 1.0 m × 2.5 m × 2.90 m encloses adjustable shelves to support wire-mesh trays where the product to be dried were placed. A 10 KW electric heater was installed at the inlet of the drying chamber in order to maintain constant and higher temperature in the chamber. The dryer can operate with or without the heater system. The configuration of the system is shown in Fig. 1.

Drying chamber is shown in Fig. 2. The volume of drying chamber was 7.25 m^3.

The wall of the drying chamber was build with a double wall with the thickness of 10 cm each and used Aluminum sheet as the cover for internal surface of walls for purpose to reduce the heat loss. In the middle of the two walls have a 2.5 cm space, which filled up by the foamed polystyrene. The hollow iron rectangular door of the chamber room is designed with two open types each with dimension 1.5 m × 1.5 m. The hollow space at the door is covered with fiberglass acts as insulation.

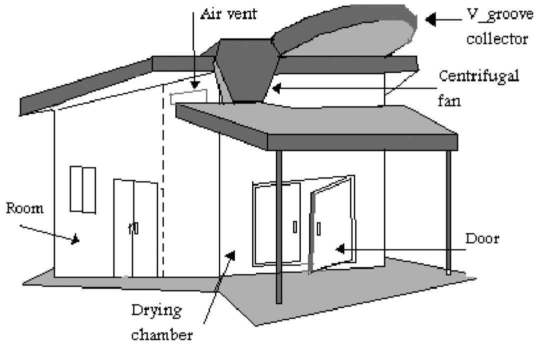

Fig. 1. The design of colar assisted drying system.

The air enters the drying chamber at the bottom through two tubes PVC with diameter 15.2 cm each and then passed through the auxiliary electric heater. An exhaust vents 20 cm × 50 cm each have also been provided for the saturated air exit to environment.

3.1. *Model for the drying chamber analytical thermal*

The performance of solar drying system depends on many parameters of which the most

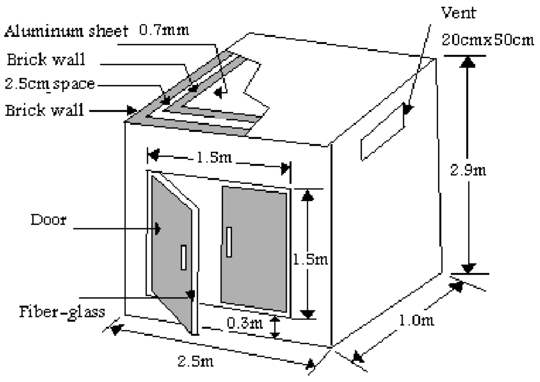

Fig. 2. Construction design of drying chamber.

important are local climatic conditions, solar radiation, building type and size. Thermophysical properties of materials used, hot air requirement capacity of auxiliary heating system, etc. (H.P. Garg and J. Prakash).

Heat loss from the drying chamber takes place by conduction, convection and radiation. The heat loss is directly proportional to the temperature difference between the indoor and outside ambient temperature. Analytical expressions used for the computation of various heat losses are given below.

The overall heat transfer coefficient U is the reciprocal of the total resistance offered by a wall or roof structure.

$$U = 1/RT \tag{1}$$

Since the wall does not consist of single material but of a layers of different materials with different thickness. The total thermal resistance RT is given as:

$$RT = 1/\mathrm{x}0 + 1/\mathrm{x}1 + R1 + R2 + R3 + R4 + \cdots \tag{2}$$

where x0 and x1 are the thermal conductance for outside and inside surfaces to air, and $R1$, $R2$, $R3$ are the thermal resistance of various layers. For single layer the resistance $R1$ is given as:

$$R1 = d1/k1 \tag{3}$$

where $d1$ is the thickness of single layer and $k1$ is the thermal conductivity of the material in the layer, Table 1 shows k for various materials.

Since the chamber room temperature is higher than ambient temperature a flow of heat from the hot side to the cold side through the wall and roof will occur.

The flow of heat is defined as:

$$q = k/d\ A(T_2 - T_1) = kA/d\ \Delta T \tag{4}$$

where

q is the rate of heat flow in Watt

d is the thickness of the material in m.

Table 1

Components	Thickness (m)	Conductivity (W/m°C)	Resistance (m^2°C/W)
Aluminum	0.7 mm	205	0.0034
Plaster	0.025	1.40	0.017
Brick	0.1	1.4	0.07
Polystyrene	0.025	0.83	0.08
Concrete	0.1	0.8	0.12
Iron	0.045	29.3	0.0015
Total Resistance			0.280

A is the area of the wall surface in m^2. ΔT is the temperature difference, causing heat flow, °C

Fig. 3 shows the structure of the drying chamber walls, it illustrates a several materials or non-homogeneous material.

The flow of heat for this case is defined as:

$$q = UA\ (T_0 - T_5\ ) = UA\Delta T\ \mathrm{W/m^2{\bullet}C} \tag{5}$$

where U is referred to as the conductance or coefficient of transmission of the material W/m^2·C.

The flow of heat for the composite material can also be specified in terms of the conductivity of the material and the conductance of the outside surface.

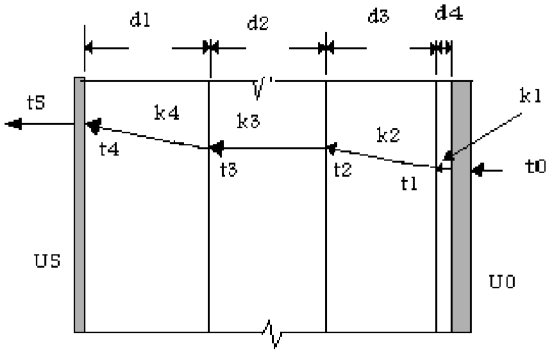

Fig. 3. Temperature distribution for the composite wall.

$$q = \frac{A(T_0 - T_5)}{1/U_0 + d_1/K_1 + d_2/K_2 + d_3/K_3 + d_4/K_4 + 1/U_5} \quad (6)$$

The conductance of a homogeneous material is expressed of:

$U = k/d$

where $R = 1/U = d/k$.

Air change ventilation heat loss

$Pa\ VCp\ \Delta T$

where pa = density of air (kg/m^3).

V = volumetric exchange rate (m^3/hr).

Cp = specific heat of air (kJ/kg °C).

Table 1 shows the thermal conductivities, resistances and thicknesses of the material, which used for the construction of the drying chamber.

4. Results and discussion

So far attention has been given to the calculation of the overall thermal transmittance or U-value of drying chamber components. The calculation of overall heat transfer coefficients are illustrated in Table 2.

A separate calculation is made for each different surface in drying chamber. The outside surface film coefficient used for calculations is considered to be 0.03 m^3. C/W. While the inside surface film coefficient was 0.08 m^3·C/W. For the exhaust-saturated air from the vents, the velocity of exit air is 1.4 m/s. The total area of the walls of the chamber room is 18 m^2 Summation is conveniently made by chamber room and for the complete structure. The overall heat transfer coefficient U is 682 W/m^2·C. Curve 1 shows the variation of three parameters, which the air temperature of the drying chamber, the inner surface of the wall and the ambient temperature with the time. The result shows that the temperature of the wall surface is less than the drying chamber temperature has been expected early.

Table 2

	Area (m)	U-value (W/m^2·C)	ΔT (°C)	Q(W)
Walls	18	3.57	23	1477.9
Roof	2.5	4.29	23	249.6
Floor	2.5	8.3	23	477.2
Door	2.25	666	23	34,500
Vents	0.2	23	3.65	
Total		682		36,705

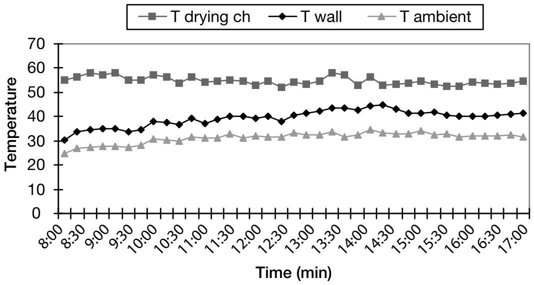

Fig. 4. Comparison of drying chamber air temperature variation in an 25 cm two layers brick wall with ambient temperature.

5. Conclusion

The study was conducted to evaluate the performance of the drying chamber of the forced convective solar assisted drying system. The consideration should be given to the materials that used for the structure of the drying chamber. The efficiency of the system is increased if the drying chamber is well constructed and provided with good insulation. In this paper the equations for determining the changes of temperature in drying chamber are based on heat transfer analysis. These expressions are founded on the method of the thermal effectiveness and the number of transfer units. These are general equations for the drying chamber of the solar drying system.

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References

[1] Albert Thumann, 8th ed., Plant Engineers and Managers Guide to Energy conservation, Indian Trail, 2001.

[2] A.F. Mills, Basic heat and mass transfer, 1995.

[3] A. Ali. Sehery, M.G. Rahya and H. Shoshan, Preservation of fruit and vegetables using solar dryers, 1999.

[4] C.G. Gronqvist, Material science for solar energy conversion syste, 1991.

[5] C. Faye and D. McQuiston and Jeral, Parker. 2nd ed. Heating ventilating and air conditioning, USA 1982.

[6] D. Chisholm, Heat exchanger technology, UK 1988.

[7] H.P. Gary and J Prakash, Solar Energy fundamentals and applications, 2002, New Delhi.

[8] J.F. Van Straaten, Thermal performance of buildings.

[9] A. John, A. Duffie and Williama Beckman, 2nd ed., Sol. engineering of thermal processes, New York, 1991.

[10] John Wiley & Sons, An introduction to solar energy for scientists and engineers (sol wieder), New York, 1982.

[11] A. John, duffic and A. William, Beckman. Solar energy thermal processes, United States, New York, 1974.

[12] C.S. Moraitis and C.B. Akritidis, Energy saving in industrial plants by partial recovery of the latent heat of the exhaust air, Drying technology, 15 (6–8) (1997) 1931–1940.

[13] K.S. Gan and K.T. Choo, Thermal performance of a solar dryer-simulation case study, (2000) 134–152.

[14] H.P. Gary and Rakesh Kumar, Developments in solar drying, 2001.

[15] M.H. Ruslan, M. Othman, B. Yatim, K. Sopian and Z. Ibarahim, Performance of V-groove force convective solar drying system, ADC 2002.

[16] M.H. Ruslan, M. Othman, B. Yatim and K. Sopian, Design and optimization of V-groove solar collector, ADC (2001).

[17] Miguel Condorf and luis saravia, The performance of forced convection greenhouse dryers, Renew. Energy, 13(4) (1998) 453–469.

[18] M.Y. Othman, M.H. Ruslan, K. Sopian, B. Yatim and W.R.W. Daud, Solar assisted drying system for cilli, ADC (2001).

[19] P. Axaopoulos, P. Aanagakis, A. Tsawdaris and D. Georgakakis, Simulation and Experimental performance of a solar heated anaerobic digester, 70(2) (2001) 155–164.