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International Journal of Recent Scientific Research
Vol. 4, Issue, 8, pp.1175- 1180, August, 2013

ISSN: 0976-3031

International Journal of Recent Scientific Research

RESEARCH ARTICLE

EXPERIMENTAL ANALYSIS OF RECIRCULATION TYPE SOLAR AIR DRYER

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ARTICLE INFO

Article History:

Received 15th, July, 2013
Received in revised form 27th, July, 2013
Accepted 12th, August, 2013
Published online 30th August, 2013

Key words:
air drier, solar collector, agricultural products, corrugated absorber plate.

ABSTRACT

Drying is simply the process of moisture removal from a product. It can be performed by various methods. In these methods, thermal drying is most commonly used for drying agricultural products. Solar heated air could be used more effectively for drying of the various products under controlled conditions. But still today, dryers appear the most attractive option for use in rural locations. In industry solar drying system is rarely used. Though solar drying system having versatile appearance but still it has some disadvantage, which restrict its use in industry. The use of solar heat into industrial production processes (SHIP) is a challenge to both: the process engineer and the solar expert. The major share of energy which is needed in industrial production processes is below 250°C temperature level, which could be well supplied by solar thermal technologies. The lower temperature level (< 80°C) can already be provided today with commercially available solar thermal collectors. Author is claiming three different types of collectors for air dryer, which will provide heated air up to 95°C. Core of the paper is discussing the thermal analysis, so that it can be easy to get dimensions for the indirect type passive solar air dryer using simple non selective type absorber plate.

INTRODUCTION

In India there is abundance potential for solar energy is available. Solar air heating can be an appropriate energy solution in the right application. Solar air dryers are used for drying agricultural and industrial products, and for space heating. This method is cheap because the source of energy is free and renewable. However, this drying technique often entails contamination, insect infestation, microbial attack, nutritional deterioration and other problems which were discussed by A. Madhlopa, 1971. Solar heated air could be used more effectively for drying of the various products under controlled conditions. To overcome the disadvantage of direct drying, drying cabinet is attached to solar dryer.

Solar Heat for Industrial Processes (SHIP)

The use of solar heat into industrial production processes is a challenge to both: the process engineer and the solar expert. The major share of energy which is needed in industrial production processes is below 250°C temperature level, which could be well supplied by solar thermal technologies. The lower temperature level (less than 80°C) can already be provided today with commercially available solar thermal collectors. Most solar applications for industrial processes are on a relatively small scale and are mostly experimental in nature. Solar Heat for Industrial Processes (SHIP) is still in its infancy and yet, there is potentially an enormous range of solar thermal applications within this sector which can supply a large portion of our total energy demand. Because solar industrial process heat is not yet widely available, it is important to understand its specific barriers to growth and consequently the best strategies to help overcome these barriers. For many industrial applications higher temperatures are needed. The solar thermal collectors must be specifically made to provide heat at temperatures above 80-100°C.

Design of Drying System

In solar air dryers air flow may be only above or below or on both sides of absorber plate. Usually a rectangular duct is employed for air passage. Such air dryers are simple to manufacture and relatively cost effective [Gupta C.L., Garge H.P., 1967, Charters W.W.S., 1971]. Balance between the solar gain, heat loss and the use of one or more covers usually depends on the range of temperature for which collector is designed. In non porous air dryer the heat transfer between air and the absorber plate is low due to convection mode. The heat transfer coefficient can be increased by roughening the absorber surface or by changing the flow geometry by incorporating fins or corrugated absorber plate [Hikmet Esen., 2007, S.K. Verma *et al.,* 2000.] Modification in duct dimensions also helps in improving the heat transfer coefficient.For a given flow passages geometry the heat transfer coefficient ‘h’ is closely related to pressure drop ‘Δp’ across the ends of the duct at the cost of pumping power. A balance therefore needs to be obtained between h, outlet fluid temperature, and efficiency and pumping power mass flow rate per unit collector area (m/A). The flow rates usually taken for air dryers are such that the corresponding Reynolds no. may

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extend from the laminar to the early turbulent regime. All these factors are to be incorporated in the design principle of the solar air dryers. The study of the system for all three flow conditions using adequate heat transfer relations has been carried out by H.P. Garg and G. Datta, 1967.

In type I, the air flow is below the absorber plate. In type II the air flow is between the cover plate and absorber plate and in type III the air flow is on both sides of the absorber plate. Parker, 1980 and A. K. Bhargava, 1982, have extensively analyzed these non porous air dryers for evaluation of the performance. According to Parker, 1980 Type I is superior to either of the other two collectors at higher operating temperatures. However for ambient air, the type III would achieve equal efficiency. The type II is always less efficient than other two collectors. According to A.K. Bhargava and H.P. Garg, 1982, in types I & III the efficiency of solar collector is more for shorter length of absorber plate, whereas situation is reverse in type II. The comparison of three types of collectors suggests that type II collector gives better performance than other two types of collectors at higher temperatures. The above analysis shows that the type III air dryer would be more suitable for drying applications since the temperature requirement is not too large is case of drying of industrial products (Range 80^0 to 100^0 C). We have, therefore, selected type III air dryer in which the flow of air is on both the sides of the absorber. The thermal analysis to optimize the design parameters have been carried out for type III air dryer.

Thermal Analysis

The analysis presented for type III by Parker, 1980 and A. K. Bhargava, 1982, probably the best available, gives many information regarding the designing of solar air dryer. They have assumed equal air flow in two channels as the duct size is same for both channels, therefore heat transfer coefficient h between absorber plate and fluid in both channels. Pawar R.S., 1993, has formulated the heat balance equations of the solar air dryer with flow on both sides of absorber plate. These equations are then coupled with those for useful heat extracted in the two channels to arrive at a basic simultaneous coupled differential equations governing space time development of the temperature in the two channels. Since the air dryer under considerations is non porous and thermal capacitance of the air is comparatively small, the transient analysis of the model is not essential. Hence, steady state analysis has been used in the present work. Explicit expression of local temperatures in the fluid channels is obtained and these are then studied numerically for various values of the air dryer parameters involved in the operation. The drying cabinet is as shown in Fig 1 having length l and width w. The product to be dried is loaded in this chamber having layer of height h. If X_1 and X_2 are initial and final moisture content of the product, the amount of water to be evaporated from the product is given by,

$$M_v = \frac{X_1 - X_2}{100 - X_2} M \qquad \text{........ (1)}$$

and

$$M = \rho.V = \rho.l.w.h \qquad \text{........ (2)}$$

The rate at which moisture should be removed from the product depends upon the safe storage time and the drying capacity of the ambient air. If the y is the drying capacity of ambient air, then to remove M_V of water from the product, (M_V/y) kg of air must be passed through the product. Taking in to consideration d safe storage time, the rate at which air should be passed through the product is given by

$$\dot{m} = \frac{M_V}{24.y.d} \qquad \text{........ (3)}$$

$$m = v.A.3600$$

In this type product is not directly exposed to solar radiation. Due to recirculation, the temperature of the air in the chamber increases, hence the drying capacity of ambient air also increases. When such heated air passes through the product, the drying speed can be divided in to two components, such that,

$$\dot{M}_V = \dot{M}_{V1} + \dot{M}_{V2} \qquad \text{........ (4)}$$

The drying speed also can be described as [45]

$$\dot{M}_V = \dot{m}\frac{\phi_2 y_2 - \phi_1 y_1}{100} \qquad \text{........ (5)}$$

The amount of water absorbed by drying air is,

$$y = \phi_2 y_2 - \phi_1 y_1 \qquad \text{........ (6)}$$

The water content of the air at saturation is approximately linear to the temperature when temperature rise is small. The change in the water content for saturated air caused by temperature rise is given by [7]

$$\Delta y = 0.00121\,\Delta T \qquad \text{........ (7)}$$

In our case, Δ_T represents the temperature rise of air due to the absorption of solar radiation by the absorber plate. Similarly we have,

$$\Delta y = y_2 - y_1 \qquad \text{........ (8)}$$

Using equations (5) and (8) we have,

$$\dot{M}_V = \frac{\dot{m}}{100}(y_1(\phi_2 - \phi_1) + \phi_2.\Delta y) \qquad \text{........ (9)}$$

by comparing equations (4.4) and (4.9) we have

$$\dot{M}_{V1} = \frac{\dot{m}}{100} y_1(\phi_2 - \phi_1) \qquad \text{....... (10)}$$

$$\dot{M}_{V2} = \frac{\dot{m}}{100}\phi_2.\Delta y \qquad \text{....... (11)}$$

Similarly by equation (7), we have,

$$\dot{M}_V = \frac{\dot{m}}{100}[y_1(\phi_2 - \phi_1) + \phi_2.\Delta T.0.00121] \qquad \text{....... (12)}$$

The energy balance equation for the dryer can be written as

$$I.A.\tau.\alpha = \gamma.\dot{M}_{V2}.1000 + \frac{\dot{m}C_p\Delta T}{3.6} + U_T.A.\Delta T + \frac{\dot{m}\Delta p}{3600.\rho_{air}} \qquad \text{..... (13)}$$

The first term on the right side of the equation (13) is the useful energy Q_u used to increase drying capacity of the air by absorbing solar radiation. The second term of the equation (13) represents the energy loss caused by the air flowing through the product. The third term represents heat lost from the dryer through insulation and cover. The last term in the equation (13) is the energy utilized to force the air through the product. This component as compared with other terms is very small and hence can be neglected. The force to move air through the product is created by the pressure difference between the air in the pipe connected after cabinet and pipe connected before cabinet. The pressure difference ΔP is given by,

$$\Delta p = \Delta\rho_{air}.g.h \qquad \text{…….. (14)}$$

The difference in density of air is caused by the temperature difference between cabinet and the pipe.

Hence we have,

$$\Delta\rho_{air} = 0.005.\Delta T \qquad \text{…….. (15)}$$

Similarly, the pressure difference across the product layer is given by,

$$\Delta p = a.h.v \qquad \text{…….. (16)}$$

A pressure balance, from equations (14) and (16) gives

$$a.h.v = 0.005\,.g.h.\Delta T \qquad \text{…….. (17)}$$

Since,

$$v = \frac{\dot{m}}{A.3600.\rho_{air}}$$

From equation (17) we get,

$$\dot{m} = \frac{18.A.g.\rho_{air}.H}{a.h.\Delta T} \qquad \text{……... (18)}$$

Substituting value of m and neglecting last term in the equation (13) we get

$$I.A.\tau.\alpha = A\rho_{air}.\frac{H}{a.h}(2.14\gamma\phi_2 + 49.1.C_p)\Delta T^2 + U_T.A.\Delta T \qquad \text{…….. (19)}$$

Let

$$z = \frac{H}{a.h} \qquad \text{……...(20)}$$

Introducing parameter z, we have

$$\rho_{air}z(2.14\gamma\phi_2 - 49.1.C_p)\Delta T^2 + U_T.A.\Delta T - I.\tau.\alpha = 0 \qquad \text{……. (21)}$$

Equation (20) is quadratic in *ΔT*, hence temperature rise ΔT will be given by

$$\Delta T = \frac{U_T}{2\rho_{air}z}.\frac{1}{(2.14\gamma\phi_2 + 49.1.C_p)} + \frac{1}{\rho_{air}z}.\frac{1}{(2.14\gamma\phi_2 + 49.1.C_p)}.$$

$$\sqrt{\frac{U_T^{\,2}}{4} + I.\tau.\alpha\rho_{air}z(2.14\gamma\phi_2 + 49.1.C_p)} \qquad \text{…….. (22)}$$

The efficiency of the dryer can be calculated since,

$$\eta = \frac{Q_U}{I.A} \qquad \text{…….. (23)}$$

Hence,

$$\eta = \frac{10.\gamma.\dot{m}\phi_2\Delta y}{I.A} = 2.14\gamma\frac{\rho_{air}z\phi_2}{I}.\Delta T^2 \qquad \text{…….. (24)}$$

Similarly, from equations (4), (10), (11) and (18),
We have,

$$\dot{M}_V = 1.77.A.\rho_{air}z[yl(\phi_2 - \phi_1).\Delta T + 0.00121\phi_2.\Delta T^2] \qquad \text{….. (25)}$$

The most important parameters for designing the dryer are

- The depth of product layer
- The height between plate and the cabinet product
- Air flow resistance coefficient (a)

All these parameters are combined in a single parameter z where,

$$z = \frac{H}{a.h} \qquad \text{…….. (26)}$$

The above three parameters are responsible for the functioning of the natural convection solar dryer.

Parameters considered for Drier

The analysis of solar air dryer comprising of two flow channels enables us to optimize the design parameters. In air dryers of this kind the shorter plate length is more efficient than longer plate lengths. The suitable choice is between 1.5 m to 2.0 m. The optimum plate length is 1.5 m .The width of collector is selected to be 1.0 m for structural convenience. The depth ratio D_R is optimized to unity and the optimum channel depth is 0.0254 m. The heat transfer coefficient h_2 and h'_2 are reasonably good for this value of channel depth. The plate thickness of the absorber plate was selected to be 2 mm for the economic considerations and structural convenience. The small thickness of metallic plate has negligible conductivity effect. Bhargava [9] has shown that increase in plate thickness does not affect the performance of the solar collector. In our case, the plate length is short and mass flow rate is higher hence selective coating will not be useful. We have, therefore applied the Ferretol black paint as a coating on the absorber plate. Two glass covers certainly improves the performance of collector but for economical purpose, single glass cover has been used for cost effectiveness. Table.1 enumerates the parameters of solar air dryer.

Table 1 Parameters of Solar collector for air drier

No.	Parameters	Value
1.	Thickness of absorber plate (mm)	2
2.	Absorptivity of coatings of absorber plate	0.9
3.	Length of absorber plate (m)	1.5
4.	Width of absorber plate (m)	1.0
5.	Depth (m)	0.0254
6.	Depth ratio	1.0
7.	No. of glass cover	1
8.	Transmissivity of glass	0.9
9.	Thickness of glass (m)	0.003
10.	Emissivity of glass	0.9
11.	Thickness of back insulation (m)	0.012
12.	Thermal conductivity of insulation ($W/m^{2\,0}C$)	0.072

Fabrication of Solar Air Dryer

Actual vertical cut section of the collector plate is as shown in figure1 is non selective type absorber plate fitted at decided depth ratio. This type is also called as suspended plate solar air dryer [7]. The absorber plate is made up of aluminum painted with Ferretol black paint. The absorber plate is suspended in a rectangular box of 56 mm thick made of MS sheet. This MS sheet box fitted inside a wooden box made up of 12 mm ply. A layer of 1.2 cm glass wool is used as insulation around metal sheet box. Drying cabinet is also made up of MS sheet having dimensions 0.3 m X 0.3 m. For cabinet also insulation is provided, fitted inside wooden box. In the box steel mesh is used as tray of size 10-20 mesh/cm^2. The glass is provided with rubber gasket to avoid leakage.

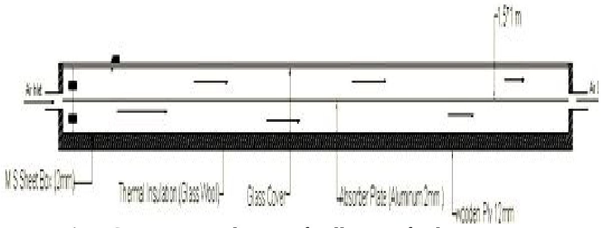

Fig.1 Cross sectional view of collector of solar air Dryer

Test rig for Solar Air Dryer

Fig. 2 showing schematic diagram and the actual experimental test rig of recirculation type passive solar air dryer. Collector will be fitted in South North direction at about 30 - 40^0 angles. Angle for collector tilt is adjustable. Solar dryer cabinet is attached using simple 1/2" MS pipe**.** Glass wool is used for insulation purpose. Air flow is measured using simple water U tube manometer. For measuring temperature at different locations multi temperature is used with RTD sensors as shown in figure 3.

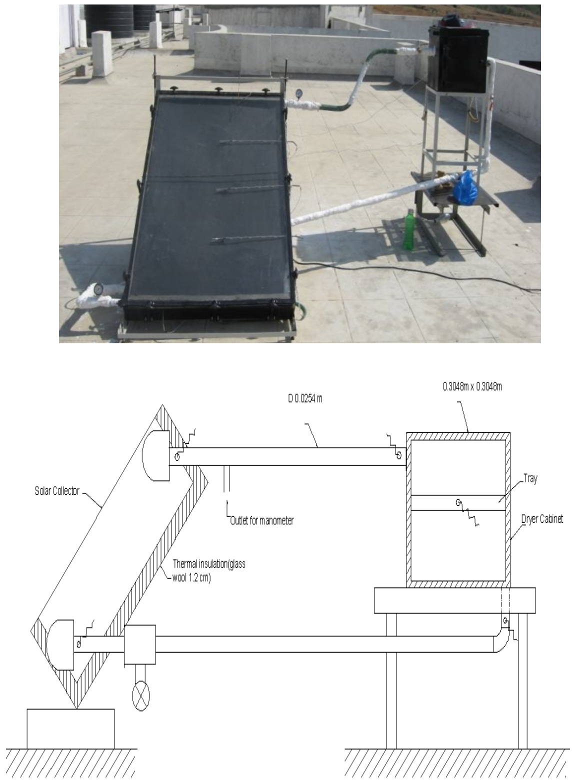

Fig.2 Experimental setup and Schematic Diagram and Test rig of Solar air Dryer

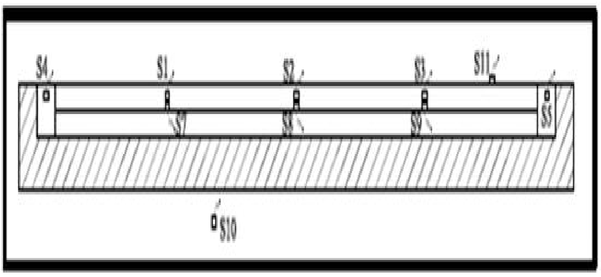

Fig. 3 Positions of Thermocouples in solar collector

RESULTS AND DISCUSSION

All the readings were taken during April – May 2010, atmospheric temperature is almost constant and in between 35 to 40^{0}C most of the time of the day. At the location, where average solar radiations falling is 2.5 to 4 KJ/cm^2/day. Three different types of geometries are considered in the solar collector for the analysis;

Type A. Plane sheet of the dimension 1015 mm × 1540 mm fitted in solar collector.

Type B. A corrugated plate of corrugations along the length of solar collector.

Type C. A corrugated plate of corrugations across the length of the solar collector.

The readings are taken for the duration of morning 10 am to 5 pm at the interval of one hour. For each type of collector the readings are recorded for a week duration and the average temperature are taken for the analysis. Figure 4 indicates variation of outlet temperature with respect to time from morning 10 am. Morning at 10.00 am, it is observed that the temperature of the Type A of collector is higher temperature than that of other two type because, the variation in density due to small variation in temperature is predominant in earlier section. Hence maximum temperature obtained in the collector outlet is faster. In Absorber type B, the resistant is low but the bulk of the collector is certainly high, so that time required to attending the temperature is high, hence the temperature is low. In absorber type C, there is resistance to flow as well as bulk is also sufficiently large, hence the temperature is further less. The Maximum Temperature observed is in between 1.00 pm to 2.00 pm. At this time sun position is nearly at the collector as well as the collector receives recirculated hot air from the oven cabimate hence the temperature at the inlet is high. Due to which the higher temperature is sufficiently high in this time. After 2.00 pm as sun position changes, this reduces the intensity of light falling on the collector.

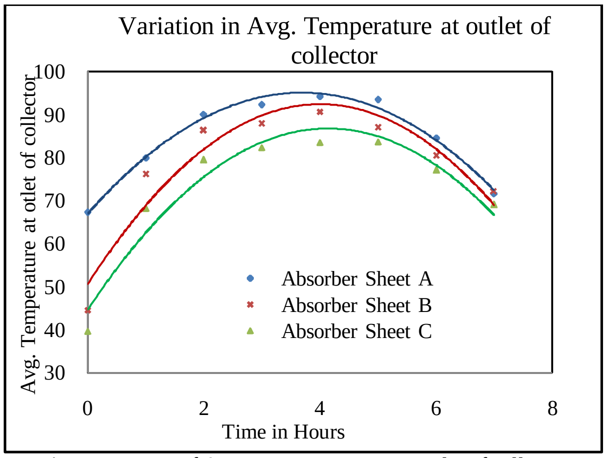

Fig. 4 Variation of Average Temperature at outlet of collector

Hence temperature starts dropping after 2.00 pm. From figure it is observed that the variation of temperature difference is sufficiently in case of flat plate type of collector at the morning as well as afternoon section because of less resistance offered to flow the hot in case of flat plate collector. Whereas in case of corrugated type of collector the increased bulk and the excessive resistance to flow reduces the temperature difference across the collector. Hence in corrugated type of collector the temperature difference is steadily rises as well as reduces at evening section as compared to flat plate collector.

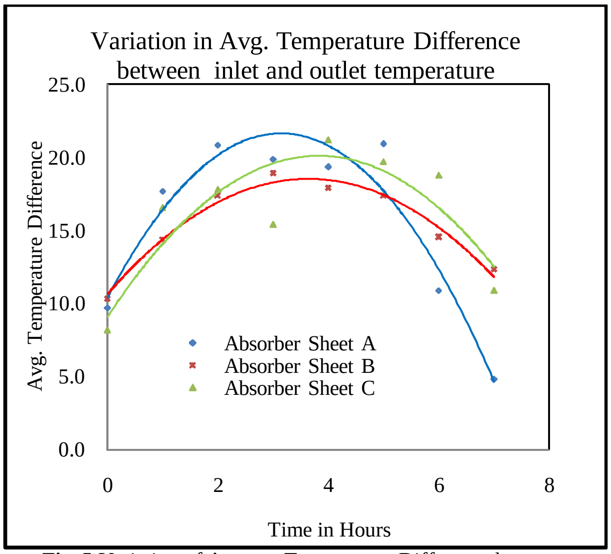

Fig. 5 Variation of Average Temperature Difference between inlet and outlet of collector

CONCLUSION

Available devices are not more efficient as user will like it. The reason is, each time cold incoming air is allowed to absorb heat. Such new air cannot exceed certain average temperature, thus, reduces the overall efficiency of the dryer.

Following inferences are drawn from results of this experimentation.

a) For Corrugations across air flow direction (C type plate) heat transferred to the circulating air is highest at 360 mm distance (at location S1) on absorber plate.
b) For duct ratio unity, after length 720 mm (at location S2), efficiency of transferring heat to air goes on decreasing for Flat plate (A type) and C type absorber plate.
c) For Corrugations along air flow direction (B type plate), efficiency of transferring heat to air goes on increasing (compare to S1 and S2) up to 1440 mm length.
d) In type B recirculation is possible for selected geometrical constraint.
e) Due to typical geometry, lowest radiation losses back to atmosphere from top cover are found in type B and highest radiation losses observed in plate A between 10.30 am to 12.30 pm.
f) For selected geometry of absorber surface type A , B and C gives best average temperature at about 700, 1100, 400 mm length of absorber plate respectively.

Although the benefits of solar air heating need to be quantified more thoroughly by collecting actual energy use data for each installation, there is sufficient evidence that the pilot solar air heating installations are saving resources and money. The options like switching from oil to solar should be made available with consideration of economics behind it and providing energy alternatives.

Nomenclature

D	Safe storage time (days).
D_R	Depth ratio.
G	Gravity constant (m/S^2)
H	Depth of the product layer (m)
H	The height between plate and the cabinet product (m)
I	Intensity of solar radiation falling on the absorber plate (W/m^2).
M	Air flow rate through the volume of product (kg/h).
M	Initial weight of the product (Kg).
M_V	Amount of water to be evaporated (kg H_2O).
$\dot{M}_V$	Total drying speed of the heated air (Kg H_2O/h).
M_{V1}	Drying speed of the ambient air (Kg H_2O/h).
M_{V2}	Drying speed cased by extra heat due to recirculation (Kg H_2O/h).
U_T	Total heat loss coefficient from the collector ($W/m^2\ {}^0K$).
V	Velocity of the air through the product (m/s).
X_1	Initial moisture content (%).
X_2	Final moisture content (%).
Y	Water absorption of the drying air (kg(H_2O)/Kg of air).
y_1	Water content of ambient air at saturation (Kg H_2O/ kg of air).
y_2	Water content of outlet air at saturation (Kg H_2O/ kg of air).
A	Absorption coefficient of the plate for solar radiation.
Γ	Heat vaporization of product (kWh/ Kg H_2O).
$\Delta\ \rho_{air}$	Difference in density of the air inside and outside the collector (kg/m^3)
ΔP	Pressure difference between heated air and the ambient air (Pa).
Δ_T	Temperature rise of the air (${}^0K$).
Δy	Change in the water content of saturated air (Kg H_2O/ kg of air).
P	Density of the product (kg/m^3).
ρ_{air}	Density of air at outlet (kg/m^3).
τ	Transmittance of glass covers for solar radiation.
Φ_2	Relative humidity of the ambient air (%).
Φ_1	Relative humidity of the outlet air (%).

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