Development of A Simulation Model For The Hybrid Solar Dryers As Alternative Sustainable Drying System For Herbal and Medicinal Plants

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Abstract. A simulation model of hybrid solar drying as alternative sustainable drying system for herbal and medicinal plants was developed. Heat absorbed from the solar radiation. Heat gained by the collector and supplemented by the burner. Heat gained or lost by the product, heat gained or lost through the drying bin wall, and the latent heat of the moisture evaporation from the product were the main components of the equations describing the drying system. The model was able to predict the moisture loss from the product at wide ranges of temperatures (55, 60, 65 and 70 C) and air circulation percentages (10, 20 and 30%). The model showed a dramatic effect of the drying air temperature on the moisture loss at the beginning of the drying process and became constant. Energy consumption at different drying temperatures was studied. Air recirculation has a profound effect on energy saving while drying herbs. In the next stages, high temperature without forcing air should be used. The model was validated by using experimental data of the drying temperatures and moisture loss under both direct sun and solar drying systems for herbal plants. The predicted values were in a reasonable agreement with the experimental data.

Keywords. Simulation model, hybrid solar dryer, herbal and medicinal plants, air recirculation, energy consumption.
Introduction

The whole industry of exporting dried herbs and medicinal plants are at risk. The drying cost of dried herbs using fossil energy with the governmental policy of liberalizing energy cost will become very crucial. At the moment, the drying cost of one Kg of mint ranges between LE 2-3 relevant to type of energy used, diesel and / or electric power. With the expected rising of energy cost, the total exporting cost will be critical compared to competitors from other producers such as India and elsewhere. This is beside the environmental hazards of using fuel as the source of energy.

Drying of herbs and medicinal plants is one of the oldest forms of food preservation methods known to man and is the most important process for preserving food since it has a great effect on the quality of the dried products. The major objective in drying agricultural products is the reduction of the moisture content to a level which allows safe storage over an extended period. Also, it brings about substantial reduction in weight and volume, minimizing packaging, storage and transportation costs (Okos, Narsimhan, Singh, & Witnauer, 1992). Solar energy is an important alternative source of energy and preferred to other energy sources because it is abundant, inexhaustible and non-pollutant. Also, it is renewable, cheap and environmental friendly (Basunia & Abe, 2001).

Thin layer equations describe the drying phenomena in a united way, regardless of the controlling mechanism. They have been used to estimate drying times of several products and to generalize drying curves. In the development of thin layer drying models for agricultural products, generally the moisture content of the material at any time after it has been subjected to a constant relative humidity and temperature conditions is measured and correlated to the drying parameters (Midilli, Kucuk, & Yapar, 2002; Togrul & Pehlivan, 2004).

Many researches on the mathematical modelling and experimental studies have been conducted on the thin layer drying processes of various vegetables, fruits and agro-based products such as bay leaves (Günhan, Demir, Hancioglu, & Hepbasli, 2005), hazelnut (O’demir & Devres, 1999), green pepper, green bean and squash (Yaldiz & Ertekin, 2001), apricot (Sarsilmaz, Yildiz, & Pehlivan, 2000; Togrul & Pehlivan, 2003), green chilli (Hossain & Bala, 2002), pistachio (Midilli & Kucuk, 2003), potato (Akpinar, Midilli, & Bicer, 2003a), apple (Akpinar, Bicer, & Midilli, 2003), pumpkin (Akpinar, Midilli, & Bicer, 2003b), red pepper (Akpinar, Bicer, & Yildiz, 2003), eggplant (Ertekin & Yaldiz, 2004), carrot (Doyraz, 2004), fig (Doyraz, 2005), Citrus aurantium leaves (Ait Mohamed et al., 2005), rosehip (Erenturk, Gulaboglu, & Gultekin, 2004), kiwi (Simal, Femenia, Garau, & Rosella, 2005).

Solar drying systems must be properly designed in order to meet particular drying requirements of specific products and to give satisfactory performance with respect to energy requirements. Designers should investigate the basic parameters namely dimensions, temperature, relative humidity, airflow rate and the characteristics of products to be dried. However, full scale experiments for different products, drying seasons, and system configurations are some times costly and not possible. The development of a simulation model is a valuable tool for predicting the performance of solar drying systems. Again, simulation of solar drying is essential to optimize the dimensions of solar drying systems and the optimization technique can be used for optimal design of solar drying systems [1–3]. However, there is no information about the mathematical models of the drying process using hybrid solar drying of herbs and medicinal plants in the literature. Therefore, the main objectives of this study are to develop an appropriate model for the hybrid drying system to study the effect drying temperature and air recirculation percentages on the drying time, final moisture content and energy consumption for some herbal plants. Testing and verifying the model results with the experimental results. Applying the model results to design and construct an appropriate hybrid solar dryer for some herbal and medicinal plants.

MODEL DEVELOPMENT

Heat and mass balances were carried out in order to describe the drying system. The temperature rise and moisture loss during the drying are described by means of the transient energy conservation equation, combined with an equation for the rate of moisture loss. However, the following assumptions were made in developing the model:

- The product is uniformity distributed in the drying space.
- Product leaves are characterized as homogenous objects and have a uniform temperature.
- A steady state condition is achieved.
- The coefficient of evaporation remains constant.

**Heat Balance:**

The heat balance equation is based on the concept that the algebraic summation of the rate of sensible energy gain, the absorbed solar heat, heat gain or heat loss from the dryer room, and the heat loss due to the moisture evaporation. These could be explained as follows:

1. Heat absorbed from the solar radiation.
2. Heat gained by the collector and supplemented by the burner.
3. Heat gained or lost by the product.
4. Heat gained or lost through the drying bin wall, and
5. The latent heat of the moisture evaporation from the product.

These components can be written as follows:

\[ Q_s + Q_c \pm Q_p \pm Q_w - Q_e = 0 \]  \(\text{(3)}\)

With reference to Fig. 1, the values of \(Q_s\), \(Q_c\), \(Q_p\), \(Q_w\) and \(Q_e\) can be calculated from the following equations:

\[ Q_s = \alpha_s H_s + \alpha_{\text{sky}} H_{\text{sky}} - Q_{\text{conv}} - E \] \(\text{(4)}\)

\[ Q_p = m_p c_p (T_{\text{in}} - T_{\text{amb}}) \] \(\text{(5)}\)

\[ Q_w = k_w A_w (T_{\text{in}} - T_{\text{amb}}) / L_c \] \(\text{(6)}\)

\[ Q_e = A_o C_t (P_s - P_{\text{amb}}) * Q_l \] \(\text{(7)}\)

**Solar radiation:**

The total solar radiation incident on a surface is the combination of the direct (subscript D), diffuse (subscript d) and ground-reflected (subscript r) irradiance of the surface, which gives:

\[ E_r = E_D + E_d + E_r \] \(\text{(8)}\)

The amount of solar irradiance was calculated according to ASHRE, 2009

**Solar collector:**

The energy balance for the air-flow through the collector given by equation (--) can be written to give the temperature rise across the collector in according to the equations cited from Dufee and Bechman, 1991

enthalpy of air entering the bed is equal to the sum of the enthalpy of ambient air and the enthalpy gain in the collector. The enthalpy gain in the collector can be expressed as:

\[ \Delta h = \frac{\eta_1 W_2 L}{m} \]

for an actual collector and as:
\[ \Delta h = \frac{Electric \ Power \ Supplied}{\dot{m}} \]

for the electrically heated simulated collector

where \( WC \) is the area of collector.

Now, the heat balance gives:

\[ h_s = h_a + \Delta h \]

where \( h_a \) is the enthalpy of ambient air and \( h_s \) is the enthalpy of air leaving the bed which is assumed to be the saturation enthalpy. The model utilizes the following relations for enthalpy and saturated humidity ratio. Those equations are approximations developed from the psychometric chart.

\[ h_a = (1 - \varphi) x T_a + \varphi x h_s \]
\[ h_s = 29.5 + (2.1 x(T-10)) + (0.07 x (T-10)^2) \]
\[ \Omega_{sat} = 0.0076 + (0.00045 x (T-10.0)) + (0.000027 x (T-10)^2) \]

First, the saturation enthalpy of ambient air is calculated from equation (75) then equation (74) is used to calculate the enthalpy of ambient air. The enthalpy of air leaving the grain bed is then computed using equation (73). Since the air leaving the bed is saturation, its temperature, which is the saturation temperature, can be obtained from equation (75).

\[ T_{sat} = 15x \left( \left(1 + \frac{h_s - 29.5}{15.75}\right)^\frac{1}{2} - 1 \right) + 10 \]

When the ambient and final temperatures are known, equation (76) is used to obtain the saturated humidity ratios of air entering the collector and that leaving the dryer. The humidity ratio of the air entering the collector can then be calculated using equation (4).

The rate at which moisture is removed can then be computed by applying a moisture mass balance for the product bed as follows:

\[ m_{rd} x \frac{dM_c}{dt} = \dot{m} x (\omega_{sat} - \omega_{ambient}) \]

where:
Mc = moisture content  
Mrd = mass of dry product in bed  

It is worth noticing that the enthalpy changes associated with moisture evaporation are far greater than the sensible heat changes associated with changes in bed temperature, therefore, bed sensible heat changes have been ignored in the above calculations.

Attention will now be turned to the equilibrium drying model. The equilibrium model is based on the assumption that near equilibrium conditions exist between drying air and grain. First, the conditions of air leaving the collector have to be determined. An energy balance for the collector allows the temperature of air leaving the collector to be calculated. This gives:

$$T_c = T_a + \frac{\Delta h}{C_a}$$

where $\Delta h$ is given by equation (71) or (72) depending on the type of the collector being used.

By using equation (76), the saturated humidity ratio for air entering the collector, $HR_{sat}$ and the air leaving the collector, $HR_{csat}$ can be determined. With the relative humidity of air entering the collector known, the relative humidity of air leaving the collector, $rh$, can be calculated using the relation:

$$\phi_c = \frac{\phi_{HR_{sat}}}{\phi_{HR_{csat}}}$$

The humidity ratio of the air leaving the collector is obtained using equation (4). With the conditions of air entering the rice bed now known, the equilibrium temperature can be obtained from equation (23). In the calculation of the equilibrium temperature, it is assumed that at the beginning of drying, the product has the temperature of ambient air. The equilibrium temperature then allows the equilibrium moisture content to be calculated. An average value of both the Chung-Pfost equation, equation (20), and the modified Henderson equation, equation (21), is used to obtain a better estimation for equilibrium moisture content. The thin layer equation, equation (25), allows the moisture content of the product $M_i(t)$, at time $t$, to be calculated. The mass of moisture transferred from the rice to each kg of air passing through the dryer during the time increment can be obtained as below:

$$\Delta \omega = \left(\frac{dM_e}{dt}\right) \frac{m_{rd}}{m}$$

The mass of water released by the rice and available for air to be removed per kg of air is given by equation (78):

where $M_e$ is the moisture content, $m_{rd}$ is the mass of dry rice and $m$ is the mass of air passing through the dryer during the time interval.

The mass of dry product can be calculated from:

$$m_{rd} = m (1.0 - M_{cw})$$
Now the humidity ratio of air leaving the bed is the sum of Au and humidity ratio of air entering the bed. However, air has a limited moisture carrying capacity. The maximum amount of moisture that one kg of air can remove is the difference between the saturated humidity ratio and the humidity ratio of that air at that temperature. Now as air passes through the rice bed, it cools down. The resulting temperature decrement can be found by equating the latent heat of vaporization from grain to the sensible heat change of drying air.

\[ h_{fg} \Delta \omega = C_d \Delta T \]

where:
\[ h_{fg} = \text{latent heat of vaporization, J/kg} \]
\[ C_d = \text{Specific heat capacity of air, J/kg.C} \]

with air absorbing moisture it cools down, as it cools down its ability to absorb moisture decreases. Therefore an iterative solution procedure has to be used to find the mass of actual moisture removed. The moisture exceeds the actual value of moisture removed is assumed to be condensed back to the rice. In this procedure, Aw is calculated using equation (81). Equation (83) is then used to find the temperature decrement, \( \Delta T \), If the calculated exit temperature indicated that the moisture content of the air in fact exceeds the saturation moisture content, a new value of \( \Delta \omega \) is guessed and the process is repeated until the conditions that give saturation at bed exit are found. With the actual value of the quantity \( \Delta \omega \) is known, a corrected moisture content of the rice is then calculated from equation (81).

### 2.2.15 The Equilibrium Moisture Content

The equilibrium moisture content, EMC, is the moisture content of a product that is in equilibrium with air at a particular mean dry-bulb temperature and relative humidity that would be attained by the grain over infinite time, at a constant value of air relative humidity and temperature. The equilibrium moisture content is expressed as a decimal on a dry basis. Several models, theoretical as well as empirical, have been suggested for the calculation of the EMC. The following Chung-Pfost and the modified Henderson equilibrium equation\(^4\) applies well to product:

\[
E_{MC} = E - F x \ln[-(T_e - C) x \ln(\Phi)]
\]

where:
\[ E_{MC} = \text{drying air equilibrium moisture content, dry basis, decimal} \]
\[ T_e = \text{the equilibrium temperature of drying air, Celsius} \]
\[ \Phi = \text{relative humidity of drying air, decimal} \]
\[ C, E, \text{and } F \text{ are equilibrium constants and have the following values for product.} \]
\[ C = 35.703 \]
\[ E = 0.29394 \]
\[ F = 0.046015 \]

The equilibrium moisture content can also be expressed by the modified Henderson Equation:
The variation of the equilibrium moisture content for rice with relative humidity at different temperatures.

2.2.16 Equilibrium Temperature of Drying Air

The drying process takes place at a temperature that is between the temperature of the air entering the crop bed and that of the air exiting the dryer. The equilibrium temperature of drying air can be calculated by establishing a heat balance between the initial conditions and the equilibrium conditions and assuming that the temperature of the equilibrium equals that of the grains and the humidity ratio at equilibrium equals that of drying air.

\[
E_{sfe} = \left[ \frac{\ln \left( 10 - \theta \right)}{-1.9187 \times 10^{-5} (T_E + 51.161)} \right]^{0.40898}
\]

The variation of the equilibrium moisture content for rice with relative humidity at different temperatures.

\[
E_{sfe} = \left[ \frac{\ln \left( 10 - \theta \right)}{-1.9187 \times 10^{-5} (T_E + 51.161)} \right]^{0.40898}
\]

which gives:

\[
T_E = \frac{C_\alpha T_r + C_v T_{Gr} \omega_r + C_G \frac{m_{rd}}{m}}{C_\alpha + C_v \omega_r + C_G \frac{m_{rd}}{m}}
\]

where:
2.3.1 Heat Transfer Coefficient

Heat is transferred from the absorber plate to the air by convection. In order to calculate the heat transfer coefficient between the plate and air, the character of the flow must be determined as turbulent or laminar. The heat transfer coefficient can be calculated from the following relationship:

\[ h = \frac{N_{UL} k}{L} \]

where:
- \( N_{UL} \) = average Nusselt number based on the length of collector
- \( k \) = thermal conductivity of air, W/m.K
- \( L \) = characteristic length of the system (taken as the length of the collector)

For natural convection laminar flow, the Nusselt number can be calculated from the following relationship:

\[ \frac{N_{UL}}{R^{0.5}_{eL}} = 1.1 \]

For forced convection laminar flow:

\[ N_{UL} = 0.664 R^{0.5}_{eL} Pr^{0.33} \]

where:
- \( Pr \) = Prandtl number
- \( R^{0.5}_{eL} \) = Reynolds number based on the length of the collector
The above equation can be rewritten in the form the heat transfer coefficient for forced convection laminar flow can then be expressed as:

$$\frac{N_{UL}}{Re^{0.8}} = 0.59$$

It is important to keep in mind that the above relations are valid for Pr > 0.1 which is the case for air.

$$h = 0.59 \frac{kRe^{0.5}}{eL}$$

3.2 COMPUTATIONAL PROCEDURE

The following flowchart is programmed using Excel Spreadsheet. Next tables represent the inputs and the outputs for the simulation model.

3. Experimental procedures

The hybrid solar dryer consists of the following components:
1- The cabinet dryer,
2- The solar collector, and
3- The supplementary power source.

1- Components of the cabinet dryer (Figure 15):

The drying chamber:
The drying chamber is the part of the dryer where the product to be dried are fed and drying takes place. The product are fed into trays first and these trays are then fed into the drying chamber.

Principle of operation of the cabinet dryer:
The products which are to be dried are loaded into the trays and the trays are then fed into the drying chamber and the door is closed sealing the system. The trays have perforated bottom. The burner incorporates a switch which has a sparking mechanism when turned clockwise. This spark produced by the clockwise rotation of the switch ignites the gas and produces a light blue flame which heats up the drying chamber. When moisture from product is vaporized, they pass through an outlet to the surrounding.

The proposed improved design of the cabinet dryer:
The improvement on the existing design of the existing cabinet product dryer has been carried out based on the problems associated with the existing design i.e., to improve upon the heat distribution, safety and ergonomics of the machine.

Design considerations:
- Drying of products to take 4 h.
- Mint is used as an example to cover a large range of products.
- With respect to ergonomics, the trolley has been made to be high enough to allow easy rolling in and out of the cabinet without unnecessary bending of the operator.
Figure: Flow chart for the experiment simulation program
The Cabinet Dryer Developed
Components of the improved dryer:
The dryer consists of three major components, namely:
- The drying chamber,
- The trolley and trays,
- The fan/blower, and

The drying chamber:
The drying chamber has a length of 2.5 m, width of 2.3 m and height of 2.6 m. It is made of galvanized steel (5 mm thickness). The selection of galvanized steel in this regard is due to its strength and heat transfer properties. Inside the drying chamber is painted with silver to reduce heat loss by radiation and each side of the chamber is insulated to reduce heat loss.

The trolley:
The trolley has a length of 2.3 m, width of 1.1 m and height of 2.4 m. The trolley is designed in such a way that it allows easy insertion of individual trays at a distance of 0.2 m apart and has tyres for easy movement of trays. The material for the trolley is stainless steel. The reason for selecting this material is due to the structural properties and its weldability.

The trays:
The trays are made of stainless steel and have a length 1.1 m, width 0.74 m and depth of 0.03 m. The trays have perforated bottom which allows heated air to pass through products.

The fan/blower:
The fan aids in heat distribution by drawing ambient air from the surrounding to the heater housing and discharging heated air to the drying chamber. A proper fan has to be selected so that proper distribution of heat is achieved.

Selection of the fan:
Length, width and height of drying chamber are 2.5, 2.3 and 2.6 m respectively:
Volume of drying chamber = 2.5×2.3×2.6 = 14.95 m³
Length, width and height of trays are 1.1 m, 0.74 m and 0.15 m respectively:
Volume of one tray = 1.1×0.74×0.03 = 0.0224 m³
Distance between subsequent trays = 0.05 m
Intended drying time = 4.0 h
Height at which products will fill the trays = 0.15 m

Volume of products per tray = 0.15×1.10×0.74 = 0.1221 m³
Total volume of products = Number of trays × volume per tray = 66×0.1221 = 8.058 m³

Using mint as an example,
bulk density of mint = 31.0 kg m⁻³
Total mass of mint = bulk density × volume = 31.0 kg m⁻³ × 8.058 m³ ≈ 250.0 kg

Moisture content = \( \frac{(M_w - M_d)}{M_w} \times 100 \)
Where:  \( M_w \) = Mass of wet product
\( M_d \) = Mass of dry product
From the moisture content table, optimum moisture content for mint when harvested = 80%.
Therefore:
\( \frac{(M_w - M_d)}{M_w} \times 100 = 80 \) \( \frac{(250.0 - M_d)}{250.0} \times 100 = 80 \) \( M_d = 87.5 \) kg
Mass of water = Mass of wet product (\( M_w \)) - Mass of dry product (\( M_d \))
Mass of water \[= 250.5 - 87.5 = 162.5 \text{ kg}\]

Quantity of heat required to remove moisture content:

\[\text{Latent heat of vaporization of water: } 2,385.29 \text{ kJ/kg}_{\text{water}}\]

Heat quantity for water removal \[= 162.5 \text{ kg} \times 2,385.29 \text{ kJ/kg}_{\text{water}} = 387,609.62 \text{ kJ}\]

Power \[= \text{Quantity of heat/time (sec)}\]

Intended drying time = 240 min = 240\times60 \text{ sec}

Power \[= 387,609.62 \text{ kJ} / (240\times60)\]

Power \[= 26.9 \text{ kW}\]

Work done by heater \[= \text{Work done on air}\]

Work done by heater \[= 27 \text{ kW}\]

Work done on air \[= \text{Mass flow rate of air} \times \text{Specific heat capacity of air} \times \text{Temperature difference}\]

Specific heat capacity of air \[= 1.005 \text{ kJ kg}^{-1} \text{K}^{-1}\]

Work done by heater \[= \text{Mass flow rate} \times \text{Specific heat capacity} \times \text{Temperature difference}\]

\[
27 \times 10^3 \text{ W} = \text{Mass flow rate kg sec}^{-1} / (1.005 \times 10^3 \times 55) = \]

Mass flow rate kg \text{ sec}^{-1} \[= 27 \times 10^3 \text{ W} / (1.005 \times 10^3 \times 55) = 0.4885 \text{ kg sec}^{-1}\]

From appendix, density of air at 40°C \[= 1.127 \text{ kg m}^{-3}\]

Specific volume \(v\) \[= 1/1.127\]

Discharge \[= \text{Mass flow rate} \times \text{Specific volume} = 0.4885 \times 0.8873 = 0.4335 \text{ m}^3 \text{ sec}^{-1}\]

\[= 0.4335 \text{ m}^3 \text{ sec}^{-1} \times 3,600 \text{ sec} / \text{h} = 1560 \text{ m}^3 \text{ h}^{-1}\]

Efficiency of blower \[= 20\%\]

Blower discharge demand \[= \text{Discharge m}^3 \text{ h}^{-1} / \text{Efficiency} \%\]

\[= 1560 \text{ m}^3 \text{ h}^{-1} \times 0.2\]

\[= 7800 \text{ m}^3 \text{ h}^{-1}\]

2- **The Solar Collector**:

The solar collector consists of three major components, namely:

- The glass cover has a sheets (12 sheets, 2.0×2.0m, 5.5 mm thickness). The reason for selecting this material is due to the structural thermal properties.
- The absorber plate, (corrugated black aluminum corrugated).
- The insulation, (thermal wool, 5.0 cm thickness).

Selection of the solar collector area:

Total Power demand \[= 27 \text{ kW.h}\]

Solar energy available \[= \text{Average solar radiation (kW/m}^2\text{)} \times \text{Collector Area (m}^2\text{)}\]

Collector Area \(\text{m}^2\) \[= \text{Total Power demand (kW)} / \text{Average solar radiation (kW/m}^2\text{)}\]

\[= 27 / 0.6 \quad = 45 \text{ m}^2 \approx (4.0 \times 12.0 \text{m})\]
### 3.1. SOLAR CALCULATIONS

Table (1): Input Parameters.

<table>
<thead>
<tr>
<th>Time Zone</th>
<th>SURFACE:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Month = 3</td>
<td>Azimuth = 0</td>
</tr>
<tr>
<td>Day = 17</td>
<td>Tilt (β) = 30</td>
</tr>
<tr>
<td>Longitude = 33</td>
<td></td>
</tr>
<tr>
<td>Latitude (L) = 28.4</td>
<td></td>
</tr>
<tr>
<td>Clearness number (CN) = 1</td>
<td></td>
</tr>
<tr>
<td>Ground Reflectivity (ρg) = 0.6</td>
<td></td>
</tr>
<tr>
<td>Day of year (η) = 76</td>
<td></td>
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<tr>
<td>Equation of Time (ET) = -9.066</td>
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<tr>
<td>Declination (δ) = -2.016</td>
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</tr>
<tr>
<td>Optimum Tilt Angle at Noon (β) = 30.416</td>
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<tr>
<td>A = 1166.701</td>
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<tr>
<td>B = 0.148</td>
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<td>C = 0.108</td>
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<tr>
<td>Solar Time Correction = 2.049</td>
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<tr>
<td>No. of Bright Sunshine Hours = 11.855 h</td>
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</tr>
<tr>
<td>Sunrise Time (Tsr) = 6.073 = 6 h 4 min 21.7</td>
<td></td>
</tr>
<tr>
<td>Sunset Time (Tss) = 17.927 = 17 h 55 min 38.3</td>
<td></td>
</tr>
<tr>
<td>Start Time = 1:00 AM</td>
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<tr>
<td>Step Time = 0:15</td>
<td></td>
</tr>
</tbody>
</table>
Table (2) shows the predicted solar radiation (W/m²), the collector efficiency and useful heat gain used to heat the air passing through the collector for the dryer.

Table (2): Output Data

<table>
<thead>
<tr>
<th>Time</th>
<th>Hour Angle</th>
<th>Altitude</th>
<th>Zenith Angle</th>
<th>Tilt Angle</th>
<th>Solar Incidence</th>
<th>Solar Radiation</th>
<th>Useful Heat Gain</th>
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</thead>
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<tr>
<td>AST</td>
<td>0°</td>
<td>Ψ</td>
<td>Ζ</td>
<td>βo</td>
<td>0</td>
<td>W/m²</td>
<td>W/m²</td>
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<td>10:57 PM</td>
<td>-165.00</td>
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<td>153.12</td>
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<td>2:00 AM</td>
<td>11:57 PM</td>
<td>-150.00</td>
<td>-61.81</td>
<td>151.81</td>
<td>151.81</td>
<td>143.00</td>
<td>0.00</td>
</tr>
<tr>
<td>3:00 AM</td>
<td>12:57 AM</td>
<td>-135.00</td>
<td>-48.66</td>
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<td>138.66</td>
<td>130.56</td>
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<td>-120.00</td>
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<td>125.50</td>
<td>117.19</td>
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<td>99.78</td>
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<td>87.54</td>
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<td>12.51</td>
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<tr>
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<td>13.95</td>
<td>76.05</td>
<td>76.05</td>
<td>61.52</td>
<td>314.45</td>
</tr>
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<td>24.28</td>
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<td>65.72</td>
<td>47.92</td>
<td>537.70</td>
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<tr>
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<td>32.81</td>
<td>57.19</td>
<td>57.19</td>
<td>35.05</td>
<td>698.19</td>
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</tr>
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<td>87.54</td>
<td>75.43</td>
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</tr>
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<td>-61.81</td>
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<tr>
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<td>8:57 PM</td>
<td>165.00</td>
<td>-74.47</td>
<td>164.47</td>
<td>164.47</td>
<td>153.12</td>
<td>0.00</td>
</tr>
</tbody>
</table>
Table (3): Dryer Specifications:

<table>
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<tr>
<th>Start</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Step</td>
<td>1:00</td>
</tr>
<tr>
<td>Type of Flow</td>
<td>1</td>
</tr>
<tr>
<td>Air Flow</td>
<td>7200 m³/h</td>
</tr>
<tr>
<td>Mixing Ratio</td>
<td>0.3</td>
</tr>
<tr>
<td>Type of Dryer</td>
<td>1</td>
</tr>
<tr>
<td>Type of Burner</td>
<td>1</td>
</tr>
<tr>
<td>SH</td>
<td>52250</td>
</tr>
<tr>
<td>BE</td>
<td>0.92</td>
</tr>
<tr>
<td>η</td>
<td>0.3</td>
</tr>
<tr>
<td>Maximum Temperature for Drying</td>
<td>60</td>
</tr>
<tr>
<td>LENGTH OF DRYING BED (M)</td>
<td>DD</td>
</tr>
<tr>
<td>WIDTH OF THE DRYING BED (M)</td>
<td>WD</td>
</tr>
<tr>
<td>HEIGHT OF Product BED ABOVE COLLECTOR OUTLET (M)</td>
<td>H</td>
</tr>
<tr>
<td>MOISTURE CONTENT AT THE BEGINNING OF TIME INTERVAL,db i</td>
<td>M I</td>
</tr>
<tr>
<td>MOISTURE CONTENT AT THE FINAI OF TIME INTERVAL,db i</td>
<td>Mf</td>
</tr>
<tr>
<td>MASS OF WET Product (KG)</td>
<td>MR</td>
</tr>
</tbody>
</table>

Table (4): Collector Calculation Data:

| Lc | collector solar energy collection length, m | 12 |
| Wc | collector solar energy collection width, m | 4 |
| Ac | collector solar energy collection area, m² | 48 |
| Cp | constant pressure specific heat, J/(kgK) | 1009 |
| Fo | efficiency factor of solar collector | 0.77 |
| g | gravitational acceleration, m/s² | 9.81 |
| GT | incident solar energy, W/m² | 1041 |
| m | air flow rate, kg/s | 2 |
| V | wind speed, m/s | 2.4 |
| hw | wind convection coefficient, W/(m²K) | 10 |
4. Model Verification

The model is capable to predict the output temperature from solar collector under various levels of solar radiation. The predicted results were compared to the measured data from a solar collector.

A solar collector with same dimensions and specification predicted by the model was built to verify the model results. On the same day of 17 March 2009, the temperature of the heated air of the collector was recorded and compared to the model results, (Fig. 1A).

Correlation, Regression and Relative Percentage of Error, RPE, \([(\text{Actual} – \text{Prediction})/\text{Actual}]\) were used as indicators of the level of agreement degree between the predicted and measured values.

The simulated temperature was fluctuated between 0.66 to -3.70 °C higher and lower than the measured temperatures for most of the 12-hour simulation (Figures 1A, 1B). The RPE for the 12 hours of simulation was -0.60% and the correlation coefficient between simulated and measured temperatures was 0.966.

![Figure (1A): Measured and predicted temperatures.](image)

After building the dryer prototype in the next phase. Complete validation of the model will be carried out including the drying phase.
Figure (1B): Measured and predicted temperatures.
5- Model Experimentation.

5-1. Useful heat gain:

The parameters of model input (table 1) of twelve selected days, a day of each month, were fed to the model. Table (5) and figure (2) show the output. It is obvious that during the summer months, the heat gain reached 200 kW per day as the radiation exceeds 400 kW per day for the 48 m² collector. In January it was 180 kW heat gain reduced to 150 kW per day in December.

Using these figures as table (5), the total heat gain per year amounts to 80,000 kW which cost financially L.E. 6,000. If we generated from fossil energy and economically L.E. 29,600.

5-2. Simulating a drying process.

Mint is used as an example for result of the drying process. The following table shows the parameters of equation (50) and table (1) shows the input parameters with respect to the mint.

<table>
<thead>
<tr>
<th>Te</th>
<th>= the equilibrium temperature of drying air, Celsius</th>
<th>Calculated</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ph</td>
<td>= relative humidity of drying air, decimal</td>
<td>Calculated</td>
</tr>
<tr>
<td>C</td>
<td>Constant</td>
<td>= 35.703</td>
</tr>
<tr>
<td>E</td>
<td>Constant</td>
<td>= 0.29394</td>
</tr>
<tr>
<td>F</td>
<td>Constant</td>
<td>= 0.046015</td>
</tr>
</tbody>
</table>

Model results as shown in figures (3, 4, 5, 6 and 7)

Simulation model run at 55-70°C drying temperature. Model results as shown in figures (8, 9, 10, 11 and 12). The predicted the moisture content of the product was reduce from 80% to 11% (the desired ratio) in about 8.5 h and 4 h for during temperature of 55 and 70°C, respectively. Figures 4 and 9 show the fuel consumption with and without the collector in both cases, i.e., during temperature of 55 and 70°C, respectively.

Figures 5 and 10 show the equilibrium moisture content of the mint, with figures 6 and 11 show the mass of the product at the beginning and at the end of the drying process.

Figures 7 and 12 show that: Inlet temperature of the collector (Ti), outlet temperature of the collector, temperature air mixing, temperature into the product and temperature out the product.

Conclusions

A simulation model of hybrid solar drying as alternative sustainable drying system for herbal and medicinal plants was developed. Heat absorbed from the solar radiation. Heat gained by the collector and supplemented by the burner. Heat gained or lost by the product, heat gained or lost through the drying bin wall, and the latent heat of the moisture evaporation from the product were the main components of the equations describing the drying system. The model was able to predict the moisture loss from the product at wide ranges of temperatures (55, 60, 65 and 70°C and air circulation percentages (10, 20 and 30%). The model showed a dramatic effect of the drying air temperature on the moisture loss at the beginning of the drying process and became constant. Energy consumption at different drying temperatures was studied.
References


