Numerical modeling of the thermal behavior of a direct solar dryer with auxiliary heating system

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Abstract
In this study, we present thermal results related to the heating effect on the microclimate of a direct solar dryer (DSD). The heating effect is tested under two types of climatically different days: a clear day and a cloudy day. A comparison of the results of the numerical simulation and the experimental tests showed a good agreement and covered two types of heating systems: one radiative and the other convective. Their performance, which depends at the same time on the ambient climatic conditions and the thermal performance of the dryer, is decisive as to the advantage and the disadvantage of each and their appropriation to maintain an appropriate setpoint temperature.

1. Introduction
The spontaneous establishment of a microclimate inside a direct solar dryer (DSD) is a direct consequence of the interactions between the components of the dryer (roofing, inside air, soil, etc.) and variables characterizing climatic conditions (solar radiation, ambient temperature, wind speed, etc.). During a day, the thermal behavior of the dryer is different between day and night. Good sealing of the dryer and low transmittance of the cover to infrared radiation are in demand during the night, because they allow a reduction of energy losses. While satisfactory transmission of solar radiation inside the dryer and adequate ventilation are required during the day, it is necessary to increase the greenhouse effect and to promote the evacuation of moist air. During winter, the weather conditions are such that during the night, temperatures can reach low values causing a significant drop in temperatures inside the dryer. In order to maintain an acceptable set temperature, in relation to the product to be dried, the period and the dryer implantation site, heating is then necessary although its use generates heavy loads due to the increase in the energy cost and occasions the installation of infrastructures that are difficult to handle.
In this study, it is proposed to determine the thermal performance of a (DSD) using a theoretical model for the numerical simulation of its thermal behavior. This model is based on the classical heat transfer equations between the different components of the system and the external environment. The simulation results are compared with the experimental ones.

2. Experimental details
2.1. Site of the installation
The experimental installation is located at the Faculty of Sciences of Rabat. Located on the west coast of Morocco, at 65 m altitude, the climate of the site of Rabat is of the Mediterranean type with a rainy and cold winter and a hot and humid summer. The ambient temperature is moderate throughout the year and represents a small daily amplitude, with an annual average close to 18.4 °C. The average seasonal temperatures are 23.1 °C in summer, 17.8 °C in autumn, 14.9 °C in winter and 18.7 °C in spring. The minimum average is close to 9.9 °C
and maximum is close to 26.2 °C. The annual solar radiation is about 1900 kWh/m² with an average insolation time of about 2770 hours per year [1].

2.2. Solar dryer
The different solar fluxes intercepted by the dryer cover (consisting of five facades) are calculated for a prototype solar dryer of standard geometry [2-8]. Four facades are made of transparent glass with a thickness of 6 mm, and the fifth one (wall facing north) of opaque plastic. A perspective view of the used dryer is shown schematically in Figure 1, below.

2.3. Measured variables
The Measured variables concern the climatic ones that characterize the external environment and the inner microclimate of the dryer. Using a CR10X data acquisition system with AM25T extension, measurements are collected every 15 minutes, and then stored in a computer file. Particular attention was accorded to the calibration of the various sensors and measuring instruments. Systematic controls and permanent verifications of the measurements were carried out.

2.3.1. Climate variables
The external ambient temperature and relative humidity are measured using K-type thermocouples and J & R hydroelectric instruments, respectively. The speed and the direction of wind are measured at a level of 10 m from the ground. The global horizontal radiation is measured by means of two calibrated Eppleypyranometers, placed above and under the glazed cover, respectively.

2.3.2. DS D operating temperatures
In addition to the climatic variables measured outside the dryer, its thermal performances are determined from its operating temperatures that relate to the temperature and relative humidity of inside air, glazing and floor temperatures. The measurement of the global radiation transmitted inside the dryer makes it possible to determine the transmission coefficient of the glazing of the cover.

2.4. Mathematical formulation
2.4.1. Limit conditions
The solar dryer is a physical system composed of nine elements: the cover (c)(five elements), the inside air (ai), the inside air vapor mass (Wai), the upper surface layer of the soil (So) and the lower soil layer (S). Nine energy balance equations are therefore needed to describe the system. Each equation is written in terms of the exchange of thermal powers. The explanation and determination of each term are given in Annex A. To define the conditions at the limits of the dryer, the following climatic variables will be considered constant during each simulation step: global solar radiation (Sg), ambient air temperature (Tt = 0) and relative humidity (HRe), Wind speed (Ue) and deep soil temperature (Tsp). The initial conditions at time t = 0 are:

\[ T_{c1} (t=0) = T_{c2} (t=0) = T_{c3} (t=0) = T_{c4} (t=0) = T_{c5} (t=0) = T_{pl} (t=0) = T_{so} (t=0) = T_{sp} (t=0) = T_{sw} (t=0) = T_{sw} (t=0). \]

Temperatures \( T_{c} \) correspond to the different faces of the glazed cover of the dryer. The plastic wall is characterized by its \( T_{pl} \) temperature.
2.4.2. Equations of thermal balances

Figure 2 schematizes the set of heat flows exchanged per unit of exchange surface area, between the dryer and its environment:

By applying the principle of energy conservation for each element of the dryer, the balance equations are expressed using these thermal fluxes according to the relationship [9-15]:

\[ \Sigma \text{Contributions} - \Sigma \text{Losses} = \text{Variation of internal energy.} \]

For each element i of the dryer:
- the absorbed solar powers will be designated by \( Q_i^S \), respectively;
- the infrared radiative powers exchanged between elements’ and j ≠ i, as well as the external environment (outside ground, outside air, celestial vault) will be designated by \( Q_i^R \);
- the sensible convective powers exchanged between the dryer element i and the indoor or outdoor air will be designated by \( Q_i^C_{\text{air}} \);
- the latent convective powers exchanged between the element i of the dryer and the indoor or outdoor air will be designated by \( Q_i^L_{\text{air}} \);
- the power exchanged by conduction at the soil layers So, S and Sp will be designated by \( Q_i^G_{S,S_i} \).

The expressions of the different heat flux powers involved in the balance equations (relations 1 to 9) as well as the different coefficients of exchange and the choice criteria of the convective modes of exchange (laminar or turbulent regimes, natural or forced convection mode) will be given in Annex A.

2.4.2.1. Thermal balance of coverage

The cover is composed of five surfaces \( S_i \) of which four are transparent glass of 6mm thickness, and the fifth is opaque plastic insulated 10mm thick. Each wall \( S_i \) will be characterized by its temperature \( T_{vi} \) (the index i denotes the different glazed faces of the dryer). The plastic wall, facing north, is characterized by its \( T_{pl} \) temperature.

The geometric and physical characteristics of the five walls of the cover and soil are given in Table 1 of Appendix B. The balance equations of surfaces \( S_i \) of the cover are expressed using relations (1) to (5):

- **Surface \( S_1 \):**
  \[
  Q_{v1}^S - Q_{v1,v2}^R - Q_{v1,v3}^R - Q_{v1,v4}^R - Q_{v1,pl}^R - Q_{v1,so}^R - Q_{v1,ae}^R - Q_{v1,al}^C - Q_{v1,al}^L = \rho_v c_v V_{v1} \frac{dT_{v1}}{dt} \tag{1}
  \]

- **Surface \( S_2 \):**
  \[
  Q_{v2}^S - Q_{v2,v1}^R - Q_{v2,v3}^R - Q_{v2,v4}^R - Q_{v2,pl}^R - Q_{v2,so}^R - Q_{v2,ae}^R - Q_{v2,al}^C - Q_{v2,al}^L = \rho_v c_v V_{v2} \frac{dT_{v2}}{dt} \tag{2}
  \]

- **Surface \( S_3 \):**
  \[
  Q_{v3}^S - Q_{v3,v1}^R - Q_{v3,v2}^R - Q_{v3,v3}^R - Q_{v3,pl}^R - Q_{v3,so}^R - Q_{v3,ae}^R - Q_{v3,al}^C - Q_{v3,al}^L = \rho_v c_v V_{v3} \frac{dT_{v3}}{dt} \tag{3}
  \]
The method makes it possible to calculate the coordinates of the following point \( t \) chosen. It is based on a

Among the many algorithms used to solve this type of equation system, the Runge-Kutta method of order 4 was chosen. For a given depth \( z \), we consider in a first approximation that the temperature \( T_{sp} \) of the deep soil remains constant.

The thermal balance of the lower soil layer is:

\[
Q_{so, v1} - Q_{so, v2} - Q_{so, v3} - Q_{so, v4} - Q_{so, pl} - Q_{so, ae} - \rho_{so} c_{so} V_{so} = \frac{dT_{so}}{dt}
\]  
\( n \)

In this equation, the power exchanged by conduction between the lower soil layer \( S \) and the deep soil \( S_p \) depends on the temperatures \( T_s \) and \( T_{sp} \).

The thermal balance of the soil surface layer \( S_0 \) is written from the thermal powers exchanged by sensible convection between the inside air and the outside air of the dryer by natural renewal through the structure of the dryer.

\[
Q_{ai, Ae} = \frac{dT_{ai}}{dt}
\]

the supply of heating power of inside air which will be considered zero in the absence of heating.

The equation governing the water balance of inside air is based on the thermal powers exchanged by latent convection between the inside air and the other dryer elements:

\[
Q_{ai, Ae} = \frac{dT_{ai}}{dt}
\]

The equation governing the water balance of the inside air is written from the thermal powers exchanged by sensible convection between the indoor air and the other elements of the dryer:

\[
Q_{ai, Ae} = \frac{dT_{ai}}{dt}
\]

\[
C_{ai} \text{ is the specific heat of air (J.kg}^{-1}.K^{-1})
\]

\[
Q_{ai, Ae} \text{ is the thermal power exchanged between the inside air and the outside air of the dryer by natural renewal}
\]

\[
Q_{ai, Ae} \text{ is the latent heat power exchanged by natural renewal between the indoor air and the outside air, through}
\]

Even resolution method

Equations (1) to (9) form a system of first-order differential equations of the type:

\[
\frac{dY_i}{dt} = f_i \left[ t, Y_1 (t), Y_2 (t), ..., Y_n (t) \right]
\]

Among the many algorithms used to solve this type of equation system, the Runge-Kutta method of order 4 was chosen. It is based on a limited development of the functions, \( Y_i \) defined as follows:

\[
Y_i (t_o + \Delta t) = Y_i (t_o) + Y_i' (t_o) . \Delta t + \frac{Y_i'' (t_o) . \Delta t^2}{2} + \ldots + \frac{Y_i^n (t_o) . \Delta t^n}{n!}
\]

The method makes it possible to calculate the coordinates of the following point \( (t_1, Y_1) \), then \( (t_2, Y_2) \) and so on, with \( t_1 = t_o + \Delta t, t_2 = t_1 + \Delta t, ..., t_n = t_{n-1} + \Delta t \), where \( \Delta t \) represents the time step.
In the system of equations (1) to (9), the initial conditions corresponding to the temperatures measured at the time \( t_0 \) are indicated previously (see § 4.1):

These boundary conditions are imposed by the characteristics of the solar dryer. The inputs of the simulation program correspond to the climatic parameters \( S_g \), \( T_ae \), \( H_{Re} \), \( U_e \) and \( T_{sp} \) which are regularly measured every 15 minutes.

The following assumptions were considered:

- heat transfers are assumed to be unidirectional;
- the temperatures of each element of the dryer are assumed to be uniformly distributed;
- during a simulation step, the ambient temperature \( T_{ae} \), the global solar radiation \( S_g \) and the wind speed \( U_e \) are assumed to remain constant or vary slightly;
- the constructive and operational parameters of the solar dryer (covering, soil, indoor air, etc.) are not explicit functions of time, they are always implicit functions of their temperature;
- the temperature of the surrounding soil surface \( T_{se} \) is assumed to be equal to that of the ambient temperature \( T_{ae} \), and the apparent temperature of the sky \( T_{sc} \) is calculated by the relation:

\[
T_{sc} = T_{sn} \times \frac{N}{8} + T_{ae} \times \left(1 - \frac{N}{8}\right)
\]

Where \( T_{sn} \) is the temperature of a covered sky, estimated from Arinze's relation [16]:

\[
T_{sn} = T_{ae} - 6 \quad (T_{sn} \text{ and } T_{ae} \text{ in Kelvin}),
\]

and \( T_{sc} \) is the temperature of a clear sky, estimated from the relationship of Swinbank [17]:

\[
T_{sc} = 0.05532 \times (T_{ae} + 273,15)^{1.5} \quad (T_{sc} \text{ and } T_{ae} \text{ in Kelvin}).
\]

N is the coefficient of cloudiness that varies between \( N = 0 \) for a very clear sky, and \( N = 8 \) for a completely covered sky.

3. Results and Discussions

In order to evaluate the temperatures of the different elements of the dryer, the model described above takes into account the meteorological data collected during one year (2013). Only two climatically different days (a clear summer day and a covered winter day) were chosen for validation of the model.

3.1. Validation of the model

Among the temperatures estimated by the model, the temperature of the inside air of the dryer is the most important because its variation directly affects the temperatures of the other elements. As a result, its knowledge is of major interest in determining the performance of the dryer, whatever the climatic conditions prevail, and without the need to resort to measurements. The following figures 3(a) and 3(b) represent the variation profiles of the measured temperature \( T_{ai} \) (in bold lines) and the calculated one (in normal lines) during the two studied days (January 15, 2013 for the winter day, and August 15, 2013 for the summer day), respectively:

The comparison of variation profiles of the measured and calculated temperature \( T_{ai} \) shows a satisfactory agreement. For the two days studied, the estimation error does not exceed 1.5 °C in the middle of the day and increases slightly at the beginning and the end of the day, due to the relatively high inertia of the air inside the dryer, which slightly delays the response of the dryer to the rapid variation in ambient climate parameters.

![Figure 3: Measured and calculated variations of the inside air temperature of the solar dryer](image-url)
These profiles show that the amplitude of variation of the temperature Tai inside the dryer between day and night is higher in summer was around 30 °C (between maximum and minimum), compared with only to 10 °C in the winter. The offset from midday of recorded maxima is more than four hours in summer, compared to about three hours in winter.

Due to the IR thermal losses, which are important in winter as well as in summer, it is observed that:
- in the night period, the temperature Tan is reversed and becomes lower by approximately 3 to 4 °C than the ambient outside temperature Ta, the deployment of a thermal screen may prove useful and will make it possible to reduce, or even prevent this inversion of temperature;
- during the day, the temperature difference between the inside and the outside is much higher in summer (around 30 °C) than in winter (8 °C). It is therefore necessary to use heating, especially during the night, because it would keep a higher temperature.

3.2 Quantification of heating power
3.2.1 Heating systems
Heating is an important cost in the operating budget of a dryer, and then the choice of heating mode and the most economical system is necessary. Among the multitude of auxiliary heating used systems, especially in winter (gas boiler, heat pump, oil heating, electric heating, pellet stove and wood insert, etc.), the minimum power to be used is dependent on the surrounding climatic conditions and the volume of the air to be heated.

In order to validate the response of our model to the use of a heating system, we tested two heating systems, one of which is radiative, and the other is convective.

3.2.1.1. Radiant heating system
The tested radiative system consists of an electric heating plate with a radiating surface of 700 cm², inserted inside the dryer. The choice of the radiating surface of 700 cm² is conditioned by the standard dimensions of the commercially available heating plates. Even the choice of a larger area (equal to the surface of the dryer for example) remains insufficient to ensure the desired setpoint temperature. The heating plate with nominal temperature Tch = 160 °C and the elements of the dryer, in particular the cover whose mean temperature Tm is much less than Tch, are considered to act as gray bodies radiating one to the other, with emissivity ε1 and ε2, respectively.

The power radiated by the surface A1 of the heating plate, to the surface A2 of the dryer cover, is given by the equation [18]:

\[ Q_{12}^R = \frac{\sigma \varepsilon_1 \varepsilon_2 A_1 F_{12}}{1 - (1 - \varepsilon_1)(1 - \varepsilon_2)F_{12} F_{21}} (T_{ch}^4 - T_m^4) \]

Where F12 and F21 are the respective shape factors between the surfaces A1 and A2, and σ the Stefan-Boltzmann constant.

Since the product, (1-ε1).(1-ε2)F12.F21 is much less than 1, this equation can be simplified and becomes:

\[ Q_{12}^R = \sigma \varepsilon_1 \varepsilon_2 A_1 F_{12} (T_{ch}^4 - T_m^4) \]

Since the surface A1 of the heating plate is very small with regard to surface A2, the shape factors will be F12 ≈ 1 and F21 ≈ 0. Furthermore, considering that the radiating surfaces are black bodies, the emissivities ε1 and ε2 are equal to 1, and the power radiated by the heating plate is reduced to the expression:

\[ Q_{12}^R = A_1 h_{12} \cdot (T_{ch}^4 - T_m^4) \]

where the radiative coefficient h12 is given by the relation:

\[ h_{12} = \sigma \cdot (T_{ch} + T_m) \cdot (T_{ch}^2 + T_m^2) \]

For a maximum temperature of the heating plate Tch = 160 °C and a minimum winter average temperature of the cover Tm = 5 °C, the radiative coefficient h12 is equal to about 10.7 W.m⁻².K⁻¹. Under these conditions, the nominal power corresponding to the maximum power which would be radiated by the heating plate is:

\[ Q_{\text{max}} = 116 \text{ W}. \]

This power remains largely below the minimum value required to maintain a desired set temperature Ta during the heating period. Therefore, the use of the heating plate is not suitable for auxiliary heating, since in addition to the high energy consumption, necessary for its supply, the thermal requirements (as indicated by curves 4.a and 4.b below), will never be met, especially when the difference between the set point temperature Ta and the cover mean temperature Tm is assumed to be close to the outside temperature Ta, is greater than 2 °C.

3.2.1.2. Aero-convective heating system

The installation outside the dryer of a convective heating system, with modular power, made it possible to ensure a heat exchange by renewing air between the outside environment and the inside air of the dryer. The heat exchange is carried out by means of a double pipe, one of which is used for blowing and the other for recovery.

Its application for model validation took into account the technical characteristics of the system, which vary considerably from one system to another (including nominal power and blowing rate) and which cover a very wide range of available products in commerce.

For a heater convection heating system, the following relation calculates the supplementary heating power: to met

\[ Q_X = D \times \rho \times c \times \Delta \theta \]

where \( D \) is the volumetric air flow rate (in \( m^3.s^{-1} \)), \( \rho \) is its density (in \( kg.m^{-3} \)), \( c \) is its specific heat (in \( J.kg^{-1}.K^{-1} \)), and \( \Delta \theta \) is the temperature difference between the inlet and the outlet of the heating system, the value of which can reach 100 °C. In the operating temperature range of the dryer, the density \( \rho \) of the air and its specific heat \( c \) are assumed to be independent of its temperature.

Depending on the required set point temperature \( T_{set} \), the minimum power required for the dryer is always available by this type of heating system.

For the electric heater we tested, with a power of 9 kW = 7740 kcal/hour, maximum hot air flow rate \( D = 100 m^3/hour \) and a deviation \( \Delta \theta = 60 °C \) with a diameter of 100 mm supply/exhaust ducts, the heating power \( Q_X \) has reached 1200 W.

The cost price of the heating system remains a major disadvantage in the overall investment cost, but can be easily damped by means of a correct dimensioning of the heating requirements.

3.2.2. Simulation of the heating system

The introduction of heating power into the dryer is necessary in order to maintain its internal air temperature higher than or at least equal to a set temperature \( T_{set} \) according to the specificity and nature of the product to be dried.

From the climate data measured on 15/1/2013 and 15/8/2013, several simulations of the heating power \( Q_X \) (voluntarily introduced as a constant in equation 8) have been realized by successive increases of its value between 0 and Nominal power \( Q_N \), the value of which is linked to external climatic conditions. The chosen variation pitch is 100 W.

The following figures 4.a and 4.b show the variation profiles of \( Q_X \) as a function of the setpoint temperature for the both studied days:

![Figure 4](image_url)

Figure 4: Temporal variations of the heating power \( Q_X \) as a function of the setpoint temperature \( T_{set} \)

An analysis of these curves shows that:

- For the winter period, it will be seen that the heating time depends on the set temperature required, which can be reached quickly or not, depending on the ambient climatic conditions. Consequently, the heating time may be longer or shorter, it can even expanded throughout the day for certain temperatures (see figure 5). For the day of 15/01/13, for example, heating is required all day for set temperatures of about 19 °C or more.

- The increase of the set point temperature \( \Delta Q_X \) is accompanied by an increase in the heating power. This increase is greater in the nighttime period (\( \Delta Q_X = 55 W/°C \)) than in the daytime period (\( \Delta Q_X = 40 W/°C \)).
- For the summer period, night heating is only necessary for the highest set temperatures (for a set temperature \( T_{\text{set}} = 30 \, ^\circ\text{C} \), for example, heating must be provided between 19h PM and 8h AM).
- In the case of lower set temperatures, heating may only occur during the beginning of the day, where the inertia of the air inside the dryer plays a role in maintaining the relatively high internal temperature during the end of day period.
- The curves of figure 4, which show the evolution of the power heating according to the set temperature \( T_{\text{set}} \), are in a range between 16 and 30 °C. Outside this range, heating is not necessary for lower set temperatures (less than 16 °C). For high set temperatures (above 30 °C), many products to be dried are not very resistant to high temperatures, especially during the night.

![Figure 4: Variation of the power heating according to the set temperature \( T_{\text{set}} \).](image)

3.3. Evaluation of heating energy cost

For the assessment of the cost of heating energy, and excluding the initial investment cost, fluctuations and variability in fuel purchase prices, ease of installation of the heating system, necessity of connection to a power source, the need for storage, etc., two scenarios are possible:
- the heating power remains constant during the heating period, the resulting energy is the product of this power by the duration of operation;
- the heating power remains constant over successive short periods and its value changes from one period to another as a function of the desired setpoint temperature, the resulting energy in this case is the sum of the successive energies calculated for each interval.

Based on the estimated heating power variation profile over a day, as a function of the requested set point temperature, the curve (a) in Figure 6 below shows the variation of the daily heating energy calculated for the winter day studied. A quasi-linear increase is obtained as a function of the required temperature of the energy, which vary for a set temperature of 16 °C from 1.5 kWh.m\(^{-3}\).day\(^{-1}\), and reaches 8 kWh.m\(^{-3}\).day\(^{-1}\) for an extreme set temperature of 30 °C.

Curve (b) of Figure 6 shows the variation of this daily supplemental heating energy calculated for the summer day studied. It can be seen that this energy, which remains zero for set temperatures below 20 °C., increases slightly for set temperatures above 20 °C., and holds only 1.2 kWh.m\(^{-3}\).day\(^{-1}\) for the set point extreme temperature of 30 °C.

![Figure 6: Variations in daily heating energy for both studied days.](image)
Conclusions
The positive comparison between the theoretical model and the experimental results concerned the variation during two climatic different days of the air temperature inside the dryer. A maximum deviation of 1.5 °C was observed between the two temperatures (theoretical and experimental).

Deviation between outside and inside temperatures is very high in summer (30 °C) but lower in winter (10 °C). The difference obtained in the winter season is insufficient to initiate the drying process without the need for auxiliary heating.

The results of simulation of the thermal behavior of the dryer as a function of the heating power show that to increase the temperature of the air inside the dryer by 1 °C, an auxiliary heating of approximately 55 W overnight against 40 W during the day is necessary in winter. In summer, the need for auxiliary heating is only desired during the night and for very high set point temperatures.

The daily auxiliary heating energy increases linearly depending on the required set point temperature. In winter, the daily heating energy is 1.5 kWh.m⁻³.day⁻¹ for a temperature set point of 16 °C, reaching 8 kWh.m⁻³.day⁻¹ for an extreme temperature set point of 30 °C. In summer, this energy, which remains zero for set temperatures below 20 °C, increases slightly for set temperatures above 20 °C and requires only 1.2 kWh.m⁻³.day⁻¹ for the extreme temperature set point of 30 °C.

NOMENCLATURE

\[ c_{ai} \]: Mass heat of moist air inside the dryer (J.kg⁻¹.K⁻¹).
\[ c_{ao} \]: Mass heat of moist air outside the dryer (J.kg⁻¹.K⁻¹).
\[ c_{c} \]: Specific heat of concrete (J.kg⁻¹.K⁻¹).
\[ c_{g} \]: Specific heat of glass (J.kg⁻¹.K⁻¹).
\[ c_{pl} \]: Specific heat of Plexiglas (J.kg⁻¹.K⁻¹).
\[ c_{si} \]: Specific latent heat of vaporization of the water (J.kg⁻¹).
\[ D \]: Hourly air outflow (Volume/hour).
\[ e \]: Thickness of upper soil layer S₀(m).
\[ F_{ij} \]: Form factor between element i and element j.
\[ Gr \]: Grashof dimensionless number.
\[ h_{ai,mc} \]: Mass transfer coefficient between the Si facade and the inside air (kg.m⁻².s⁻¹).
\[ h_{ai,mc} \]: Mass transfer coefficient between the Si facade and the outside air (kg.m⁻².s⁻¹).
\[ h_{ai,sec} \]: Sensitive convective exchange coefficient between the Si facade and the inside air (W.m⁻².K⁻¹).
\[ h_{ai,sec} \]: Sensitive convective exchange coefficient between the Si facade and the outside air (W.m⁻².K⁻¹).
\[ h_{ai,sec} \]: Sensitive convective exchange coefficient between the S pl facade and the inside air (W.m⁻².K⁻¹).
\[ h_{ai,sec} \]: Sensitive convective exchange coefficient between the S pl facade and the outside air (W.m⁻².K⁻¹).
\[ h_{ai,sec} \]: Sensitive convective exchange coefficient between the upper soil surface S₂ and the outside air (W.m⁻².K⁻¹).
\[ HRe \]: Relative humidity of outside air (%).
\[ Nu \]: Nusselt dimensionless number.
\[ Pr \]: Prandtl dimensionless number.
\[ Q^C \]: Convective power exchanged between inside air ai and outside air ae(W).
\[ Q^G \]: Convective power exchanged between inside air ai and outside air ae(W).
\[ Q^S \]: Power exchanged by conduction between soil layers (W).
\[ Q_{ai,se} \]: Latent convective power exchanged between the dryer element i and the inside or outside air (W).
\[ Q_{ai,se} \]: Infrared radiative power exchanged between element i and the inside or outside air (W).
\[ Q_{ai,se} \]: Infrared radiative power exchanged between element i and elements j (W).
\[ Q^S \]: Solar power absorbed by element i (W).
\[ QX \]: Inside air heating power (W).
\[ Re \]: Reynolds dimensionless number.

\[ S \]: Surface of the lower soil layer (m²).
\[ So \]: Surface of the upper soil layer (m²).
\[ S_{1} \]: Surface of the south inclined glass (m²).
\[ S_{2} \]: Surface of the south vertical glass (m²).
\[ S_{3} \]: Surface of the east vertical glass (m²).
\[ S_{4} \]: Surface of the west vertical glass (m²).
\[ S_{g} \]: Global solar radiation (W.m⁻²).
\[ S_{pl} \]: Surface of the north wall (m²).
\[ St \]: Surface exchange between solid and inside air (m²).
\[ T_{i} \]: Temperature of body i(K).
\[ T_{p} \]: Temperature of the north wall Sₚ(°C).
\[ T_{se} \]: Temperature of the southern inclined glass Sₙ(°C).
\[ T_{s} \]: Temperature of the lower soil layer (°C).
\[ T_{t} \]: Temperature of the outside soil(K).
\[ T_{w} \]: Temperature of a cloudy sky (°C).
\[ T_{w₁} \]: Temperature of the upper soil layer (°C).
\[ T_{w₂} \]: Temperature of the south inclined glass S₟(°C).
\[ T_{w₃} \]: Temperature of the south vertical glass Sₜ(°C).
\[ T_{w₄} \]: Temperature of the east vertical glass Sₜ(°C).
\[ T_{w₅} \]: Temperature of the west vertical glass Sₜ(°C).
\[ T_{sp} \]: Deep Soil Temperature (°C).
\[ U \]: Wind speed of the outside air (m.s⁻¹).
\[ U_e \]: Wind speed of the outside air (m.s⁻¹).
\[ V_{ai} \]: Volume of the facade Sₐ of the dryer (m³).
\[ V_{pl} \]: Volume of the facade Sₚ of the dryer (m³).
\[ V_{s} \]: Volume of the lower soil layer of the dryer (m³).
\[ V_{so} \]: Volume of the upper soil layer of the dryer (m³).
\[ W_{ai} \]: Inside air vapor mass (kg/kg).
\[ W_{ai} \]: Saturated humidity at the temperature of facade Sₐ(kg/kg).
\[ W_{pl} \]: Saturated humidity at the temperature of facade Sₚ(kg/kg).
\[ W_{so} \]: Saturated humidity at the temperature of the upper soil surface Sₐ(kg/kg).
\[ ε_{i} \]: Emissivity of body i.
\[ λ_{s} \]: Thermal conductivity of the upper soil surface Sₐ (W.m⁻¹.K⁻¹).
\[ ρ_{c} \]: Density of concrete (kg.m⁻³).
\[ ρ_{g} \]: Density of glass (kg.m⁻³).
\[ ρ_{pl} \]: Density of Plexiglas (kg.m⁻³).
\[ ρ_{ai} \]: Density of the inside air (kg.m⁻³).
\[ σ \]: Constant of Stefan-Boltzman (W.m⁻².K⁻⁴).
ANNEX A

1) Exchanged thermal powers
Power of the convective heat flux exchanged between the facades Sand the outside air (the index i denotes the various glass sides of the dryer):

\[ Q_{vi,ae}^C = h_{vi} S_i (T_{vi} - T_{ae}) \]

Power of the convective heat flux exchanged between the facades Si and the inside air:

\[ Q_{vi,ai}^C = h_{vi,ai} S_i (T_{vi} - T_{ai}) \]

Power of the latent convective heat flux exchanged between the facades Si and the outside air:

\[ Q_{vi,ae}^L = C_{vi,ae} h_{vi,ae} S_i (W_{vi} - W_{ae}) \]

Power of the latent convective heat flux exchanged between the facades Si and the inside air:

\[ Q_{vi,ai}^L = C_{vi,ai} h_{vi,ai} S_i (W_{vi} - W_{ai}) \]

Convective heat flux due to air leakage:

\[ Q_{ai,ae}^C = D_p a_i \frac{V_i}{S_{ai}} (C_{ai}, T_{ai} - C_{ae}, T_{ae}) \]

Power exchange by conduction between two successive layers of soil:

\[ Q_{S_i,So}^G = \frac{\lambda_{So}}{e} S_i (T_{S_i} - T_{So}) \]

Power of the radiative flux exchanged between surface Si and surface Sj:

\[ Q_{vi,vj}^R = \frac{\sigma \epsilon_{vi} \epsilon_{vj} F_{vi,vj} S_i}{1 - (1 - \epsilon_{vi})(1 - \epsilon_{vj}) F_{vi,vj} F_{vj,vi}} (T_{vi}^4 - T_{vj}^4) \]

Power of the radiative flux exchanged between the surface Si and the sky:

\[ Q_{vi,sc}^R = \sigma \epsilon_{vi} \epsilon_{sc} F_{vi,sc} (T_{vi}^4 - T_{sc}^4) \]

Power of the radiative flux exchanged between the Si surface and the celestial vault:

\[ Q_{vi,vse}^R = \sigma \epsilon_{vi} \epsilon_{vse} F_{vse} (T_{vi}^4 - T_{vse}^4) \]

2) Convective exchange coefficients
The convective coefficient \( h_{c,ae} \) (W.m\(^{-2}\).°C\(^{-1}\)) between the solar dryer cover and the outside air is assumed to be related only to wind speed \( U_e \) (m.s\(^{-1}\)) according to McAdam's law [19-20]:

\[ h_{c,ae} = \begin{cases} 7.5 + 3.88 \times U_e & \text{if } U_e < U_{limite} = 7.72 \text{ (m.s\(^{-1}\))} \\ 7.3 \times (U_e)^{0.80} & \text{if } U_e > U_{limite} \end{cases} \]

The convective coefficient \( h_{c,ai} \) (W.m\(^{-2}\).°C\(^{-1}\)) between the inside air and the solar dryer cover is calculated from the relation:

\[ h_{c,ai} = \frac{Nu \lambda_{ai}}{L_{ci}} \]

The convective coefficient \( h_{c,ai} \) (W.m\(^{-2}\).°C\(^{-1}\)) between the inside air and the solar dryer cover is calculated from the relation:

\[ h_{so,ai} = \frac{Nu \lambda_{ai}}{L_{soil}} \]

\( \lambda_{ai} \) : the thermal conductivity of the air (W.m\(^{-1}\).K\(^{-1}\))

\( L_{ci} \) : the characteristic length of the coverage considered equal to the ratio between its surface and its perimeter (m).

\( L_{soil} \) : the characteristic soil length corresponding to the ratio of the dryer / floor area.

The dimensionless number of Nusselt Nu depends on the nature of the convection (natural or forced) and the mode of flow (laminar or turbulent) [21]:

\[ Nu = f (Gr, Re) \]

The boundaries between these different modes are defined by the dimensionless numbers of Grashof (Gr), Reynolds (Re) and Prandtl (Pr).

The limit between laminar and turbulent regimes is defined by the product \( L_{ci}^3 \times \Delta T \).
Table 1: Geometric and physical parameters of the solar dryer cover

<table>
<thead>
<tr>
<th>Settings</th>
<th>Materials</th>
<th>surfaces (m²)</th>
<th>Orientation</th>
<th>Tilt</th>
<th>Density ρ (kg/m³)</th>
<th>Thermal conductivity λ (W/m.K)</th>
<th>Massic heat Cm (J/kg.K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Façade (S1)</td>
<td>Glass Of 6mm</td>
<td>2,18</td>
<td>South</td>
<td>34</td>
<td>2500</td>
<td>0,8</td>
<td>840</td>
</tr>
<tr>
<td>Façade (S2)</td>
<td></td>
<td>1,5</td>
<td>South</td>
<td>90</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Façade (S3)</td>
<td></td>
<td>1,71</td>
<td>East</td>
<td>90</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Façade (S4)</td>
<td></td>
<td>1,71</td>
<td>west</td>
<td>90</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Façade (S5)</td>
<td>Plexiglas Of 10 mm</td>
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<td>North</td>
<td>90</td>
<td>1170</td>
<td>0,7</td>
<td>1380</td>
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<tr>
<td>Soil (S)</td>
<td>Concrete of 15cm</td>
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<td>-</td>
<td>-</td>
<td>2200</td>
<td>1,8</td>
<td>1000</td>
</tr>
</tbody>
</table>

References
