Experimental study and analysis of porous thin plate drying in a convection dryer☆

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1. Introduction

Drying represents one of the most important and energy-intensive processes in thermal engineering applications which involve simultaneous heat and mass transfer [1–5]. In some cases the simultaneous heat and mass transport of the moisture may even couple with the mechanical behavior of the material to be dried, which will affect the quality of the drying products [6–8]. Therefore, a well-designed system and the proper operating conditions are essential for attaining high quality drying [9–13]. To control the drying process, the physics of the transient moisture transport phenomena during the drying process needs be understood. In this study, experiments and an analysis were carried out to determine the moisture transfer coefficients that vary with the moisture content of the plate. The analysis was also compared with the experimental and analytical results from reported studies.

In addition to the requirements of fast drying and good temperature control, the energy efficiency of a drying system is another critical consideration. A hybrid drying system might well utilize the energy coming from the outlet of the dryer to make the system more efficient and cost-effective. For example, the hot and humid air at the outlet of a dryer can be used to maintain the desired temperature and humidity of a small space, and then used to drive a turbine to generate electricity.

Exergetic analysis appears to be an ideal tool for evaluating the available energy at the outlet of the system. Applications of exergetic analysis for solar thermal energy conversion systems have included solar desalination systems [14], solar-powered fuel cell micropower plants [15], solar refrigeration systems [16] and solar air heaters [17]. For drying processes, some studies have been reported on energy and exergetic efficiency analyses [2,18–22]. These investigations encompass analyses of a variety of dryers such as those using direct sunlight drying, natural and forced convection dryers, and mixed-mode (drying chamber and solar heat collector) dryers. One potential application of the dryer in this study is for the emerging high value agricultural products drying. In most cases, agricultural products such as fruits, vegetables, and wood, are made to thin plates to allow more efficient drying [2,7,18,23]. Therefore, we used porous silicon plates to simulate the drying of agricultural products, where the porosity, size, and initial moisture content can be well controlled. Energy and exergetic efficiency analyses were carried out to investigate the effects of drying conditions on the dryer efficiency and the available energy at the dryer outlet.

2. Experimental

Fig. 1 schematically shows the experiment apparatus. The system includes a dryer, an electric fan, electric heater, electrical controllers, thermo-hydrometers, and thermocouples with data logger, an electronic balance, and a hot-wire anemometer. The bottom of the chamber which is made of Bakelite (width × length × thickness = 95 ×...
95 × 32 mm) with a hole for inserting air inlet tube in the center. The chamber side-walls and the lid are made of 23 mm thick acrylic plate. A 10 mm diameter hole in the center of the lid is connected to the outlet with an electric exhaust fan on the top. A stainless steel mesh is placed in the center of the chamber to support the porous plate to be dried. To simulate a stable heating source, a controllable electrical heater set at 100 °C was used to heat the air at the inlet of the dryer. In practice, the heater can be replaced by concentrated solar light. A thin porous silicon plate (Sillon Sink CYS-516-A) as shown in the inset of Fig. 1 was used for the drying study. The original mass of the porous plate was 14.20 g, where the size is 56.4 × 56.6 × 2.5 mm, and the porosity is 29%. A single plate was used in order to eliminate the complexity caused by stacking plates. However, the dryer can be readily used to dry multiple plates at a time. K-type thermocouples (temperature range: −270 to 1260 °C, accuracy: ±2.2 °C) were used to measure the air temperature, ±0.1% RH for relative humidity. A lab-made controller was used to control the fan on the top and thus the air flow rate through the chamber, which was measured by a hot-wire anemometer (Kanomax KA31, accuracy: ±2%). An electronic balance (model UW6200HV, Shimadzu Corporation®, accuracy: ±0.02 g) was used to measure the mass changes of the plate. The plate was fully saturated with distilled water before the drying experiment was started. The experiments were conducted under both natural and forced convection conditions. For natural convection (Cases A and E), the fan was not used. For forced convection (Cases B, C, D and F), the fan was used to maintain stable air flow rates as shown in Table 1. The heater was also turned on and maintained the required temperature. When the measured temperature and air flow rate of the drying system became stable, the porous silicon plate was placed inside the dryer and the drying experiment was initiated. These experiments were conducted in the laboratory where the ambient temperature was between 23 and 25 °C and the relative humidity was between 30 and 53%. Each experiment ended when the moisture in the plate dropped below 10% of the initial moisture content.

3. Analysis

3.1. Moisture transfer analysis

The study of drying process involves coupled heat and mass transfer analyses that were usually solved using numerical methods [1,3–6,24, 25], where the mass and heat transfer coefficients can be determined from experimental results [26–29]. Dincer and Dost [28] carried out an analytical study and proposed a method for determining the effective moisture diffusion coefficient and the moisture transfer coefficients of a porous thin plate from referenced experimental results of drying rate. Akpinar and Dincer [29] later also utilized the analysis from Dincer and Dost [28] and compared the analytical results with other experimental studies. In the current effort we carried out a different analysis which allows the mass transfer coefficients to vary with the average moisture content of the plate. Consider drying of a thin porous plate as shown in Fig. 2, from mass conservation and assuming that the primary transport is one-dimensional [29,30], the governing equation is given by:

$$
\frac{\partial}{\partial x} \left( D_{\text{eff}} \frac{\partial m}{\partial x} \right) = \frac{\partial m}{\partial t}.
$$

(1)

Note that the effective moisture diffusion coefficient ($D_{\text{eff}}$) is used to include the effects of the porous structure of the plate on mass transfer [31]. In general $D_{\text{eff}}$ varies with respect to the moisture concentration in the plate ($m(x,t)$) which is related to the different periods of moisture transfer in a drying process as shown in Fig. 3 [32]. It is noted that it is possible to convert Eq. (1) by using the Kirchhoff transformation as has been done for heat conduction problems where the thermal conductivity varies with temperature [33,34], but difficulties still remain for the problem of interest. The governing equations and boundary conditions can be solved numerically for specific cases but we seek a more general methodology. We note that Crank [30] has utilized a time-dependent diffusion coefficient assumption and we follow this approach. To provide a useful approach, we assume that $D_{\text{eff}}$ and $h_m$ are functions of time and vary so that $h_m/D_{\text{eff}}$ is constant. Then making the transformations $\xi = x/L$ and $\tau = (D_{\text{eff}}/L^2)t$, the following equations are obtained:

$$
\frac{\partial^2 \phi}{\partial \xi^2} = L^2 \frac{\partial \phi}{\partial \tau} - \frac{\partial \phi}{\partial \tau},
$$

(2)

where $\phi = \frac{m - m_i}{m_s - m_i}$. The boundary conditions and initial condition are given as:

$$
\left. \frac{\partial \phi}{\partial \xi} \right|_{\xi = 1} = B_{\text{im}} \phi(1, \tau)
$$

(3)

$$
\left. \frac{\partial \phi}{\partial \xi} \right|_{\xi = 0} = 0
$$

(4)
where \( \tau(0) = \int_0^t \frac{D_{\text{eff}}}{L^2} \, dt = 0 \).

The zero mass flux boundary condition is applied at the upper surface \( (\xi = 0) \) where the mass flux is much smaller than that at the bottom surface \( (\xi = 1) \). Note that \( \tau \) is equivalent to the Fourier number \( [35] \). The transformed Eqs. (2) to (5) are now solved by the method of separation of variables \([34]\). The solution for \( \phi \) is given by:

\[
\phi = \sum_{n=1}^{\infty} C_n \exp \left[ -\lambda_n^2 \tau \right] \cos(\lambda_n \xi), \quad \text{where} \quad (\lambda_n) \tan(\lambda_n) = \text{Bi}_m, \quad n = 1, 2, 3, \ldots \infty.
\]

Introducing the orthogonality property of the eigenfunctions and applying the initial condition it yields:

\[
C_n = \frac{4 \sin(\lambda_1)}{2 \lambda_1 + \sin(2 \lambda_1)} \exp \left[ -\lambda_1^2 \tau \right] \cos(\lambda_1 \xi) = \frac{4 \sin(\lambda_1)}{2 \lambda_1 + \sin(2 \lambda_1)}.
\]

As noted by Akpinar and Dincer \([29]\), for values of \( \tau > 0.2 \), it is a good approximation to take the first term of the infinite series solution of Eq. (6) \([36]\), i.e.:

\[
\phi \approx C_1 \exp \left[ -\lambda_1^2 \tau \right] \cos(\lambda_1 \xi)
\]

where

\[
C_1 = \frac{4 \sin(\lambda_1)}{2 \lambda_1 + \sin(2 \lambda_1)}
\]

and

\[
\lambda_1 \tan(\lambda_1) = \text{Bi}_m.
\]

Later the first term approximation will be shown to be valid when the average moisture content of the plate is less than approximately 82% of its original value.

### 3.2. Thermal energy and exergetic efficiency analyses

In the present analysis, the drying process is assumed to be a steady flow process. The energy conservation equation is given by \([2,37,38]\):

\[
\dot{Q} - W = \dot{m}_{a,i} \left( h_{a,0} + \frac{V_e^2}{2} \right) - \dot{m}_{a,i} \left( h_{a,i} + \frac{V_e^2}{2} \right)
\]

where \( \dot{m}_{a,i} \) and \( \dot{m}_{a,o} \) are the inlet and outlet mass flow rates of drying air, respectively. The transient variation of the energy associated with the variation of mass and temperature of the moisture inside the dryer was assumed insignificant. Air entering the dryer is composed of dry...
The specific humidity \( w \) is defined as the water vapor mass \( (m_w) \) per unit mass of drying air \( (m_a) \). The energy provided by the heater to the dryer \( (Q_{d,i}) \) can be determined from the following equation:

\[
Q_{d,i} = m_{a,i}C_{pa}(T_{a,i} - T_{amb})
\]

where \( C_{pa} \) is the specific heat of drying air at the dryer inlet and can be determined by:

\[
C_{pa} = C_{pa, dry} + w_{i}C_{w}.
\]

It is assumed that only the energy being supplied to and absorbed by the air flow is dominant and the energy loss to the environment from the well-insulated heater is negligible. From Eq. (11), assuming there is no net work done by the dryer and the velocity differences of the air flow between inlet and outlet of the dryer is negligible, the energy utilization \( (Q_{d,util}) \) for drying the plate is calculated as follows [2,37,38]:

\[
Q_{d,util} = m_{a,i}(h_{a,i} - h_{a,o}).
\]

The energy utilization ratio (EUR) is then defined as [2,37,38]:

\[
EUR = \frac{m_{a,i}(h_{a,i} - h_{a,o})}{m_{a,i}C_{pa}(T_{a,i} - T_{amb})}.
\]

### 3.3. Exergetic analysis

The exergy rates corresponding to the inlet and outlet of the dryer are [20,37,38]:

\[
\dot{E}_{d,i} = m_{a,i}C_{pa} \left( T_{a,i} - T_{amb} \right) - T_{amb} \ln \left( \frac{T_{a,i}}{T_{amb}} \right)
\]

\[
\dot{E}_{d,o} = m_{a,o}C_{pa} \left( T_{a,o} - T_{amb} \right) - T_{amb} \ln \left( \frac{T_{a,o}}{T_{amb}} \right).
\]

The exergy efficiency is defined as the ratio of exergy outflow to the exergy inflow for the dryer [37]:

\[
e = 1 - \frac{\dot{E}_{dest}}{\dot{E}_{d,i}}.
\]

where \( \dot{E}_{dest} = \dot{E}_{d,i} - \dot{E}_{d,o} \) denotes the exergy destruction. The exergetic efficiency represents the magnitude of irreversibility that occurs during the drying process. A high value of exergetic efficiency means less exergy loss in the dryer. This information is important for the design and optimization of drying systems where the available energy at the outlet of the dryer is considered for further utilization. In order to clearly indicate the dryer performance, the time averaged EUR (EURavg) and time averaged exergetic efficiency (\( \varepsilon_{avg} \)) are defined as:

\[
EUR_{avg} = \frac{\int_{0}^{\tau_{drying}} (EUR)dt}{\tau_{drying}}
\]

\[
\varepsilon_{avg} = \frac{\int_{0}^{\tau_{drying}} \dot{e}dt}{\tau_{drying}}.
\]

These overall performance parameters, as shown in the following discussion, provide important information in a succinct and clear format and enable to make decisions for efficient drying mode and dryer design.

### 4. Results and discussion

#### 4.1. The analysis of moisture transfer coefficients

Experimental conditions for drying of the porous thin plate are listed in Table 1. To demonstrate the use of Eqs. (8)–(10) to determine \( D_{eff} \) and \( h_{in} \), the following example is now provided. The dimensionless average moisture content in the plate \( (\phi_{avg}) \) is defined as:

\[
\phi_{avg} = \frac{m_{avg}(t) - m_{e}}{m_{avg}(0) - m_{e}}
\]

where \( m_{avg}(t) \) is the average moisture concentration of the plate at time \( t \) defined as:

\[
m_{avg}(t) = \frac{1}{V} \left( \int_{V} m(x, t)dV \right)
\]

where \( V \) is the volume of the plate.

The experimental result for \( \phi_{avg} \) versus time for Case A is shown in Fig. 4. The curve that is fitted to the experimental data is given by:

\[
\phi_{avg} = \exp \left(-A t^{3/2} \right).
\]

where \( A = 1.58 \times 10^{-7} \, [s^{-3/2}] \).

From the definition of \( m_{avg} \) and utilizing the relations of \( \phi \) and \( \phi_{avg} \) yields:

\[
\phi_{avg} = C_1 \exp \left(-\lambda_1 \tau^2 \right) \int_{0}^{1} \cos(\lambda_1 \xi) d\xi
\]

\[
= \left[ \frac{4 \sin(\lambda_1)}{2\lambda_1 + \sin(2\lambda_1)} \right] \left[ \frac{\sin(\lambda_1)}{\lambda_1} \right] \exp \left(-\lambda_1 \tau^2 \right).
\]

Comparing Eq. (24) with Eq. (25), it is seen that:

\[
\left[ \frac{4 \sin(\lambda_1)}{2\lambda_1 + \sin(2\lambda_1)} \right] \left[ \frac{\sin(\lambda_1)}{\lambda_1} \right] = 1
\]
\[
\lambda_1^2 \tau = A t^{3/2}.
\]

(27)

Eq. (26) yields \(\lambda_1 = 0.995\). Note that this value of \(\lambda_1\) is the same constant for all of the drying cases discussed in this study. From Eq. (10), \(B_{\text{im}} = \lambda_1 \tan(\lambda_1) = 1.53\); thus, \(B_{\text{im}}/D_{\text{eff}} = 1.53/L = 612 \text{[m}^{-1}]\) (for \(L = 2.5 \times 10^{-5}\text{m}\) in this experiment). From the definition of \(\tau\):

\[
\frac{\tau}{t^{3/2}} = \frac{\int_0^t D_{\text{eff}} dt^+}{L^2 t^{1/2}} = \frac{A}{\lambda_1^2}.
\]

(28)

Eq. (28) is satisfied with the following assumption for \(D_{\text{eff}}\):

\[
D_{\text{eff}} = B t^{1/2}
\]

(29)

so that Eq. (29) yields:

\[
\int_0^t B t^{1/2} dt^+ \left( \frac{2B}{3L^2} \right) = \frac{A}{\lambda_1^2 t^{1/2}}.
\]

(30)

Thus,

\[
B = \left( \frac{3L^2}{2} \right) \left( \frac{A}{\lambda_1^2} \right)
\]

(31)

and

\[
D_{\text{eff}} = \left( \frac{3L^2}{2} \right) \left( \frac{A}{\lambda_1^2} \right)t^{1/2}.
\]

(32)

From Eqs. (24) and (29), it is shown that:

\[
\phi_{\text{avg}} = \exp(-At^{3/2}) = \exp \left[ -\left( \frac{A}{B^3} \right) (B t^{1/2})^3 \right] = \exp \left[ -\left( \frac{A}{B^3} \right) D_{\text{eff}}^3 \right]
\]

(33)

\[
D_{\text{eff}} = -\frac{B^3\ln(\phi_{\text{avg}})}{A}^{1/3}.
\]

(34)

Results for the variations of \(D_{\text{eff}}\) with \(\phi_{\text{avg}}\) for Case A are shown in Fig. 5. It is noted, again, that \(h_m\) can be obtained from \(h_m/D_{\text{eff}} = 1.53/L\) with the specific value for \(L\). Values of \(h_m\) for Case A varied from \(1.98 \times 10^{-7} \text{[m/s]}\) to \(1.02 \times 10^{-7} \text{[m/s]}\) over the range of \(\phi_{\text{avg}}\) from 0.2 to 0.8. It is seen that both \(D_{\text{eff}}\) and thus \(h_m\) increase as \(\phi_{\text{avg}}\) decreases. This is related to the different mass transfer characteristic at different periods during the drying process as shown in Figs. 3 and 4. In the beginning of the drying process, most of the heat is being used to increase the temperature of the plate and there is little evaporation. As the temperature increases, significant evaporation takes place, and the moisture transfer coefficients increase with time. As the moisture content in the plate decreases, more pores in the interior of the thin plate, which were previously blocked by water, become available. Therefore, more pores permit moisture to be transferred and vaporized, and thus the transport and the effective moisture transfer coefficients increase. It is noted that the increasing \(D_{\text{eff}}\) with decreasing \(\phi_{\text{avg}}\) was also reported in other porous media drying studies [39–44]. Note that the simplification of \(\phi\), i.e. Eq. (8) is valid as long as the Fourier number (\(F_o = \tau\)) is greater than 0.2 [28,36]. From Eqs. (25) and (26), for \(\tau = 0.2\) the dimensionless average moisture content \(\phi_{\text{avg}}\) is approximately 0.82. Therefore, the first term approximation solution is valid when the average moisture content of the plate is less than 82% of its original value. The comparison of the assumption of \(\phi_{\text{avg}} = \exp(-At^{3/2})\) with the experimental and theoretical results for thin plate drying [35,45] is shown in Fig. 6. Note that the same experimental data of Sawhney et al. [45] were also used by Dincer and Hussain [35] to validate their theoretical results. Utilizing Eqs. (31) and (35) with \(A = 1.72 \times 10^{-6} \text{[s}^{-3/2}]\) and \(L = 2.5 \times 10^{-3}\text{[m]}\) [45] gives the variation of \(D_{\text{eff}}\) with \(\phi_{\text{avg}}\) for the experimental results by Sawhney et al. [45] as shown in Fig. 7. Values of \(h_m\) calculated from Eq. (35) varied from \(9.75 \times 10^{-7}\) [m/s] to \(5.05 \times 10^{-7}\) [m/s] over the range of \(\phi_{\text{avg}}\) from 0.2 to 0.8. It is noted that the values of \(D_{\text{eff}}\) and \(h_m\) for the primary drying period are close to those attained by Dincer and Hussain [35] where the value of \(B_{\text{im}}\) given by Dincer and Hussain is 2.42 [35], and the value of \(B_{\text{im}}\) from Eq. (10) is 1.53.

4.2. The analysis of thermal energy efficiency

Figs. 8 and 9 show the variation of the inlet and outlet temperatures during the drying process. Each experiment was repeated five times, and the data averaged for each case with the error bar denoting the
standard deviation of each set of data are presented to keep the figures clear. The deviations in the measured air temperature result primarily from ambient changes. The air at ambient temperature flows into the heater at the inlet of the drying chamber where the temperature of the heater was fixed at 100 °C. As shown in Fig. 8, in most cases the inlet air temperature is higher with lower air flow rates because more time is needed for air flow to be heated, except for Case A where the inlet air temperature is lower than that for Case B. This is due to the natural convection where the cold air inside the chamber flows towards the inlet opening and mixes with the warm air from the inlet. From Fig. 9, it is seen that for the same inlet tube diameter, the air temperature at the outlet of the drying chamber increases with increasing air flow as a result of effective transport of thermal energy by the strong convective flow. In the forced convection mode, the outlet temperature in each case decreases rapidly at the beginning of the experiment, and remains steady except towards the end of the drying process. During the primary drying period, the thermal energy being utilized is approximately constant since the drying rate is steady. In the natural convection mode, the outlet temperature first decreases as the thermal energy of the drying air is partially utilized to heat the plate and the chamber to resume a stable temperature distribution inside the chamber, and then remains steady as in the forced convection mode. The second drop (near 400 min) in the outlet air temperature for Case A is due to the re-heating and evaporation of moisture that was condensed near the outlet of the chamber as was observed during the experiment. Towards the end of the drying process, the outlet temperature increases in all cases because the drying rate decreases and less energy is utilized in the dryer.

The changes in \( \phi_{\text{avg}} \) of the plate during the drying process are presented in Fig. 10. In all cases, the initial mass of the wet plate is 17.80 g with 25.3% humidity. The humidity of the plate is defined as the weight of water in the plate divided by the weight of the dry plate. Results of the drying time are summarized in Table 2. It is seen that the drying rate increases with increasing air flow because this promotes both heat and mass transfer in the dryer. Comparing Cases C and F, it is seen that for the same drying air flow rate, the drying rate increases with increasing the inlet tube size, even though the inlet air temperature for the bigger tube is lower. Increasing the size of the air inlet increases the area on the bottom surface of the plate exposed to the air flow. Thus, for this configuration, the moisture transfer is dominated by the convective flow rather than the thermal effects of moisture evaporation. Fig. 11 shows the variation of the energy utilization ratio (EUR) for all cases. At the beginning of the drying process, some of the inlet thermal energy heats the plate and vaporizes the moisture in the plate; thus the EUR increases. After the temperature of the plate and the air inside the dryer become stable, the inlet hot air is used for vaporizing the moisture in the plate. As previously discussed, the drying rate...
is approximately constant during the primary drying period in each case, and thus the energy utilized for vaporization and the EUR remain approximately steady. The EUR decreases towards the end of the drying process because there is less moisture to be removed from the plate and so less energy is used during the completion period. As shown in Table 2, the time averaged EUR decreases with increasing air flow rate or increasing inlet tube size. The highest value of EUR is obtained in the natural convection mode with a small air inlet tube. Based on the results shown in Table 2, without considering, for now, the time required for drying, the natural convection mode yields more efficient energy utilization than the forced convection mode. A comparison of those cases with forced convection also confirms that the energy utilization efficiency is improved by decreasing the air flow. From the experimental results, it is also seen that the air temperature at the outlet in the forced convection mode is significantly higher than the ambient temperature (cf. Fig. 9), which means that a significant amount of the input thermal energy is still available for use. For example, a thermoelectric device or a turbine can be connected to the dryer outlet to further convert and utilize the energy carried by the air flow. This concept would be useful in a solar dryer where the excessive solar thermal energy being collected by the dryer can be converted to other types of energy to be used.

The amount of energy available for further use is investigated by conducting the exergetic analysis of the dryer. Fig. 12 presents the variation of exergetic efficiency with respect to time. The exergetic efficiency is high at the beginning of the experiment since the outlet air temperature is high (cf. Fig. 10). The exergetic efficiency then decreases rapidly, which indicates less energy is available at the outlet of the dryer because the energy is indeed being used to heat the plate and for evaporation and for heating the dryer during the early stage of the drying process. After the early stage, the exergy destruction and thus the exergetic efficiency remain approximately constant since the water evaporation rate and thus the available energy being utilized for drying are steady as discussed previously. Near the end of the drying process, energy consumption for vaporizing water in the plate decreases as its moisture content drops. Consequently, more energy becomes available at the outlet; i.e. the exergy outflow increases and thus the exergetic efficiency increases. A comparison of Figs. 11 and 12 also shows that the exergetic efficiency variation is inversely related to the energy utilization ratio. The time averaged exergetic efficiencies ($\varepsilon_{avg}$) for all cases are shown in Table 2. In general, $\varepsilon_{avg}$ for natural convection mode (Cases A and E) is lower than that for the forced convection mode. Nevertheless, for the dryer with a small inlet tube (Cases A–D) the magnitude and the differences of the exergetic efficiencies are small. Thus, no significant amount of energy is available at the outlet of the dryer for further utilization. Comparing the forced convection cases with the same air flow rate (Cases C and F), $\varepsilon_{avg}$ is higher for the dryer with the bigger inlet tube (i.e. Case F). The drying rate in Case F is higher than that in Case C. In addition, the inlet air temperature in Case F is lower than that in Case C, so that the total input thermal energy for Case F is less than that in Case C since the air flow rates are the same for both cases. Therefore, this result demonstrates that in the evaluation of the overall performance of the dryer, a forced convection dryer with a bigger air inlet tube should be used so that the available energy at the outlet is utilized.

5. Conclusions

The drying process of a porous thin plate using a convection dryer was studied both experimentally and analytically. Transport analysis of the porous plate drying was carried out to determine the moisture content dependent transfer coefficients ($D_{eff}$ and $h_m$) in the present experiments. The same analysis was also used to obtain the moisture

<table>
<thead>
<tr>
<th>Case</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Time averaged energy utilization ratio ($\varepsilon_{avg}$)%</td>
<td>93.0</td>
<td>92.9</td>
<td>91.4</td>
<td>89.3</td>
<td>78.1</td>
<td>69.2</td>
</tr>
<tr>
<td>Time averaged exergetic efficiency ($\varepsilon_{avg}$)</td>
<td>0.5</td>
<td>0.6</td>
<td>0.9</td>
<td>1.3</td>
<td>5.3</td>
<td>10.2</td>
</tr>
<tr>
<td>Drying time for $\phi_{avg}$ reaches 10% (minutes)</td>
<td>890</td>
<td>470</td>
<td>275</td>
<td>200</td>
<td>120</td>
<td>95</td>
</tr>
</tbody>
</table>
content dependent $D_{a}$ and $h_{m}$ and compared with the values of $D_{a}$ and $h_{m}$ from other studies. The variation of $D_{a}$ and $h_{m}$ on the average moisture content ($\phi_{aw}$), especially in the early and the late stages of the drying process, obtained in the current study can be used to analyze and understand the variation of moisture transfer characteristics with $\phi_{aw}$ of thin porous plate drying.

Thermal energy and exergetic efficiency analyses were also carried out to investigate the performance of the dryer under a variety of inlet flow configurations (i.e. changing the inlet tube diameter and the flow rate in this study). The thermal energy utilization ratio (EUR) in the forced convection mode is generally lower than that of natural convection. In the forced convection mode, the EUR decreases with either increasing the air flow rate or increasing the inlet tube diameter. However, the drying time can be significantly reduced. In addition, while the EUR is decreased, exergetic analyses results indicated that increasing the air flow rate or the inlet tube diameter significantly improves the exergetic efficiency (from 0.5% to 10%). Consequently, a significant amount of available energy at the outlet could be utilized in the dryer design. This study completed an overall transport phenomena analysis and performance evaluation including the transient moisture transfer coefficients, drying rate, energy utilization ratio and exergetic efficiency which provide comprehensive and important insights for the thermal physics of porous thin plate drying processes and the design of solar thermal hybrid (e.g., drying, thermoelectric, turbine, heating and humidifying) systems.

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References