Performance Enhancement of a Sludge Continuous Feed Heat Pump Drying System by Air Deflectors and Auxiliary Cooling Subsystems

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Abstract: This study proposes an improved design for a typical sludge continuous feeding drying system connected with three air-source heat pumps. The system’s performance was further improved using air-deflectors on the drying chamber’s internal sidewalls, enhancing the heat and mass transfer between the conveyor sludge and circulating airflow. In this study, numerical analysis was performed to elucidate the deflector designs on the airflow field and thermal temperature field distributions in the drying chamber. The specific moisture extraction rate (SMER) value was quantified to evaluate the system’s overall improvement during experiments. With a suitable deflector design, the average percent water content in sludge could be further reduced to 22.2% with drying time of 18.3 h, and the SMER value could be enhanced from 1.38 kg/kWh to as high as 1.83 kg/kWh with an increment of 32.44%. Moreover, to prevent overloading and frequent shutdown of the compressors, an auxiliary cooling subsystem was designed to attain stable operational conditions. By the auxiliary cooling subsystem design, the compressors’ shutdown can be avoided, the temperature difference between airflow inlets and outlet of the drying chamber can be increased, and SMER value can be further increased to a value of 1.94 kg/kWh.

Keywords: sludge; heat pump; auxiliary cooling; specific moisture extraction rate

1. Introduction

Approximately 2.4 million tons of sludge are being produced in Taiwan annually. With rapid industrial development, the total amount of sludge that needs treatment is increasing subsequently. In 2011 the processing fee for one metric ton of sludge was about 64–95 USD in Taiwan and it has risen to 222–254 USD by 2019 [1]. Traditional sludge drying processes use diesel, oil, or natural gas which are burned to generate high temperature steam for sludge dewatering. However, several disadvantages of using these combustion fuels have to be considered, including their low burning efficiency, high operational cost and carbon dioxide emissions. To address this problem, meet energy-saving goals and reduce carbon emissions, heat pump drying systems with high operating efficiency, low operational cost and low environmental impact have been gradually adopted [2–5].

Air-source heat pump systems are often adopted for food drying. In the drying process, the high temperature and low humidity circulating air generated by the heat pump system is used to reduce the water content in foods. The water vapor in the circulating air is condensed into liquid water in the heat pump’s evaporator during the drying process to remove the foods’ water. To dry...
mint leaves, for example, Aktas et al. [6] proposed a heat pump system integrated with a heat recovery unit and a cylindrical drying chamber. The highest recovered energy was obtained at an air velocity of 2 m/s at 35 °C. In the best case, the coefficient of performance (COP) of the drying system was 3.94, the corresponding power consumption was 3.164 kWh and the heat recovery unit could recover nearly 48% of consuming power. Hii et al. [7] investigated the drying kinetics of individual layers of cocoa beans at dehumidified air temperatures of 28.2 °C, 40.4 °C and 56 °C, respectively. Product quality analysis suggested that the percentage of retention of cocoa polyphenols was improved from 44% to 73%, a 29% improvement. The hardness of drying cocoa beans was found to be similar to that of the commercial products and the hardness increased as the water content decreased. The effectiveness of continuous drying and intermittent drying by the heat pump system was investigated by Zhu et al. [8]. They analyzed seven intermittent drying modes to carry out the drying experiments on raw soybeans and it was observed that the increase in the intermittence ratio could reduce the drying time and increase the specific moisture extraction rate (SMER). The performance of solar dryer and solar-assisted heat pump dryers for drying cassava chips were analyzed by Yahya et al. [9]. They found that the average thermal efficiencies of the solar dryer and the solar-assisted heat pump dryer were 25.6% and 30.9%, respectively. The solar-assisted heat pump dryer thus has better overall efficiency than the solar dryer.

Considering sludge substances, the drying efficiency of the heat pump drying system has been enhanced by adding mixed substances in the sludge. Zhang et al. [10] compared the drying rate of a single sludge and a mixture of lignite and sludge at 60 °C to 180 °C drying temperature and 1.5 m/s wind speed. When the drying temperature is more than 120 °C in the first falling rate period, the average drying rate of lignite and sludge mixture is about 2.6% to 11.4% and is higher than that of the single sludge. In the second falling drying rate, the average drying rate of lignite and sludge mixture at 60 °C to 180 °C was about 2.9% to 33.4%, respectively, higher than that of the single sludge.

Li et al. [11] investigated the drying efficiency of the sludge adhering to the drying system’s heating surface during the drying process. They found that moisture is a crucial factor affecting the sludge adhesion during the drying process. The mass of sludge adhesion to the heating surface increases as the water content of the sludge decreases. When the water content is about 60%, the sludge’s adhesion amount reaches a maximum value. Moreover, higher drying temperature may cause more sludge adhesion on the heating surface in the drying temperature range of 80 °C to 160 °C.

Ploteau et al. [12] proposed a mathematical model of sludge drying to calculate the SMER value and the performance of the continuous drying process, in which a heat pump model and a circulating air stream are coupled with intermediate exchangers. From the experimental results, air entering the evaporator with higher humidity can consume less compressor power. Li et al. [13] investigated the effect of sludge and sawdust mixture on the convective drying process and studied the effect of sawdust’s different mass percentages (10%, 20%, 30%, 40%) towards the net drying efficiency. The initial porosity, volume, and total surface area were found to be related to the percentage of sawdust. The results indicate that sawdust’s addition can improve sludge drying efficiency due to increased initial heat exchange surface.

Numerical analysis methods are adopted to analyze and design a sludge drying system to reduce the drying system design cost [14]. Janetti et al. [15] used Calcium silicate as the building material for experiments under normal atmospheric conditions (temperature 25 °C and 60% relative humidity), and an infrared camera measured the surface temperature of the building material. The analysis highlighted that the drying rate was related to the wet-bulb temperature of the surrounding. Lower wet bulb temperature during the drying process leads to a faster drying rate. Purlis et al. [16] proposed the theoretical and numerical analysis of the multiphase porous medium model to establish a heat and mass transfer for low-strength and low-temperature convective drying with hygroscopic porous materials.

Among the thermal drying methods, the traditional drying method by electric heating has low thermal efficiency and high operating cost. In contrast, the heat pump drying system has high thermal efficiency and low operating costs. The economics and efficiency of the heat pump sludge
dryer and the ordinary dryer were compared by Yu et al. [17]. From their experimental results, the dehumidification energy consumption of the heat pump sludge dryer and the ordinary dryer were 931 kJ/kg and 3384 kJ/kg of water, respectively. The heat pump dryer’s operational energy consumption was less than half of the dryers’ operational energy consumption with electricity or natural gas. Yang et al. [18] proposed a control method of a closed-cycle heat pump drying system to improve the accuracy of drying temperature and a superheating degree. They controlled the superheat degree by a proportional-integral-derivative (PID) controller, enhancing the electronic expansion valve’s response and accuracy. Şevik et al. [19] proposed a solar-assisted heat pump system with a flat-plate collector and a water source heat pump to dry mushrooms. During the drying process, the dry air temperature, relative humidity, and mushroom weight were monitored under different conditions. They observed that a solar-assisted water-source pump system could significantly improve the thermal efficiency. Due to solar energy use, the COP of the solar-assisted heat pump system is observed to be enhanced compared to that of a heat pump system.

Only a few studies have investigated the drying efficiency of a continuous heat pump system. The heat pump drying method was previously used for drying vegetables, meat, and fruits for storage. Nevertheless, it is used less in a continuous feeding drying system with a large drying capacity, which is essential to large scale industry. Using the heat pump drying system, the treated sludge’s water content, weight and volume can be reduced with a net increase in energy efficiency and a reduction in carbon dioxide emission. In this study, the airflow and temperature fields in a sludge drying chamber of the drying system are discussed to enhance the heat and mass transfer between the sludge and circulating airflow in the drying chamber of a continuous feeding air-source heat pump drying system. Further, the drying chamber’s internal deflector was appropriately designed to enhance the developed system’s performance. The airflow field and thermal temperature fields in the dry chamber were calculated by performing numerical simulation to find a suitable deflector design inside the chamber. Moreover, an additional cooling subsystem was designed to avoid the compressor overload and frequent shut down to remove the excessive heat from the drying chamber’s outlet. Using the additional cooling subsystem design, the refrigeration cycle’s condensation temperature could be maintained below a specific value, and the overload and frequent shutdown of the compressor could be prevented. Finally, the actual performance improvement of the heat pump drying system with a suitable chamber and additional cooling subsystem designs was verified through actual measurements for a long operational period.

2. Configuration of Heat Pump Drying System and Research Methodology

2.1. Principle and Configuration of Heat Pump Drying System

The studied heat pump drying system consisted of three air-source heat pump subsystems. High-temperature airflows are continuously fed into the drying chamber from the three heat pumps using supplied-air ducts. Subsequently, the chamber’s high humidity air was drawn to return to the corresponding heat pump through three other return air ducts (Figure 1a). Each heat pump subsystem possesses two compressors in the drying system that operate in parallel (Figure 1b). A stainless-steel conveyor belt with small holes on the belt surface was utilized to feed the sludge into the drying chamber from the upper right inlet. The high-temperature airflows from the heat pumps’ condensers induce heat and mass transfer with the continuous feeding sludge on the conveyor belt, further increasing the water vapor pressure inside the sludge. Subsequently, the sludge’s water content evaporates into the vapor due to the enhanced vapor pressure difference between the sludge and the circulating airflow. Subsequently, humid air was introduced into the heat pump’s evaporator through the return air ducts. The evaporators of the heat pumps cool down highly humid return airflows, and part of the water vapor in the return airflows are condensed into water by the evaporators. After the condensed water was precipitated, the comparatively dry air medium is introduced to the condensers for subsequent heating process and re-circulated to the drying chamber.
Figure 1. Schematic diagram of the sludge continuous feeding heat pump drying system used in this study. (a) Schematic diagram of a drying system with a heat pump. (b) Schematic diagram of a continuous sludge feeding drying system with three heat pump subsystems.

Figure 2 illustrates the sludge continuous feeding drying chamber’s configuration containing three layers of stainless-steel conveyor belts with small holes. Past a dehydrator, the sludge is transported to the right upper inlet of the chamber by a screw-type feeder. The sludge is then conveyed from the sludge inlet to the left sidewall along with the first conveyor layer, as shown in Figure 2a. At the left end of the first conveyor layer, the drying sludge drops from the first conveyor layer to the second conveyor layer. Similarly, the sludge is conveyed to the right side of the second layer, then drops to the third layer, and is finally conveyed to the chamber’s sludge outlet. The length, width, and height of the drying chamber are 8.65, 1.7 and 2.3 m, respectively. The width and length of each belt conveyor are 1.5 and 8 m, respectively. Small holes of 2 cm diameter and with 3 cm spacing between two holes are designed on each conveyor layer. At the lower part of the right sidewall, three air duct inlets with dimensions of 0.51 m and 0.51 m are designed for introducing high temperature circulating airflow from condensers of the three heat pumps. Three returning air duct outlets with dimensions of 0.54 m and 0.8 m are designed on the chamber’s top wall. Three high temperature and low humidity circulating airflow from the condensers are introduced into the drying chamber, which is then moved upwards to conduct the heat and mass transfer with the sludge on each conveyor belt. Finally, the circulating airflows with high humidity are sucked into each returning air duct and pass through the corresponding evaporators of the heat pumps. Part of the moisture in the circulating airflow condenses into water and is removed from the circulating airflows. To increase the circulating air’s detention time in the chamber and further enhance the efficiency of heat and mass exchange between the sludge and the circulating airflows, the air
deflectors with different off-set arrangements are designed. The chamber’s internal configuration is shown in Figure 2b.

Figure 2. Schematic diagram of a continuous sludge feeding drying chamber. (a) Front view of the continuous sludge feeding drying chamber. (b) Internal configuration and air deflector design inside the drying chamber.

The sludge drying capacity of the system in this study is 6 tons/day. The drying system consists of three heat pumps. Each heat pump consists of two compressors, two evaporators, two
condensers, and one energy recovery ventilator. The specification of each compressor in the heat pump is ZP182 (380 V, Copeland, St. Louis, MO, USA), with a cooling capacity of 59.5 kW, and the refrigerant mass flow rate is 323 g/s.

To verify the sludge drying system’s drying efficiency, the circulating air velocity, temperature, and humidity were measured by six air velocity meters, temperature meters, and humidity meters at both the supplied air inlets and return air outlets of the drying chamber. The power consumption of the three heat pumps in operation was measured by three power meters. The detection signals from the corresponding sensors every minute were transmitted to a data recorder with the specification of GL-840M (Graphitec, Tokyo, Japan) and measurement accuracy of ±0.002%. In the study, the humidity meter was a type EPDTH04101 manufactured by Embed Electronics Corp. (Philomath, OR, USA). The measurement accuracy was in the range of 0–100 °C ± 0.4 °C and 0–100% and 0–100% RH ± 6% RH. The air volume measuring device of type TES-3142 was manufactured by TES Electronics Corp. (Taipei, Taiwan). The measurement accuracy was ±1%. The anemometer (Testo 420, Testo Industry Corporation, Titisee-Neustadt, Germany) measurement range was 40–4000 m³/h and the accuracy error was ±3%. The power analyzer used was a harmonic clamp meter of type TES-3600. The measurement accuracy was ±1%. According to Kline and McClintock [20], the total measurement uncertainty of drying efficiency is ±6.9% in this study. The overall uncertainty, Δy, was calculated as the root of the sum of the square of each uncertainty using the equation Δy = [(Δu₁)² + (Δu₂)² + ⋯ + (Δuₙ)²]¹/₂, where Δu₁, Δu₂, ⋯, Δuₙ are the uncertainties of the individual variables.

The circulating air mass flow rate can be obtained from the average air velocity at the inlet of the drying chamber, the inlet’s cross-section area, and the density of the circulating airflow:

\[ \dot{m} = \rho A \bar{V}, \]

where \( \dot{m} \) is mass flow rate (kg/s), \( \rho \) is air density (kg/m³), \( A \) is cross section area (m²) and \( \bar{V} \) is air inlet velocity (m/s). The density (\( \rho \)) of the circulating air is 0.938 kg/m³ at a temperature of 50 °C and relative humidity of 20%.

Moisture removal rate \( (M_{vap}) \) by the heat pump drying system can be calculated in terms of the mass flow rate of the circulating air and humidity ratio difference between the inlet and the outlet of the drying chamber:

\[ M_{vap} = \dot{m} \times \Delta \omega, \]

where \( M_{vap} \) is instantaneous moisture removal rate (kg/sec), \( \Delta \omega \) is the humidity ratio difference between the inlet and the drying chamber’s outlet (kg/kg dry air). By integrating the instantaneous moisture removal rate over time, the total water removal amount in a drying period \( (G_{vap}) \) can be obtained:

\[ G_{vap} = \int_{0}^{\tau} M_{vap} \, dt, \]

where \( G_{vap} \) is total water removal amount (kg) and \( \tau \) is the drying time.

Specific moisture extraction rate \( (SMER) \) can be quantified by the ratio of the total water removal amount to total power consumption within the drying period:

\[ SMER = G_{vap}/E, \]

where \( E \) is total power consumption in the drying period (kWh).

2.2. Numerical Modeling

To extend the detention time of circulating airflow in the drying chamber, in addition to three belts of the conveyor, the deflectors are designed on the sidewalls of the chamber to guide circulating air to pass through each conveyor. A suitable deflector arrangement inside the drying
chamber was designed to enhance the heat and mass transfer between the circulating airflows and the conveyed sludge. For this, a computational fluid dynamic analysis was conducted to analyze the flow and temperature fields inside the drying chamber. Based on the simulation results, the average temperature difference of the circulating air flow between the air inlets and the chamber outlets can indicate heat transfer efficiency. Under the same airflow volume and air inlet temperature, a larger average temperature difference of the circulating air flow between the air inlets and the chamber’s outlets may indicate a higher heat transfer effect within the drying chamber. In the simulations, to simplify the problem, the mass transfer between the conveyed sludge and the circulating airflows is not considered. The standard $k-\varepsilon$ turbulent model was employed to consider the turbulent effect due to its minimal computational complexity.

2.2.1. Governing Equations and Boundary Conditions

To simplify the physical model, incompressible, Newtonian flow and isotropic porous material are used. The momentum equations are expressed as the generalized Darcy-Brinkman-Forchheimer model [21,22].

Mass conservation equation:

$$\frac{\partial U_j}{\partial x_j} = 0, j = 1, 2, 3$$

Momentum conservation equation:

$$\frac{\rho}{\beta} \frac{\partial U_i}{\partial t} + \frac{\rho}{\beta} U_j \frac{\partial U_i}{\partial x_j} (U_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \frac{(\mu_e + \mu_T)}{\beta} \frac{\partial U_i}{\partial x_j} \right) - \left( F \frac{\rho}{\sqrt{\kappa}} |u| + \frac{\mu}{K} \right) U_i + \rho f_i$$

Here $U_j$ is the air velocity component in the directions $x_j$ ($j = 1, 2, 3$), respectively, and $p, \mu, \mu_T, \rho, K, \beta$ denotes the average pressure, dynamic air viscosity, effective viscosity, air density, permeability, porosity, respectively. The Forchheimer coefficient, $F$, also known as the form-drag term, depends on the permeable membrane’s geometry and cannot be measured directly or determined analytically due to the lack of model equation relating them to fundamental quantities [23]. Forchheimer coefficient for packed-sphere bed and permeability is related to the porosity $\beta$ and the diameter of the solid particle, $d_{pr}$, of the porous medium [24,25]. $\rho f_i$ is known as the volumetric body force in the $j$ direction:

$$K = \frac{d_{pr}^2 \beta^2}{A(1-\beta)^2}, F = \frac{2}{(150\beta^3)^2/2},$$

where $A$ and $B$ are empirical constants found by Ergün [26] where $A = 150$ and $B = 1.75$.

Energy conservation equation:

$$\frac{\partial(T U_j)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( \frac{\nu}{Pr} \frac{\partial T}{\partial x_j} - \overline{T U_j} \right),$$

where $U_j$ is the average velocity in the direction $x_j$, $T$ is the average temperature. $P_-$ is the average pressure depression, which is the difference between pressure and ambient pressure, respectively. The Boussinesq approximation is employed. $\beta$ is the thermal expansion coefficient. The turbulent stress and turbulent heat flux $-\overline{T U_j}$ are derived from the turbulence closure model, stated by:

$$-\overline{u_i u_j} = 2\nu_t S_{ij} - \frac{2}{3} k \delta_{ij} - \overline{T U_j} = \frac{\nu_t}{Pr_t} \left( \frac{\partial T}{\partial x_j} \right)$$

where $\nu_t$ is the turbulent kinematic viscosity and $Pr_t$ is the turbulent Prandtl number (taken equal to 0.86); $k$ is the turbulent kinetic energy, $k = \frac{(u_i^2 + u_j^2 + u_k^2)}{3}$ and $\delta_{ij}$ is the Kronecker Delta. The $k-\varepsilon$ model, which involves two transport equations for the turbulent kinetic energy, $k$, and the turbulent dissipation, $\varepsilon$, was used to solve the turbulence closure problem.
The commercially available software ANSYS Fluent 15.0 (ANSYS, Inc, Cecil, PA, USA, 2014), was used for numerical analysis of the flow as well as temperature field of the computational domain.

2.2.2. Operation and Boundary Conditions

The dimensions of the drying chamber were 8.65 m × 1.7 m × 2.3 m (length × width × height). The cross-section area of the three air inlets of the drying chamber is 0.26 m², whereas the cross-section area of return air outlets is 0.43 m². Within the drying chamber, several air deflectors with a length of 1.4 m were placed in an offset arrangement on the internal wall to enhance the chamber’s heat and mass transfer. The supplied circulating air temperature to the chamber is 50 °C with an average air velocity of 7.4 m/s at three inlets. The enclosure of the drying chamber is assumed to be adiabatic. The sludge thickness on the conveyor belt is 0.05 m, and the average heat flux of the sludge on the conveyor belt is 23.4 kJ/(m² s), which was obtained from the latent heat of evaporation of water in the drying period. In the numerical analysis, the sludge’s water content reduced from the initial value of 80% to a final value of 30% in the drying period. For the drying system with a sludge drying capacity of 6 tons/day, the sludge’s required removal water content is 4286 kg. The evaporation latent heat of water is 2260 kJ/kg, and the average density of the drying sludge is 1250 kg/m³ from the actual measurement. The volume of the drying sludge of 6 tones is 4.8 m³. Thus, the volumetric heat source of 23.4 kJ/(m³ s) can be calculated.

3. Results and Discussions

3.1. Deflector Designs on Performance of Drying Chamber

To improve the heat and mass transfer between the sludge and circulating airflows inside the drying chamber, deflectors in the drying chamber are designed to guide the circulating airflows to pass through each conveyor belt and increase the detention time of the circulating airflows inside the drying chamber. In this study, seven different types of air deflector designs were proposed and numerical analysis for airflow and temperature fields of the seven proposed designs was carried out to find a suitable air deflector design. The seven different types of air deflector designs are shown in Figure 3. In the initial design of the drying chamber without the air deflector design inside the drying chamber (type I in Figure 3), most of the supplied airflow from the three air inlets directly flow towards the opposite wall of the chamber, then move upwards to its top wall and are sucked into the three return air outlets, as shown in the side view of type I. Based on the initial chamber's internal configuration design, small air circulation in the chamber results in insufficient airflow detention time and minimal airflows passing through each conveyor layer in the chamber, which would worsen the heat and mass transfer inside the chamber. For the type II deflector design, as shown in its side view in Figure 3, at the left sidewall of the chamber above each conveyor belt, a deflector plate with dimensions of 1 m × 0.35 m and a contact angle of 30° is designed to introduce the airflow from the bottom of the chamber to pass through each conveyor belt. In the side view of type II (Figure 3), the deflector plates are seen in a staggered arrangement with a gap between two deflector plates of 1.14 m to generate a lateral flow along each conveyor. As illustrated in the second column of Figure 3, the type III, IV, and VI deflector designs inside the drying chamber are similar to that of Type II. The length of each deflector plate in the type III deflector design enlarges to 1.2 m, and the gap between every two deflectors increases to 1.65 m as well. In the deflector design of type IV, the length of each deflector plate enlarges to 1.4 m, and the gap between two deflectors increases to 1.45 m, while in the type V deflector design, each conveyor connects two end sides of the chamber without gape space, as shown in the front view of type V in Figure 3. In type VI and VII deflector designs, to form a serpentine airflow along each conveyor from the inlet to the chamber’s upper outlet, deflector plates are designed on both sidewalls of the chamber with a contact angle of 30° as well. In the type VI deflector design, the deflector plates with a width of 0.35 m are arranged above the first and the third conveyor layers on the left sidewall, while on the right inner sidewall, the deflector plates are designed above the second conveyor layer,
shown as the side view in Figure 3. Along the conveyor moving direction, deflector plates on both sidewalls are in a staggered arrangement with a gap of 1.45 m between two deflector plates, shown as the front view of type VI in Figure 3. In the type VII deflector design, each deflector plate on both sidewalls extends to connect two end sides of the chamber without gap space, shown as the front view of type VII in Figure 3.
To evaluate the performance of heat and mass transfer between the sludge and the circulating airflow, the numerical analysis for airflow and temperature fields in the drying chamber with different deflector design types are carried out to find the suitable deflector design. The commercially available finite volume analysis software, ANSYS Fluent 15.0, was used for numerical analysis. The corresponding side view and front view of the airflow distributions in simulations are shown in Figure 4.

For the airflow distribution of type I without the deflector design on the sidewall (Figure 4, side view) the circulating airflow from the air inlet of the chamber flows directly to the opposite sidewall, then most of the circulating air moves upwards along the left sidewall of the chamber, and is drawn into the return air outlet of the chamber. In the type I design, a small volume of circulating air is guided to cross over each conveyor layer, which results in a low air circulation inside the chamber and worsens the heat and mass transfer effect between the sludge and the circulating air.

In the airflow distributions of type II to type IV, the circulating airflow from the lower inlet of the chamber approaches the left sidewall and moves upwards along the wall due to the deflector design on the left sidewall of the chamber, as shown in side view in Figure 4. Parts of the upward-moving circulating air touches the deflector plate above the third and the second conveyor layers and are guided to cross over these conveyor layers, due to staggered deflector arrangement on the left sidewall. Similarly, most of the airflow from the left sidewall touches the right sidewall, moves upwards along this wall, and is directly drawn into the chamber's air outlets. Considering the deflector plates above the first conveyor layer, most of the upward moving airflow is guided to pass over the third and the second layers; therefore, less upward-moving airflow is guided to the first conveyor layer.

In the airflow distribution of type V (Figure 4), each deflector connecting two sides ends along the conveyor transporting direction without any gap, therefore, most of the upward moving airflow from the bottom of the left sidewall touches the long deflector plate, and are guided to flow over the third conveyor layer, which then moves upwards along the right sidewall. Finally, the upward moving airflow along the right sidewall is directly drawn into the chamber’s air outlets. Thus, in type V deflector design, only a small circulating airflow volume is guided to flow over the second and the first conveyor layers. Which then moves upwards along the right sidewall. Finally, the upward moving airflow along the right sidewall is directly drawn into the chamber's air outlets. Thus, in type V deflector design, only a small circulating airflow volume is guided to flow over the second and the first conveyor layers.
In the airflow distributions of type VI (Figure 4), the deflector design on the chamber’s left sidewall has a staggered arrangement. Therefore, portions of the upward moving circulating air from the bottom left sidewall touches the deflector plates above the third and the first conveyor layers and are guided to cross over the third and the first conveyor layers, respectively, due to staggered deflector arrangement on the left sidewall. The airflow across the third conveyor touches the right sidewall and moves upwards along the wall; then, a portion of the airflow is guided by the deflector plates to flow conveyor above the second conveyor layers on the right sidewall and above the first conveyor layer on the left sidewall. A portion of the airflow is directly drawn into the air outlets along the right sidewall. As shown in the side view of type VI in Figure 4, a serpentine airflow distribution is formed from the third to the first conveyor layer inside the chamber.

However, in type VII’s airflow distributions, the deflector plates are arranged on both sidewalls and extend to connect the side ends along the conveyor moving direction. Because of no staggered deflector design, the upward moving circulating airflow from the left bottom sidewall touches the deflector above the third conveyor layer and is guided to flow over the third conveyor layer to the right sidewall. After touching the right sidewall, a portion of circulating airflow moves downwards and merges with the circulating airflow from the chamber’s inlet. The deflector further guides another portion of circulating air moving upwards on the right sidewall above the second conveyor layer to flow over the second conveyor to the left side wall conveyor. After touching the left sidewall, the upward moving circulating airflow is guided to flow over the first conveyor and is then drawn into the chamber’s outlets. In the type VII deflector design, a strong serpentine-like circulating airflow from the bottom of the third deflector stretches along the channels between any two conveyor layers, which is then drawn into the outlets of the chamber.
Figure 4. Airflow distributions inside the drying chamber with different deflector designs. (a) side view and front view of airflows in the drying chamber with type I deflector design. (b) side view and front view of airflows in the drying chamber with type II deflector design. (c) side view and front view of airflows in the drying chamber with type III deflector design. (d) side view and front view of airflows in the drying chamber with type IV deflector design. (e) side view and front view of airflows in the drying chamber with type V deflector design. (f) side view and front view of airflows in the drying chamber with type VI deflector design. (g) side view and front view of airflows in the drying chamber with type VII deflector design.

Figure 5 and Table 1 illustrate the average temperature differences and the average pressure difference between the chamber’s inlets and outlets with different deflector designs through numerical analysis. A higher temperature difference indicates a better heat exchange effect between the circulating airflows and the conveyed sludge. A more significant pressure difference between the inlets and outlets of the chamber indicates that greater energy consumption for the fans is required to sustain the required air circulating volume. According to Figure 5, the highest average temperature difference of 6.4 °C between the inlets and outlets of the chamber can be attained by the staggered deflector design with a deflector length of 1.4 m on the left sidewall (type IV), which enables the circulating air to conduct heat exchange with the conveyed sludge efficiently. In the staggered deflector design on the left side with different deflector gap spaces (type II to type V), quite the resulting average air pressure drop between the inlets and outlets of the chamber is in the range of 39.04 Pa to 40.9 Pa and the corresponding average temperature differences between the inlets and outlets of the chamber are from 6.02 °C to 6.36 °C. The deviations in average temperature differences and air pressure drop in the cases with the staggered deflector design on the left side are relatively small. However, compared to the case without deflector design, the average temperature differences can be increased from 4.67 °C to the highest value of 6.63 °C, but the average pressure drop only increases from 40.18 Pa to the highest value of 40.91 Pa in the cases with staggered deflector designs on the left side.

In the cases with staggered deflector design on both sidewalls of the drying chamber (type VI and type VII), compared to the case without the deflector design, the average temperature differences can be enhanced from 4.67 °C to the highest value of 4.97 °C, but average pressure drop increases from 40.18 Pa to the highest value of 50.42 Pa in the cases with staggered deflector designs on both sidewalls. The numerical results indicate that the serpentine-like airflow is formed inside the chamber by the staggered deflector design on both sidewalls in type VI and type VII cases, however, the average temperature differences are not apparent between the inlets and outlets, but the average pressure drop is quite obvious. This indicates that in the staggered deflector design on both sidewalls, limited enhancement in heat exchange effect between the circulating airflows and the conveyed sludge gains, but the greater energy consumption is required.
Figure 5. Average temperature differences and pressure drops with different air deflector designs in simulations.

Regarding the staggered deflector design on the left sidewall from type II to type V, with less energy consumption, there can be a significant increase in heat exchange effects between the circulating air and the conveyed sludge. Especially, in type IV, the highest average temperature difference of 6.4 °C between the inlets and outlets of the chamber can be attained by the staggered deflector design with a deflector length of 1.4 m on the left sidewall, which enables the circulating air to efficiently conduct heat exchange with the conveyed sludge, and a limited pressure drop in the chamber. Therefore, in the following improvement experiments, the type VI deflector design was adopted.

Table 1. Average temperature differences and pressure drops in simulations of different air deflector designs.

| Type | The Average Temperature Difference between the Inlet and Outlet (°C) | The Average Pressure Difference between the Inlet and Outlet (Pa) |
|------|---------------------------------------------------------------|---------------------------------------------------------------|
| I    | 4.6693                                                        | 40.1789                                                       |
| II   | 6.048                                                         | 39.2836                                                       |
| III  | 6.024                                                         | 39.3043                                                       |
| IV   | 6.365                                                         | 40.78623                                                      |
| V    | 6.213                                                         | 39.0371                                                       |
| VI   | 6.114                                                         | 40.9134                                                       |
| VII  | 4.653                                                         | 49.6086                                                       |

3.2. Drying Efficiency Improvement by Deflector Design

To evaluate the sludge’s weight reduction per day by the heat pump drying system with type IV deflector design, experiments were conducted for seven successive days within the drying chamber (Figure 6). The average sludge weight per day before and after drying was about 4.93 metric tons and 1.31 metric tons, resulting in 3.63 metric tons of water ejection using the proposed heat pump drying system. It can be noted that the average power consumption per day in the measurement period was 1985 kWh. Table 2 illustrates the water content before and after the sludge drying through the seven-day process and indicates the dehumidification capacity per day in the drying chamber, \( G_{\text{vap}} \), by considering the circulating humidity ratio’s differences in airflows between the supplied air inlets and return air outlets.
Table 2. Water content percentage before and after sludge drying treatment in experiments.

| Time   | Sludge Weight Before Drying (kg) | Water Content Percentage Before Drying (%) | Water Content Before Drying (kg) | Sludge Weight After Drying (kg) | Water Content Percentage After Drying (%) | Water Content After Drying (kg) | Water Content Reduction (kg) | $G_{vap}$ per Day (kg) |
|--------|----------------------------------|-------------------------------------------|--------------------------------|---------------------------------|------------------------------------------|--------------------------------|---------------------------|---------------------------|
| Day 1  | 5043.4                           | 79.2                                      | 3994.4                         | 1398.7                          | 25.1                                     | 351.1                          | 3643.3                    | 2082.6                    |
| Day 2  | 6040.1                           | 82.5                                      | 4983.1                         | 1531.9                          | 31.0                                     | 474.9                          | 4508.2                    | 2090.2                    |
| Day 3  | 5067.5                           | 78                                        | 3952.6                         | 1278.5                          | 12.8                                     | 163.7                          | 3789.0                    | 2299.2                    |
| Day 4  | 3585.0                           | 77.2                                      | 2776.6                         | 1058.8                          | 22.8                                     | 241.4                          | 2535.2                    | 1267.5                    |
| Day 5  | 5020.6                           | 80.1                                      | 4066.7                         | 1320.8                          | 24.4                                     | 322.3                          | 3744.4                    | 2035.9                    |
| Day 6  | 4840.0                           | 78.1                                      | 3780.0                         | 1280.2                          | 17.2                                     | 220.2                          | 3559.8                    | 1993.5                    |
| Day 7  | 4933.8                           | 79.5                                      | 3923.7                         | 1308.7                          | 22.2                                     | 291.2                          | 3632.5                    | 1967.9                    |

According to Table 2, the drying chamber can reduce about 74% of the sludge weight by average, with the deflector design inside the drying chamber. Based on the energy conservation for the heat pump’s refrigeration cycle, the ejected heat from the condensers to the circulating airs should be equal to the sum of heat extracted from the circulating airs in evaporators and compression by the compressors. Meanwhile, when the system is in operation, the heat ejected from the condensers to the circulating airflows is greater than the heat extracted from the circulating airflow by the evaporators. This results in a gradual increase in temperature in the circulating airflows with an increase in operational time. The increase in air temperature in the drying chamber may result in a higher condensation temperature of the heat pump and might further lead to overloading the compressors. Some compressors in the heat pump system are enforced to shutdown and high temperature circulating airflows in the drying chamber are released to the surrounding to avoid overloading. Once the chamber’s temperature reduces to a specific value, the compressors earlier shut down, restarted. As the high-temperature air is released to the surrounding, the calculated dehumidification capacities, $G_{vap}$, (quantified by humidity ratio differences between inlets and outlets) of the drying chamber are less in comparison to the corresponding sludge reductions (Table 2). The SMER value can be obtained by dividing the total dehumidification amount ($G_{vap}$) and total energy consumption ($E$) and for this study, it is nearly 1.83 kg/kWh.

Table 3 compares the average temperature differences between the supplied air inlets and return air outlets of the drying chamber, average water removal amount per day, average water content after drying treatment, and average SMER values with and without air deflectors. The average temperature differences between the inlets and outlets of the drying chamber with deflector design and without deflector design in numerical analysis are less than those in experiments by a margin of 2 °C to 2.2 °C, respectively. The average sludge weight reduction with air deflectors and without air deflector was 3632.5 kg/day and 3598.6 kg/day with drying times of 23.8 h and 18.3 h, respectively. Further, with the addition of air deflectors, the SMER value can reach a value as high as 1.83 kg/kWh compared to the value of 1.38 kg/kWh without an air deflector design.
Table 3. Evaluation of the difference between the two cases with or without air deflector.

|                      | Simulation | Experiments |
|----------------------|------------|-------------|
|                      | Average Air Temperature Difference (°C) | Average Air Temperature Difference (°C) | Average Dewatering Amount per Day (kg/days) | Average Water Content Percentage after drying | Average Required Drying Time per Day (hours) | SMER (kg/kWh) |
| Without air deflector | 4.7        | 6.8         | 3598.6      | 30.4%          | 23.8          | 1.38         |
| With type IV deflector | 6.4        | 8.4         | 3632.5      | 22.2%          | 18.3          | 1.83         |

3.3. Effect of Auxiliary Cooling Sub-System for Stable Operation of Drying System

The heat gain in the circulating airflow from the condenser is greater than the evaporators’ heat exchange for the circulating airflows inside the drying system. With the operational time-lapse, the circulating airflow temperature inside the drying system becomes increasingly higher due to the unbalance of heat gain and loss of circulating airflows in the circulating loop, which increases the condensation pressure of the refrigeration cycle. Excessive higher condensation pressure in the refrigeration cycle would trigger the overloading protection mechanism to shut down the operational compressor mandatorily. However, a frequent shutdown and start-up of the compressor would result in a high initial current of the compressor, a shortening of the compressor’s life cycle, and uneven moisture content of the discharged sludge. Moreover, to prevent the compressor from overloading, in the original system, a portion of the high-temperature circulating air is usually ejected to the ambient and the low-temperature fresh air is introduced into the air channel to cool down the circulating air. The ejection of circulating air from the drying system to the surrounding may cause environmental and air pollution problems.

To avoid the problems mentioned above and further enhance the operational performance of the drying system, an auxiliary cooling subsystem is proposed to remove the excess heat from the circulating air to attain stable operational conditions, which consists of a cooling tower, a cooling water pump, an inverter, and three fin-tube heat exchangers. Figure 6 illustrates the schematic diagram of the heat pump drying system with an auxiliary cooling system. Three fin-tube heat exchangers are installed behind the drying chamber’s return air outlets to remove the circulating air’s excess heat by the circulating cooling water from the cooling tower. Three temperature sensors are installed at the return air ducts behind the three heat exchangers to detect the temperature of the return air into the corresponding heat pumps and transmit the signals to the controller of the inverter which can modify the frequency of the driving power to further adjust the circulating water volume in terms of temperature variations of the circulating air, as shown in Figure 6. The return air temperatures entering the heat pump are maintained below 50 °C. With the operational time-lapse, the circulating air temperature gradually increases due to the unbalance of heat gain and the circulating air’s loss. If the temperature of the returning air entering the heat pump is above 42 °C, then the cooling water pump is launched to supply cooling water to the fin-tube heat exchanger with a low driving frequency of 50 Hz. With the increase or decrease in the circulating air temperature, the driving frequency rises and reduces linearly. When the detected circulating air temperature is less than 37 °C, the cooling water pump is shutdown.
Inside the three heat pumps, three energy recovery ventilators (ERVs) are designed to pre-cool the return air before the evaporator and pre-heat the return air after the evaporator. Figure 7 presents the difference in circulating air temperature before and the rear of the heat exchanger in the return air duct connecting the third heat pump in experiments, it can be seen that the return air temperature from the chamber fluctuates in a range of 39 °C to 47 °C; however, after passing through the heat exchanger, the return air temperature can be maintained at a temperature about 29.5 °C by linear increase or decrease in the driving frequency of the cooling water pump in terms of variations of average circulating air temperature into the heat pumps. With the varied cooling water volume, the circulating air temperature entering the third heat pump can be maintained at a stable value with less fluctuation. The average circulating air temperature from the chamber is about 42.3 °C (Figure 7). The circulating air temperature difference before and behind the heat exchanger can reach 12.8 °C.
Figure 7. Circulating air temperature difference before and the rear of the heat exchanger connecting the third heat pump in experiments.

Figure 8 illustrates the power consumption of the third heat pump without an auxiliary cooling subsystem in operation. With the operational time-lapse, due to the unbalance in heat gain and loss in the circulating air resulting in compressor overloading, the shutdown of one compressor of the third heat pump, and a sudden drop in the power consumption from its initial value of about 20 kW. The sudden shutdown frequency of the third heat pump without auxiliary cooling subsystem accounts four times per hour.

Figure 9 illustrates the measurement power consumptions of the heat pumps with an auxiliary cooling subsystem in operation. With the cooling water control to prevent the compressor from overloading, the circulating air temperatures can always be controlled below 50 °C in operation, and the frequent shutdown of the compressors can be avoided. As seen in Figures 9a–c, the three heat pumps’ power consumption varied at 17.5, 24.4 and 17.3 kW in the first, second, and the third pumps, respectively, without a sudden drop in power consumption. There was no sudden drop in power consumption during the data collecting periods, indicating that the three heat pumps can operate in stable conditions without a sudden shutdown of compressors.
Figure 9. Power consumption of the heat pumps with auxiliary cooling subsystem in operation. (a) Power consumption of the first heat pump. (b) Power consumption of the second heat pump. (c) Power consumption of the third heat pump.
Figure 10. The measurement difference in relative humidity between the inlet and outlet of the chamber connected to the third heat pump with auxiliary subsystem.

Figure 10 illustrates the relative humidity difference between the inlet and outlet of the chamber connected to the third heat pump with the auxiliary subsystem. The relative humidity of the supplied air from the third heat pump can be maintained at a stable and low value of 13% RH. The relative humidity of the return air at the outlet of the chamber increases due to the diffusion of moisture from the drying sludge, which can be maintained at about 30% RH with less fluctuation.

Figure 11 illustrates the measurement circulating air temperature difference between the inlet and outlet of the chamber connected to the third heat pump with an auxiliary subsystem. The temperature of the supplied air from the third heat pump can be stably maintained at about 50 °C. The temperature of the return air from the chamber’s outlet of the chamber can be increased to about 60 °C. The circulating air temperature difference between the inlet and outlet of the chamber is accounted about 10 °C.

Figure 11. Measurement circulating air temperature difference between the inlet and outlet of the chamber connected to the third heat pump with the auxiliary subsystem.

As shown in Figures 12a,b, throughout the experimental period, the measurement average percent water content and the average weight of the sludge feeding into the drying chamber per day, were in the range of 76.6% to 82.5% with an average value of 78.7%.
In particular, the average percent water content and the average weight of the sludge feeding into the drying chamber per day and from 5150 kg to 5920 kg with an average value of 5483 kg, respectively. After the drying treatment, the water content percentage could be reduced to 12.8% to 31% with an average value of 23.4%, and the sludge weight could be reduced to 1288 kg to 1598 kg with an average value of 1443 kg. The average reduction in water content percentage and sludge weight per day was 55.3%, and 4040 kg.

4. Conclusions

In this study, to increase the heat and mass transfer effect in the drying chamber and further enhance the heat pump drying system’s performance, air deflectors in the drying chamber were designed. The heat and mass transfer within the drying chamber was simulated through numerical analyses to find a suitable air deflector arrangement in the drying chamber. The off-set staggered deflector plate design with deflector length of 1.4 m and gap space of 1.45 m between two deflectors was selected for further investigation due to its high airflow temperature difference from numerical analysis results and less pressure drop between supplied-air inlets and return air outlets of the drying chamber. Actual experiments were conducted to verify the suitable air deflector design effect on the heat pump drying system’s performance. Using this design, the drying chamber can reduce about 74% of the sludge weight by average. In comparison to the drying chamber without deflector design, the average percent water content could be further reduced from 30.4% with a drying time of 23.8 h to 22.2% with a drying time of 18.3 h, and the SMER value could be enhanced.
from 1.38 kg/kWh to as high as 1.83 kg/kWh. Considering the sludge drying system, a total of 6 metric tons of sludge could be dried daily with high efficiency.

Besides, to prevent the overloading and frequent shutdown of the compressors in the heat pumps, an auxiliary cooling subsystem was adopted to remove excessive heat from the circulating airflow and provide stable operational conditions. Therefore, with the auxiliary cooling subsystem design, frequent shutdown of the compressors can be avoided and the SMER value can be further increased to a value of 1.94 kg/kWh.

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**References**

1. Industrial Development Bureau, Ministry of Economic Affairs. https://www.moeaidb.gov.tw/iphw/opc/service/training/25.pdf (accessed on 15 December 2020)
2. Luo, W.J.; Faridah, D.; Fasya, F.R.; Chen, Y.S.; Mulki, F.H. Performance Enhancement of Hybrid Solid Desiccant Cooling Systems by Integrating Solar Water Collectors in Taiwan. *Energies* **2019**, *12*, 3470.
3. Chang, C.C.; Luo, W.J.; Lu, C.W.; Chang, Y.S.; Tsai, B.Y.; Lin, Z.H. Effects of Process Air Conditions and Switching Cycle Period on Dehumidification Performance of Desiccant-Coated Heat Exchangers. *Sci. Technol. Built Environ.* **2017**, *23*, 81–90.
4. Li, K.Y.; Luo, W.J.; Hong, X.H.; Wei, S.J.; Tsai, P.H. Enhancement of Machining Accuracy Utilizing Varied Cooling Oil Volume for Machine Tool Spindle. *IEEE Access* **2020**, *8*, 28988–29003.
5. Luo, W.J.; Li, K.Y.; Huang, J.M.; Yu, C.K. Water Heating and Operational Mode Switching Effects on the Performance of a Multifunctional Heat Pump. *Energies* **2020**, *13*, 4896.
6. Aktaş, M.; Khanlari, A.; Aktekelı, B.; Amini, A. Analysis of a new drying chamber for heat pump mint leaves dryer. *Int. J. Hydrog. Energy* **2017**, *42*, 18034–18044.
7. Hii, C.L.; Law, C.L.; Suzannah, S. Drying kinetics of the individual belt of cocoa beans during heat pump drying. *J. Food Eng.* **2012**, *108*, 276–282.
8. Zhu, Z.; Yang, Z.; Wang, F. Experimental research on intermittent heat pump drying with constant and time-variant intermittency ratio. *Dry. Technol.* **2016**, *34*, 1630–1640.
9. Yahya, M.; Fudholi, A.; Hafizh, H.; Sopian, K. Comparison of solar dryer and solar-assisted heat pump dryer for cassava. *Sol. Energy* **2016**, *136*, 606–613.
10. Zhang, X.Y.; Chen, M.Q.; Huang, Y.W.; Xue, F. Isothermal hot air drying behavior of municipal sewage sludge briquettes coupled with lignite additive. *Fuel* **2016**, *171*, 108–115.
11. Li, H.; Zou, S.; Li, Y.; Jin, Y. Characteristics and model of sludge adhesion during thermal drying. *Environ. Technol.* **2013**, *34*, 807–812.
12. Ploteau, J.P.; Noel, H.; Fuentes, A.; Glouannec, P.; Louarn, S. Sludge convection drying process: Numerical modeling of a heat pump assisted continuous dryer. *Dry. Technol.* **2020**, *38*, 1261–1273.
13. Li, J.; Wu, C.W.; Fraikin, L.; Salmon, T.; Toye, D.; Nistajakis, E.; Léonard, A. Convective drying of sawdust-sludge mixtures: The effect of the sludge matrix. *Dry. Technol.* **2019**, *37*, 920–927.
14. Jamaleddine, T.J.; Ray, M.B. Drying of Sludge in a Pneumatic Dryer Using Computational Fluid Dynamics. *Dry. Technol.* **2011**, *29*, 308–322.
15. Janetti, M.B.; Colombo, L.P.M.; Ochs, F.; Feist, W. Effect of evaporation cooling on drying capillary active building materials. *Energy Build.* **2018**, *166*, 550–560.
16. Purlis, E. Modelling convective drying of foods: A multiphase porous media model considering heat of sorption. *J. Food Eng.* **2019**, *263*, 132–146.
17. Yu, W.; Yang, J.; Li, P.; Hu, Y.; Li, J.; Yue, Y.; Hu, N.; Yang, G. Sewage Sludge Drying Based on Heat Pump. In Challenges of Power Engineering and Environment; Publisher: Springer Berlin/Heidelberg, Germany, 2007; pp. 1285–1288.

18. Yang, Z.; Zhu, Z.; Zhao, F. Simultaneous control of drying temperature and superheat for a closed-loop heat pump dryer. Appl. Therm. Eng. 2016, 93, 571–579.

19. Şevik, S.; Aktaş, M.; Doğan, H.; Koçak, S. Mushroom drying with solar assisted heat pump system. Energy Convers. Manag. 2013, 72, 171–178.

20. Kline, J.; McClintock, F.A. Describing uncertainties in single sample experiments. Mech. Eng. 1953, 1, 3–9.

21. Chung, L.P.; Derek, R. Using Numerical Simulation to Predict Ventilation Efficiency in a Model Room. ROOMVENT 1994, 94, 16–27.

22. Costa, V.A.F.; Oliveira, L.A.; Baliga, B.R.; Sousa, A.C.M. Simulation of coupled flows in adjacent porous and open domains using a control-volume finite-element method. Numer. Heat Transf. A 2004, 45, 675–697.

23. Lage, J.L. The Fundamental Theory of Flow through Permeable Media from Darcy to Turbulence in Transport Phenomena in Porous Media; Ingham, D.B., Pop, I., Eds.; Elsevier: Amsterdam, The Netherlands, 1998; pp. 1–30.

24. Baytaş, A.C. Thermal non-equilibrium natural convection in a square enclosure filled with a heat generating solid phase, non-Darcy porous medium. Int. J. Energy Res. 2003, 27, 975–988.

25. Al-Amiri, A.M. Analysis of momentum and energy transfer in a lid-driven cavity filled with a porous medium. Int. J. Heat Mass Transf. 2000, 43, 3513–3527.

26. Ergün, S. Fluid flow through packed columns. Chem. Eng. Prog. 1952, 48, 89–94.

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