Performance Simulation of Desiccant Wheel under Dynamic Conditions: Comparison between Detailed and Simplified Models

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Abstract

In the last few decades, European countries have been facing an increasing demand for active air-conditioning (cooling and dehumidification) in the summer period. As a good alternative to energy demanding vapor compression cooling-based air dehumidification, building HVAC systems integrating desiccant-based dehumidification has drawn increasing attention. These technologies offer the possibility to significantly reduce the energy requirement for air dehumidification and post-heating due to excessive cooling. In fact, air-conditioning systems that use solid or liquid desiccant offer the interesting capacity of separating dehumidification and sensible cooling of air and realizing high-energy-efficiency systems. However, the complexity perceived by technicians towards the design of air-conditioning systems based on these technologies actually limits their adoption in HVAC systems, mainly due to the difficulties in predicting the performance of the desiccant devices, which is the crucial component of the system. On the one hand, many simplified approaches commonly adopted to simulate and optimize the dehumidification performance are based on steady-state models and their reliability under unsteady conditions is questionable; on the other hand, accurate detailed models available for the design and development of components do not turn out to be particularly suitable for simulation of energy systems, due to their high computational cost. The present work focuses on desiccant wheels, whose performance is not only directly related to the properties of the sorption material, but also depends strongly on operating conditions, such as rotational speed, regeneration temperature and inlet air conditions, which are typically non-stationary in real application. In this context, the purpose of this paper is to assess the reliability of a simplified model to predict the behavior of a desiccant wheel under dynamic conditions. To do so, a detailed model of a desiccant wheel is developed and validated against experimental data available in the literature. Finally, a comparison between the developed detailed model and the simplified model under dynamic conditions is carried out.

1. Introduction

The adoption of new and efficient dehumidification technologies as an alternative to condensation is attracting increasing interest, both in civil application and industrial production. Compared with the traditional vapor compression dehumidification method, the absorption dehumidification method can save up to 40 % energy (Du & Lin, 2020) and make full use of renewable energy sources. Moreover, adsorption dehumidification systems allow improved control of systems with advantages for occupants' thermohygrometric comfort. As clearly discussed in many review papers (Ge et al., 2014; Daou et al., 2006; Sultan et al., 2015), desiccant wheel systems are attracting increasing interest because they offer advantages over other air conditioning systems, such as the possibility of:

i) using water as a natural refrigerant and other environmentally-friendly desiccant materials (such as silica gel and zeolites);
ii) making energy-efficient cooling systems that work with the sensitive and latent loads, the application of which is possible under different environmental conditions;
iii) meeting the requirements of miniaturization and being less subject to corrosion compared with the liquid desiccant system (in which the liquid and air directly interact);
iv) integrating low-grade heat sources (such as solar energy, geothermic energy and waste heat), hence significantly reducing the operating costs;

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v) overcoming the discontinuous problem of the fixed-bed desiccant cooling system.

The rotary desiccant wheel is a relatively mature technology, yet its wide application is still limited due to the complexity perceived by technicians towards the design of air-conditioning systems based on this technology. The design of HVAC systems based on this technology is quite complex because of the difficulties in predicting the performance of the desiccant wheel, which is the crucial component of the system. In fact, the performance of the desiccant wheel is critical to the capability, size and cost of the whole system (De Antonellis et al., 2010). Desiccant wheel performance strongly depends on regeneration temperature on the revolution speed, inlet airflow conditions (temperature, humidity and flow rate) and on the coupled heat and mass transfer within the desiccant. These aspects greatly complicate the development of models which can accurately predict the performance of a desiccant wheel under non-stationary conditions without incurring in high computational load and complexity. On the one hand, many detailed desiccant wheels models have been based on a detailed physical approach and they are particularly suitable for the development and design of components (Ge et al., 2008); on the other hand, simplified approaches based on practical correlations (Angrisani et al., 2012; De Antonellis et al., 2015; Jurinak, 1982; Panaras et al., 2010) are commonly used to simulate and optimize the dehumidification performance, but their reliability under unsteady conditions is questionable. The purpose of this work is to assess the reliability of simplified models to predict the behavior of a desiccant wheel under dynamic conditions. The rotary desiccant dehumidifier model contained in the TESS Component Libraries for Trnsys18™ was chosen to represent the simplified models based on the correlations. A detailed model of a desiccant wheel was developed and validated against experimental data available in the literature and, finally, a comparison between these two models was carried out.

A desiccant wheel is a cylindrical rotating device generally consisting of a structure of several channels. The channels run in the axial direction of the wheel and are parallel to each other. Depending on the manufacturing process, they can have usually a rectangular, triangular or sinusoidal shape. The structure is made by supporting material impregnated with an adsorbent substance (desiccant) in a typical content f of 70–80 %. The most widespread support materials are paper, aluminium, synthetic fibers or plastic, while common adsorbents are silica gel, zeolite and activated alumina (De Antonellis et al., 2015).

In its basic configuration, the wheel is divided into two sections, where the air streams are arranged in counter-flow. In the process section, the air stream is dehumidified and undergoes heating. In the regeneration section, an air stream is heated before passing through the wheel to increase its moisture-holding capacity; the regeneration air stream passing through the wheel removes vapor from the desiccant material and exits cooled and humidified. A diagram of a desiccant wheel is shown in Fig. 1.

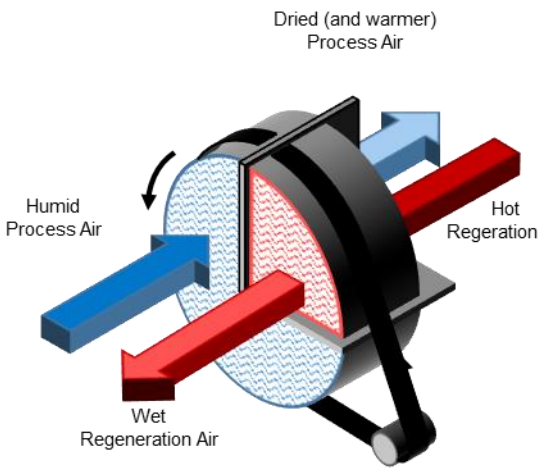

Fig. 1 – Diagram of a desiccant wheel

2. Model Description

Two models were considered in this paper to simulate the performance of a desiccant wheel such as the one shown in Fig. 1. In the first defined detailed model, the coupled heat and mass transfer within the wheel is modeled in detail. The simplified model is a correlation-based model that can immediately provide output conditions without the need to specify detailed parameters of the desiccant wheel.

In the detailed model, to reflect the actual transfer processes occurring in the desiccant wheel, a gas and solid side resistance was applied, where also the solid side heat conduction and mass diffusion resistances were considered. Compared with only gas-side resistance models, gas and solid side resistance models are more related to the actual process in the desiccant wheel (Ge et al., 2008). However, the diffusion and adsorption processes inside the desiccant are lumped into the mass and heat transfer coefficients. The precision of these models is considered satisfactory because the desiccant layer is rather thin (Ge et al., 2008). The model also takes also into account heat and mass transfer from the desiccant to the air stream and the developing temperature and velocity profiles along the desiccant wheel channels.

The numerical analysis is based on the following assumptions:

a) Heat and mass transfer from the wheel to the surroundings are negligible;
b) The channels are considered identical and uniformly distributed throughout the wheel;
c) Supporting and desiccant materials are evenly distributed in the layer.
d) The properties of the dry desiccant material, as well as of the supporting material are constant;
e) Heat and mass transfer between adjacent channel are negligible: temperature and moisture content gradients in circumferential and radial directions are not considered;
f) The hygroscopic capacity of supporting material is negligible compared with the adsorbent;
g) The inlet air conditions are uniform and the air flow is one-dimensional;
h) Air leakages between the two streams are negligible;
i) Heat conduction in humid air is negligible;
j) Pressure loss of the air stream is negligible for heat and mass transfer processes (the thermodynamic properties are unaffected)
k) Axial heat conduction and mass diffusion in the air are small compared with convective processes;
l) The vapor enters the pores, diffuses in the pores, and, meanwhile, is adsorbed.
m) The influence of the pressure drop in axial direction on heat and mass transfer is neglected;
n) The heat of adsorption is set free in the layer immediately when the vapor enters the porous layer and is partially convected into air stream.

The schematics of a channel segment in the desiccant wheel is shown in Fig. 2.

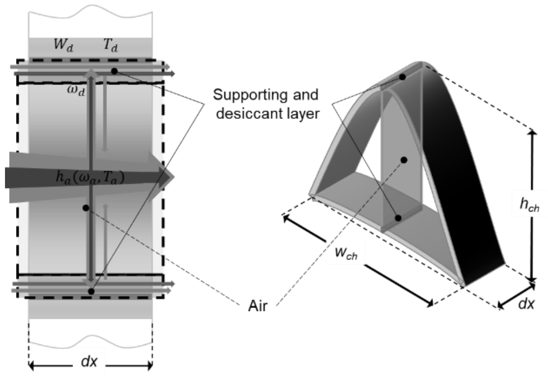

Fig. 2 – Schematic of the control volume in a channel segment (right) and mass and energy transfers in a control volume (left)

In the detailed model, one channel segment with an infinitesimal length dx is selected as the control volume (Fig. 2 - left). The control volume is separated into two nodes, one is the humid air in the channel and the other is the layer composed of supporting and desiccant materials (Fig. 2 - right). The layer of supporting and desiccant material is shared by two channels: therefore, the thickness of the layer in one control volume is half of its actual value and the middle of this layer is considered to be adiabatic.

Referring to Fig. 2, the following laws were applied to the infinitesimal control volume.

Mass balance of water in the desiccant material and adsorbed water:

$$\begin{aligned}\rho_a \varepsilon A_{ch}(1-f)\frac{\partial \omega_d}{\partial t} + \rho_d(1-\varepsilon)A_{ch}(1-f)\varphi\frac{\partial W}{\partial t} \\ = \rho_a \varepsilon A_{ch}(1-f)D_{eff}\frac{\partial^2 \omega_d}{\partial x^2} \\ + \rho_d A_{ch}(1-f)D_s\frac{\partial^2 W}{\partial x^2} \\ + h_m P_{ch}(\omega_d - \omega_a)\end{aligned} \tag{1}$$

Energy balance for the desiccant material, supporting material and adsorbed water:

$$\begin{aligned}\rho_s(1-\varepsilon)A_{ch}(1-f)c_{p,s}(1-\varphi)\frac{\partial T_d}{\partial t} \\ + \rho_d(1-\varepsilon)A_{ch}(1 \\ - f)c_{p,d}\varphi\left(\frac{\partial T_d}{\partial t} - \frac{\lambda_d}{\rho_d c_{p,d}}\frac{\partial^2 T_d}{\partial x^2}\right) \\ = h_{th}P_{ch}(T_a - T_d) \\ + h_m P_{ch}(\omega_a - \omega_d)c_{p,v}(T_a - T_d) \\ - h_m P_{ch}(\omega_a - \omega_d)(1-\eta)i_{ad}\end{aligned} \tag{2}$$

Mass balance of water in the air stream:

$$\rho_a f A_{ch}\left(\frac{\partial \omega_a}{\partial t}+u\frac{\partial \omega_a}{\partial x}\right)=h_m P_{ch}(\omega_d-\omega_a) \tag{3}$$

Energy balance in the air stream:

$$\begin{aligned}\rho_a f A_{ch}(c_{p,a}+\omega_a c_{p,v})\left(\frac{\partial T_a}{\partial t}+u\frac{\partial T_a}{\partial x}\right)\\ &= h_{th}P_{ch}(T_d-T_a)\\ &+h_m P_{ch}(\omega_d-\omega_a)c_{p,v}(T_d-T_a)\\ &-h_m P_{ch}(\omega_d-\omega_a)\eta i_{ad}\end{aligned} \tag{4}$$

To solve the system of partial differential equations, a set of boundary and initial conditions is needed. Assuming adiabatic and impermeable boundaries at the entrance and exit flow channel leads to a negligible error: according to Simonson and Besant (1997), the transfer area at the inlet and outlet of the wheel correspond to less than 0.1 %. Therefore, the following relationships apply for the support and desiccant layer:

$$\begin{aligned}\left.\frac{\partial T_d}{\partial x}\right|_{x=0}&=\left.\frac{\partial T_d}{\partial x}\right|_{x=L}=0\\ \left.\frac{\partial \omega_d}{\partial x}\right|_{x=0}&=\left.\frac{\partial \omega_d}{\partial x}\right|_{x=L}=0\end{aligned} \tag{5}$$

The temperature, humidity ratio and velocity boundary conditions for the air are given by Dirichlet boundary conditions periodically switching between process and regeneration air stream:

$$\begin{aligned}T_a(0,t)&=\begin{cases}T_{p,inlet}\\ T_{r,inlet}\end{cases}\\ \omega_a(0,t)&=\begin{cases}\omega_{p,inlet}\\ \omega_{r,inlet}\end{cases}\\ u_a(0,t)&=\begin{cases}u_{p,inlet}\\ u_{r,inlet}\end{cases}\end{aligned} \tag{6}$$

Assuming uniform initial temperature and humidity ratio of the air and of the support and desiccant, we have:

$$\begin{aligned}T_a(x,0)&=T_{a0}\\ \omega_a(x,0)&=\omega_{a0}\\ T_d(x,0)&=T_{d0}\\ \omega_d(x,0)&=\omega_{d0}\\ W(x,0)&=W_0\end{aligned} \tag{7}$$

Additional equations are needed to solve the initial-boundary-value problem.

The equilibrium water uptake in the desiccant material can be expressed by a general sorption curve that directly links the water uptake *W* to the relative humidity.

The isosteric heat of adsorption i_{ad} of silica gel calculate using the equation recommended by San (1993). The effective diffusion coefficient D_{eff} accounts for both molecular diffusion and Knudsen diffusion. However, as reported by Pesaran and Mills (1987), since most of the pores of silica gel are less than 100 10^{-10} m, ordinary diffusion can be ignored in usual silica gel applications. The surface diffusion *Ds* is evaluated with the relationship proposed by Pesaran and Mills (1987).

The heat transfer coefficient h_{th} is derived from the local Nusselt, calculated following equation of Niu and Zhang (2002). Assuming a sinusoidal geometry for the channel, the Nusselt number for the fully developed flow and the equivalent diameter were calculated through the correlations proposed by Kakaç et al. (1987). The mass transfer coefficient h_m was derived from the Sherwood number.

Given the initial and boundary condition, the partial differential equations system of the four non-linear and coupled heat and mass transfer equations is implemented and solved in Matlab™ environment.

In the simplified model developed by Howe (1983) and based on the original work of Jurinak (1982), the outlet air conditions (humidity ratio and temperature) are provided through two combined potentials *F1* and *F2* for a silica gel desiccant defined in the following way:

$$F1=\frac{-2865}{T^{1.490}}+4.344\,\omega^{0.8624} \tag{8}$$

$$F2=\frac{T^{1.490}}{6360}+1.127\omega^{0.07969} \tag{9}$$

In order to obtain the process air outlet condition, Eqs. (8) and (9) should be numerically solved to get the corresponding values of temperature and humidity ratio. The model computes the values of *F1* and *F2* for a given set of design conditions of both the process and regeneration streams, then uses an iterative process to guess and then converge to the values of the outlet conditions.

3. Validation and Model Comparison

To validate the models, the model results were compared with the experimental data of a commercial desiccant wheel produced by the Japanese manufacturer Seibu Giken Co. Ltd. (Kodama et al., 1993) available in the literature. The supporting layer of the wheel in consideration is made of ceramic porous fiber paper, impregnated with silica gel. The constant thermophysical properties and geometrical parameters of the wheel are listed in Table 1.

Table 1 – Thermophysical and geometrical parameters assumed for the simulation

Angle of regeneration	90°
Desiccant material	Silica gel type A
Porosity	0.4
Volume ratio of desiccant in the layer	0.7
Supporting material	Ceramic fiber sheets
Channel pitch (w x h)	3.2 x 1.8 mm
Wheel diameter	320 mm
Wheel length	200 mm
Rotation speed	6 rph

The structure contains between 70 % and 80 % type A silica gel. The regeneration zone occupies a quarter of the frontal area of the wheel, while the remaining area is dedicated to process air dehumidification. There is 20 mm of brass between the two zone with no air flow. However, the presence of these two separators was not considered in the detailed model. All three experimental series were obtained at the optimum wheel speed, equal to 6 rph.
Table 2 reports the experimental data used in the comparison for the inlet process air conditions. For the inlet regeneration conditions, the same humidity ratio of the process air and a constant regeneration temperature of 140°C is assumed.

Table 2 – Experimental inlet conditions for the process air (Kodama et al., 1993)

	Series 1	Series 2	Series 3
Temperature	24.4 °C	23.7 °C	23.3 °C
Humidity ratio	14.2 g/kg	8.9 g/kg	7.3 g/kg

The *F1* and *F2* potentials of the simplified model were calculated by taking as reference the outlet conditions for process and regeneration air from another series of data of the same experiments taken under similar conditions (Kodama et al., 1993). Then, these potentials were used to evaluate the output conditions for the three series considered here.
The two models were compared under dynamic conditions with varying inlet air temperature and humidity. The other model inputs (regeneration temperature, rotation speed, air flow rate) were kept constant, as were the characteristics of the desiccant wheel. In order to provide a representative input for a real application, inlet temperature and humidity ratio are generated from a monitoring data set obtained from a weather station installed at the Free University of Bozen-Bolzano. The sampling time is 1 minute.

4. Results and Discussion

Fig. 3 and Fig. 4 show the outlet process air humidity ratio and temperature as a function of the angular position. These figures provide a graphical comparison of the experimental and simulated outlet conditions as function of the angle. The data depicted in these figures refer to Series 1 (in blue) Series 2 (in red) and Series 3 (in green). The experimental angular values are plotted with markers, the dashed line is the angular distributions obtained from the detailed model, while the dotted lines refer to the outputs of the simplified model. It has to be mentioned that the plotted angular distributions of the detailed model are those obtained once the transient period has ended and the outlet conditions have reached stable values.

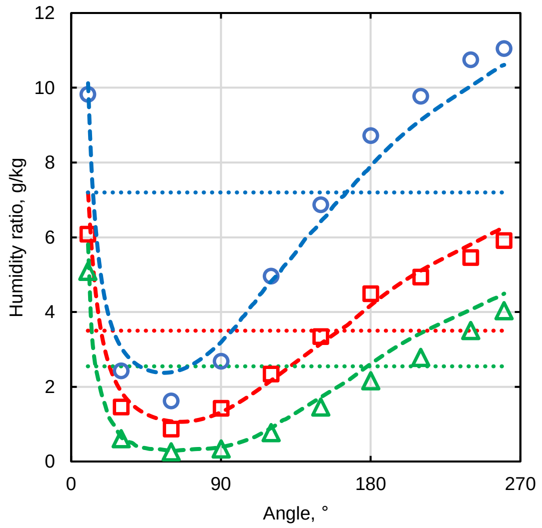

Fig. 3 – Angular distribution of the humidity ratio of processed air at the outlet

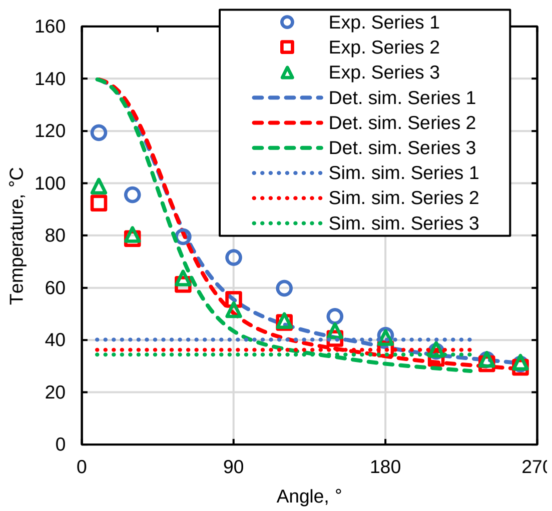

Fig. 4 – Angular distribution of the temperature of process air at the outlet

The detailed model is able to capture the physics of the problem and reproduce with fidelity the angular distribution of the outlet humidity ratio; however, the errors obtained for the single angular values of the humidity ratio may be significant (up to 47 %). The angular distribution of the outlet temperature differs more from the experimental data than does the humidity ratio. The largest differences between simulated and experimental values occur at the first angular positions where the transition between regeneration and process has just occurred. For all the experimental series, the detailed model underestimates the average temperature and humidity ratio of the outlet process air (calculated as the average of the angular values). The difference in the process air average outlet conditions between simulated and the experimental values is always below 12 % (largest error for Series 1) for the humidity ratio and 10 % for the temperature (largest error for Series 2).

Regarding the simplified model, as shown in Fig. 3 and Fig. 4, this cannot provide the angular distribution of temperature and humidity ratio, but only a constant value corresponding to the average outlet conditions of the process air leaving the desiccant wheel. Comparing the values provided by the simplified model against the experimental data for Series 1 and Series 3, we have an overestimation of the output humidity ratio, with an error of 5 % and 21 %, respectively, while for Series 2, the model overestimates the dehumidification capacity of the wheel (difference in the outlet humidity ratio equal to 4 %). The temperature in the outlet process air is always underestimated by the simplified model with an error ranging between 31 % and 36 %.

The two models were compared under dynamic conditions with varying inlet air temperature and humidity. The other model inputs (regeneration temperature, rotation speed, air flow rate) were kept constant, as were the characteristics of the desiccant wheel. The humidity ratio profile obtained from the two models of the are shown in Fig. 5.

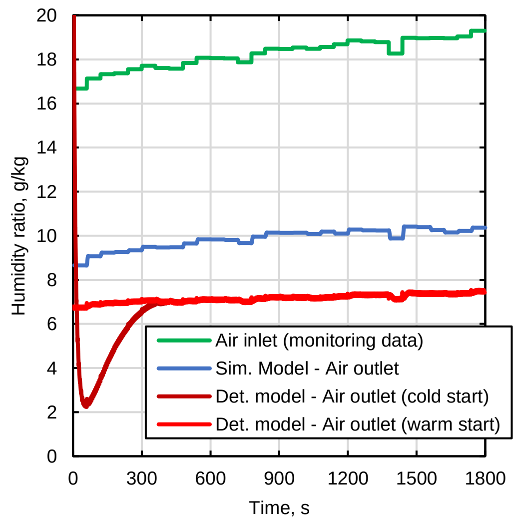

Fig. 5 – Inlet and outlet humidity ratio under dynamic conditions

Although involving a significant difference in terms of absolute values, simplified and detailed model (warm start) return similar trends for outlet air conditions. This is true when the initial wheel state in the detailed model is derived from an earlier operation state (warm start). On the other hand, if a cold start is considered (the initial state of the wheel is assumed to be in equilibrium with the environment), the detailed model reproduces typical transient trends and, once exhausted (after about 400 s), it leads to the reconciliation of the profiles with those obtained with the warm start.

5. Conclusion

A detailed and a simplified model of a desiccant wheel were implemented. The performance of the desiccant wheel simulated by the two models was compared against experimental data. Both models are able to provide the humidity of the air leaving the wheel with small errors. Within the limits of the conditions considered in this study, the simplified model presents larger errors in simulating the outlet temperature in comparison with the detailed model. Comparison under dynamic conditions shows that the simplified model cannot reproduce any transient regime of the desiccant wheel. This limitation may lead to errors in the prediction of output conditions, especially in the cold start case, as evidenced by the simulations conducted. However, once the steady state condition is reached, the simplified model returns results with trends similar to the detailed model with significant computational cost savings.

The main limitations of the simplified model considered in this study are related to the fact that it must be initialized with reference conditions that are bound to rotational speed of the wheel, regeneration conditions and air flow rate. This implies that the simplified model can return reliable results only when the simulated conditions are similar to the reference conditions under consideration.

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Nomenclature

Symbols

A_{ch}	Cross sectional area of the channel (m^2)
c_p	Isobaric specific heat (J kg^{-1} K^{-1})
D_{eff}	Effective diffusivity (m^2 s^{-1})
D_s	Surface diffusivity (m^2 s^{-1})
f	Area ratio of flow passage in the channel
h_m	Mass convection coefficient (kg m^{-2} s^{-1})
h_{th}	Heat convection coefficient (W m^{-2} K^{-1})
i_{ad}	Isosteric heat of adsorption (J kg^{-1})
P_{ch}	Perimeter of the flow passage (m)
t	Time (s)
T	Temperature (K)
u	Air velocity in the flow passage (m s^{-1})
W	Water uptake in the desiccant (kg$_w$ kg$_d$$^{-1}$)
x	Axial coordinate
ε	Porosity
η	Fraction of ads. heat convected to air
λ	Thermal conductivity (W m^{-1} K^{-1})
ρ	Mass density (kg m^{-3})
ω	Humidity ratio (kg$_v$ kg$_{da}$$^{-1}$)

Subscripts/Superscripts

a	Air
d	Desiccant material
s	Supporting material
v	Water vapor

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