

Performance assessment of a solar-powered adsorption air conditioning system

C E Panaite¹, A Popescu¹, V M Homutescu¹ and K Uzuneanu²

¹Mechanical Engineering and Automotive Department, “Gheorghe Asachi” Technical University of Iași, Iași, Romania

²Thermal Systems and Environmental Engineering Department, “Dunărea de Jos”, University of Galați, Galați, Romania

E-mail: epanaite@tuiasi.ro

Abstract. Rotary desiccant air conditioning is a thermally driven and CFCs free technology which mainly uses low grade heat sources, such as solar energy and waste heat. The dehumidifier is the critical component of the desiccant air conditioning system and its performance affects in a decisive manner the effectiveness, size and manufacture cost of the whole system. This paper presents an adequate simulation tool to predict the behaviour of packed bed dehumidifier of rotary wheel type under appropriate working conditions to a solar-assisted configuration. The analysis is carried out through a one-dimensional gas-side resistance model of heat and mass transfer processes on air-solid desiccant interface and provides a large number of numerical results that can be used for design and operational optimization of the desiccant wheel.

1. Space cooling – facts and trends

In the European area, heating and cooling is the largest energy use sector, being in charge of 546 Mtoe, half of the EU's final energy consumption and it is forecasted to maintain its leading position under both business-as-usual and decarbonisation scenarios by 2030 and 2050 [1,2]. Out of the total amount of energy assigned to heating and cooling, 45% is used in residential sector, 37% in industry and 18% in services [1]. The largest share of the energy used for heating and cooling purposes comes from fossils fuels (about 75%), which raises concerns about environmental effects and energy supply security.

Buildings account for about 40% of energy consumption and provide 36% CO₂ emissions in EU area and similar figures have been reported globally [3]. Significant proportion of this energy (about 85%) is used by HVAC equipment to meet thermal comfort requirements. Space heating dominates energy use in buildings and represents over half of total consumption. Even that space cooling accounts a modest share of 2% at European level, energy demand for space cooling has pursued a steady increase during the last decades, in both residential and services sub-sectors [4]. Studies on this topic show that this will increase sharply due to the growing thermal loads driven by climate change, new architectural trends such as increase of the ratio of transparent to opaque surface in the building facades but especially due to higher indoor comfort demand of the occupants sustained by higher income and lower prices of AC units in the emerging economies.

By far, the large majority of the space cooling is supplied by electrically driven devices based on vapour compression refrigeration cycle. The fast growing demand for air conditioning is resulting in



higher consumption of electricity and implicitly a higher consumption of primary energy, dominated by fossil fuels. A rough estimate of the world electricity consumption for air conditioning is of one trillion kWh annually and the energy for space cooling could increase by 10 times by 2050 [5].

The responsibility to secure fossil resources for future generations and the necessity to reduce the emission of Green House Gases (GHG) are two of the greatest challenges we have to deal with this century. Sustained efforts are needed to reach the EU climate and energy targets for the years 2030 and 2050 and, in this regard, a particular attention is paid to renewables energy. New technologies for air conditioning are investigated to overcome the environmental and economic issues and one of the most promising is the solar-powered cooling technology. The fact that peak cooling demand in summer is associated with high solar irradiance makes this concept very attractive [6].

Solar cooling technologies are grouped into two broad categories: solar thermal cooling (the heat needed for cooling process is provided by solar collectors) and solar electric cooling (that uses photovoltaic panels to generate electricity for vapour compression chillers) [7]. Due to higher efficiency and also a lower initial cost of solar thermal collectors compared to that of photovoltaic panels, during the last decades more interest has been paid to the solar thermal-drive cooling technologies, especially solar sorption (adsorption and absorption) [8]. More than 1,200 solar thermal cooling systems were installed worldwide by the end of 2014 and the market shares depiction looks like this: absorption cooling systems - 70%; solid desiccant cooling systems - 14%; adsorption cooling systems - 13%; liquid solar desiccant cooling systems - 2 % and others - 1% [2, 9]. Even if the overall number of the systems installed up to now indicates that solar cooling is still a niche product, its market potential is huge. The International Energy Agency (IEA) estimates that by 2050, 17% of the total global demand for cooling shall be covered by solar cooling, which means 1,5 EJ per year [10].

In this context, the present paper provides an approachable and accurate method to analyze solar desiccant cooling systems, with a focus on the major component, the dehumidifier of rotary wheel type. A detailed mathematical model of heat and mass transfer processes manifesting at the moist air-solid desiccant interface has been developed. The main goal was to predict the desiccant wheel behaviour under different working conditions in terms of its performance and effectiveness.

2. Principle of desiccant evaporative cooling

Desiccant cooling systems are open cycle systems that use water as refrigerant and a hygroscopic material (desiccant) as sorbent. The cooling effect is produced through a combination of air dehumidification and adiabatic cooling that is why these systems are generally known as desiccant evaporative cooling (DEC) systems [11]. Both parameters, air temperature and air humidity, can be adjusted to the indoor comfort requirements and the necessary fresh air is also supplied at the same time. For continuous operation of the system, heat provided by solar collectors or other energy source is used for desiccant regeneration.

The interest from the researchers and manufacturers for DEC technologies is justified by the advantages they offer [11-13]: low driving temperatures, in range 60°C - 90°C, which make them suitable for low grade heat supply, like solar or waste heat sources; high COP values, especially for liquid-based technologies; environmental friendly refrigerant (water); working without noise and vibrations; low operating cost; inexpensive desiccant materials; relatively short payback period.

Different desiccant materials are used, in either liquid or solid phases, the most commonly being silica gel, calcium chloride, lithium bromide, lithium chloride, activated carbon and zeolites [14]. The desiccant units currently used are based on five configurations: liquid spray towers, solid packed tower, rotating horizontal bed, multiple vertical bed and rotating desiccant wheel [15].

Most of the DEC systems available on the market use solid desiccant materials, especially for large scale installation (centralised operation coupled with air handling unit) and the most commonly used configurations include a rotary desiccant wheel.

The operating principle of a solid desiccant evaporative cooling system is depicted in figure 1 [7,12,16]. The dehumidifier of rotary wheel type is divided in two sections with air streams in counter-flow arrangement.

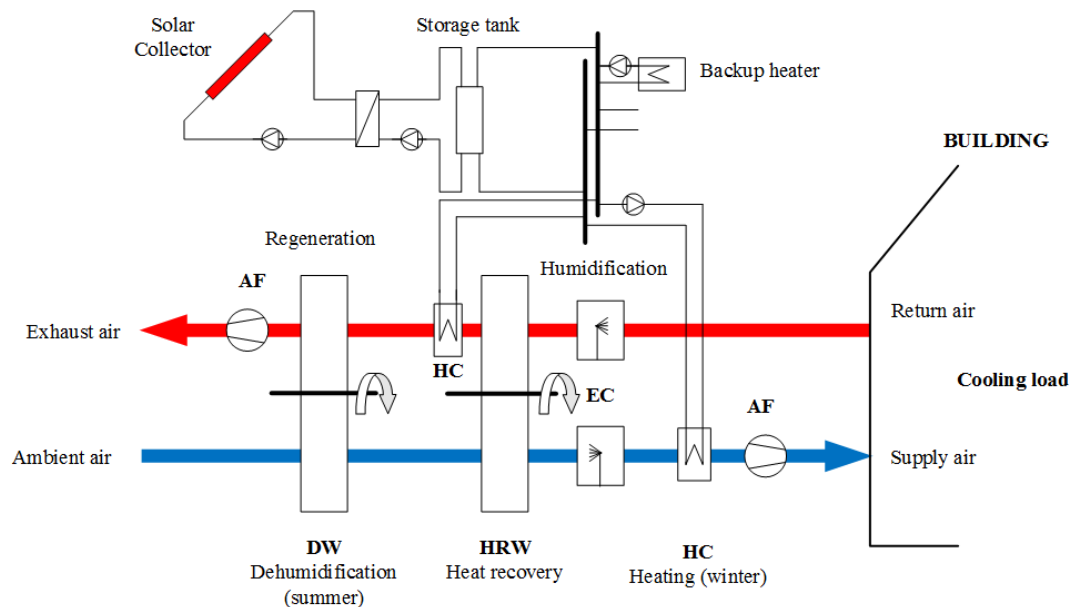


Figure 1. Basic configuration of the Desiccant and Evaporative Cooling (DEC) System
 AF – Air fan; HC – Heating coil; EC – Evaporative cooler; DW – Desiccant wheel; HRW – Heat recovery wheel.

In the supply air section, the solid desiccant partially adsorbs the moisture of the outdoor air. The released adsorption heat and the hot desiccant particles coming from the regeneration side increase the temperature of the air. Therefore, the air exits the process section of the rotary wheel hot and dry. Passing through the heat recovery wheel, a part of the added adsorption heat is rejected on the regeneration section where the desiccant material is reactivated. The evaporative cooling process reduces substantially the air temperature, while the humidity achieves the proper level for comfort requirements. On the regeneration section, the return air extracted from the building is successively cooled using evaporative cooling device and pre-heated by the heat recovery wheel. Finally, the solar collector provides the heat needed to regenerate the desiccant material in order to obtain a continuous operation. Heating coil (that is used only in the heating season) and backup heater are two other components that contribute to the extending of the operating range of desiccant system.

Rotary desiccant wheel is the key element of the desiccant cooling system which has a significant potential for improving the open-cycle performance and reducing the size and operating costs of the whole system [6,17]. Comprehensive studies related on the desiccant wheel for air conditioning applications have been carried out and the topic is still in the attention of the researchers around the world. These studies are focused mainly on two critical aspects: the development of new, advanced desiccant materials and the optimization of system configurations suitable for low grade heat sources, including solar energy [18]. The objectives pursued are the increase of competitiveness compared to the conventional cooling systems and the expansion of the current niche market.

3. Mathematical model for rotary desiccant dehumidifier

Mathematical modelling and numerical simulation are widely used for detailed analysis and investigation related to desiccant wheel design and operation. The advantages are obvious: it takes less time and needs lower funds than experimental methods for predicting the behaviour and the performance of a desiccant wheel; it can produce a large number of results, so it is very convenient to perform parametric study and optimization analysis; it can reveal some aspects in research areas that

are inaccessible to experimental approach (e.g. inside the desiccant particle) [17]. Consequently, plenty of mathematical models have been constructed based on the combined heat and mass transfer processes occurring within the desiccant wheel. The models can be divided in two main categories: gas-side resistance (GSR) models and gas and solid-side resistance (GSSR) models. In the GSR models, heat and mass transfer within desiccant particles is neglected, therefore “no intra-particle mass and temperature gradients” hypothesis is assume. In this case, convection is the dominant mode of heat and mass transfer from the air stream to the desiccant surface. GSSR models consider heat and mass transfer in both solid and air sides, and can well explain the actual transfer processes occurring in the desiccant particles, such as heat conduction and mass diffusion. A number of physical mechanisms are involved: surface reaction coupled with the adsorption process; inter-particle channel diffusion and adsorption within the layer; micro-pore and micro-pore diffusion and adsorption within solid phase [19]. The complexity and the precision of the governing equations is thus greatly increased.

Different configurations for desiccant dehumidifier have been modelled including solid packed bed, with axial and radial air flow, rotating horizontal bed, multiple vertical bed, rotating honeycomb or fluidized bed [20]. Many research studies were focused on heat and mass transfer processes in packed bed adsorber which is one of the most used models for dehumidification and cooling purposes. Several packing and desiccant solutions were developed and silica gel was among the most common materials due to their high affinity for water vapour. To improve the desiccant bed performance, some composite mixture formulas were proposed. So, Majumdar et al. [19] used in their study a mixture of silica gel particles and inert particles with different compositions and thermo-physical properties and Rady et al. [21] included in desiccant layer macro-encapsulated phase change materials to decrease the effect of heat of adsorption on the adsorption capability. A new composite desiccant was developed by Aristov et al. [22] by impregnating silica gel with calcium chloride in order to enhance drying capacity. Starting from the observation that the silica gel particle does not fully participate in adsorption process, Ramzy et al. [23] propose a composite formula obtained by coating an inert particle with a layer of silica gel.

To reduce the disadvantage of non-homogeneous operation of the vertical packed bed, Awad et al. [20] proposed a new bed configuration: hollow cylindrical dehumidifier with radial air flow.

Even if a large amount of works have been conducted on modelling and analysing solid packed bed adsorbers, further efforts are still needed. The study whose results are presented in this paper follows the same research direction. The overall goal of this study is to develop a theoretical model to analyze the heat and mass transfer processes in packed bed adsorber of rotary wheel type for air conditioning purposes. The model is a GSR ones and is intended to be a useful tool to assess the effect of design parameters and operating conditions on the performance of the dehumidifier.

3.1. Model assumptions

The mathematical model of heat and mass transfer processes manifesting within the packed bed absorber of rotary wheel type is based on the following assumptions: the rotary bed is adiabatic, that is why the effects of the heat transfer at the sides of the bed are neglected; heat transfer within the rotary bed takes place only through convection mode; conduction and radiation are neglected; temperature and humidity/water content gradients on radial direction are neglected; one adsorption component is only considered (water vapor); heat and mass transfer resistance is concentrated in the external film of the desiccant pellets that means no temperature and water content gradients exist within the pellets; air flows in axial direction; because of the relative small dimensions of the pellets (3 mm diameter) and the continuity of the interstitial spaces among the pellets, the changes of flow direction are neglected.

3.2. Governing equations

The equations of the mathematical model result from applying the mass and energy balances for both solid desiccant and moist air, to a fixed control volume in cylindrical coordinates (figure 2). The control volumes placed in different positions within the desiccant wheel form a two dimensional network of steady-state heat and mass exchangers within which there is a cross flow of humid air and

solid desiccant. Adsorption/desorption processes are followed in step-by-step procedure by successive movement of the control volume along axial direction (z) and angular/circumferential direction (θ).

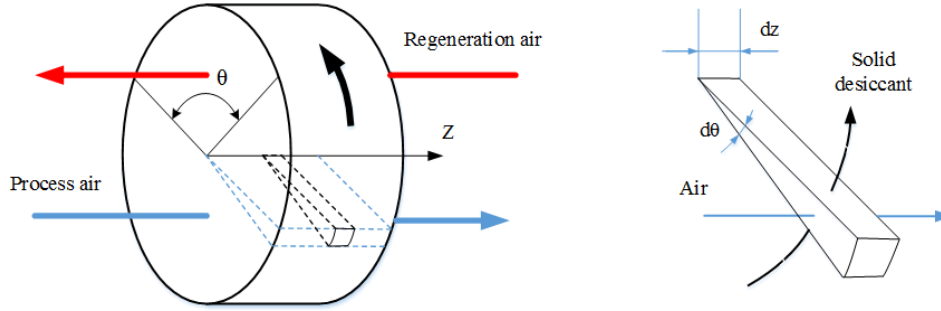


Figure 2. Control volume of the desiccant wheel.

Using the above mentioned assumptions, the four balance equations are as follows:

- gas phase energy balance

$$\frac{\partial T_g}{\partial z} = \frac{(A_v / F_s)(\rho_l / \rho_s)}{(c_{pa} + Y c_{pv}) \rho_g u_g} \{h_c (T_s - T_g)\} \quad (1)$$

- solid phase energy balance

$$\frac{\partial T_s}{\partial \theta} = \frac{1}{c_{ps} + W c_w} \frac{A_v}{\omega \rho_s} K_y (Y - Y^*) \cdot [h_{fg} + q_B - c_{pv} (T_s - T_g)] - \frac{1}{c_{ps} + W c_w} \frac{A_v}{\omega \rho_s} h_c (T_s - T_g) \quad (2)$$

- gas phase moisture balance

$$\frac{\partial Y}{\partial z} = - \frac{K_y}{\rho_g u_g} \cdot \frac{A_v}{F_s} \cdot \frac{\rho_l}{\rho_s} (Y - Y^*) \quad (3)$$

- solid phase moisture balance

$$\frac{\partial W}{\partial \theta} = \frac{K_y A_v}{\omega \rho_s} (Y - Y^*) \quad (4)$$

The nomenclature used in governing equations is presented below: T_g , T_s - gas and solid desiccant temperature [K]; Y - humidity ratio [kg/kg]; Y^* - humidity ratio in equilibrium with desiccant [kg/kg]; W - water content of solid desiccant [kg/kg]; h_c - gas-side convective heat transfer coefficient [W/m²K]; K_y - gas-side mass transfer coefficient [kg/m²s]; c_{pa} , c_{ps} , c_{pv} , c_w - specific heat of dry air, solid desiccant, water vapor and water [J/kg K]; ρ_g - gas (air) density (kg/m³), ρ_l - bulk density of packed bed [kg/m³]; ρ_s - density of desiccant particle [kg/m³]; A_v - transfer area per unit volume [m²/m³]; F_s - ratio of free flow area to cross section area of rotary dehumidifier; h_{fg} - latent heat of vaporization [J/kg]; q_B - formation enthalpy [J/kg]; u_g - axial air velocity [m/s]; ω - angular velocity [rad/s].

3.3. Auxiliary relations

Heat and mass transfer coefficients for the gas side were calculated using Gnielinski's relations [24] based on the local conditions/parameters.

$$Nu = \frac{h_c \cdot d_p}{k_g} = \sqrt{Nu_{lam}^2 + Nu_{turb}^2} \quad (5)$$

$$Nu_{lam} = 0.664 \cdot Re^{0.5} \cdot Pr^{0.33} \quad (6)$$

$$Nu_{turb} = \frac{0.037 \cdot Re^{0.8} \cdot Pr}{1 + 2.443 \cdot Re^{-0.1} \cdot (Pr^{0.67} - 1)} \quad (7)$$

where Nu , Re and Pr are Nusselt, Reynolds and Prandtl numbers, d_p is diameter of the desiccant particle [m], k_g is air thermal conductivity [W/m²K];

$$Sh = \frac{K_y \cdot d_p}{\rho_g \cdot D_{12}} = \sqrt{Sh_{lam}^2 + Sh_{turb}^2} \quad (8)$$

$$Sh_{lam} = 0.664 \cdot Re^{0.5} \cdot Sc^{0.33} \quad (9)$$

$$Sh_{turb} = \frac{0.037 \cdot Re^{0.8} \cdot Sc}{1 + 2.443 \cdot Re^{-0.1} \cdot (Sc^{0.67} - 1)}^{24} \quad (10)$$

where Sh and Sc are Sherwood and Schmidt numbers, D_{12} is mass diffusivity of water vapor in the air [m²/s].

$$D_{12} = 1.97 \cdot 10^{-5} \cdot \frac{p_0}{p} \cdot \left(\frac{T}{T_0} \right)^{1.685} ; \quad p_0 = 1,013 \cdot 10^5 \left[\frac{N}{m^2} \right] ; \quad T_0 = 256 [K] \quad (11)$$

The thermo-physical properties of air and solid desiccant (specific heat, viscosity, thermal conductivity) included in governing equations were calculated in terms of local temperature and humidity/water content. The axial air velocity depends on the local temperature and pressure and it's calculated in the hypothesis of a constant mass flow rate.

Pressure drop through the bed was calculated with well-known Ergun equation:

$$\frac{\Delta p}{dz} = 150 \cdot \frac{(1 - \varepsilon)^2}{\varepsilon^3} \cdot \frac{\mu_g \cdot u_g}{d_p^2} + 1.75 \cdot \frac{(1 - \varepsilon)}{\varepsilon^3} \cdot \frac{\rho_g \cdot u_g^2}{d_p} \quad (12)$$

where dz is the thickness of each control volume [m], μ_g is dynamic viscosity of air [N·s/m²] and ε is the bed porosity.

Overall pressure drop can be calculated by summing up the pressure drops of all control volumes in axial direction.

The equilibrium relative humidity on the surface of silica gel particle is given by the following formula [24]:

$$\Phi^* = 2490 \cdot W^5 - 1708 \cdot W^4 + 409.6 \cdot W^3 - 42.4 \cdot W^2 + 3.64 \cdot W - 0.0079. \quad (13)$$

The relationship between the humidity ratio and the relative humidity is described as:

$$Y^* = 0.622 \frac{\Phi^* \cdot p_{vs}}{p - \Phi^* \cdot p_{vs}}, \quad (14)$$

where p is air pressure and p_{vs} is saturation pressure of water vapour,

$$p_{vs} = \exp(20.896 - 5204.0093 / T_s) \text{ [mbar]}. \quad (15)$$

The differential adsorption heat released in the sorption process is computed using the following algorithm:

$$q_{ad} = h_{lg} + q_B ; \quad h_{lg} = h_{lg0} - (c_w - c_{pv})(T_g - 273) ; \quad q_B = \frac{\varepsilon_P}{M}, \quad (16)$$

where ε_P is Polanyi potential [kJ/kmol] for couple silica gel-water [25], q_B is formation enthalpy and M is molecular weight of adsorbed molecules.

$$\varepsilon_p = \frac{655.2}{W + 0.052} - 450. \quad (17)$$

3.4. Boundary conditions

The mathematical model consists of governing equations (1), (2), (3), (4) and equilibrium equation (13) and there are four unknown variables: T_g , T_s , Y and W . This model has been solved numerically using finite difference scheme and the following boundary conditions were considered:

- for process air, $\theta \in (0, \theta_{ads}) \rightarrow T_g(0, \theta) = T_{g0}$ and $Y(0, \theta) = Y_0$;
- for regeneration air, $\theta \in (\theta_{ads}, 2\pi) \rightarrow T_g(0, \theta) = T_{g0,reg}$ and $Y(0, \theta) = Y_{0,reg}$;
- for desiccant, at inlet on the adsorption stage (first rotation), $z \in (0, L) \rightarrow T_s(z, 0) = T_{s0}$ and $W(z, 0) = W_0$.

Temperature and water content of silica gel on the inlet of the regeneration sector were provided from the last computational step of the previous adsorption stage.

4. Results and discussion

The governing equations for the proposed gas-side resistance model are solved using forward scheme finite difference method with prescribed boundary conditions as given by the previous equations. A grid size of 0.1 mm in axial direction and 0.05 degrees in circumferential direction is used to ensure numerical stability and accuracy.

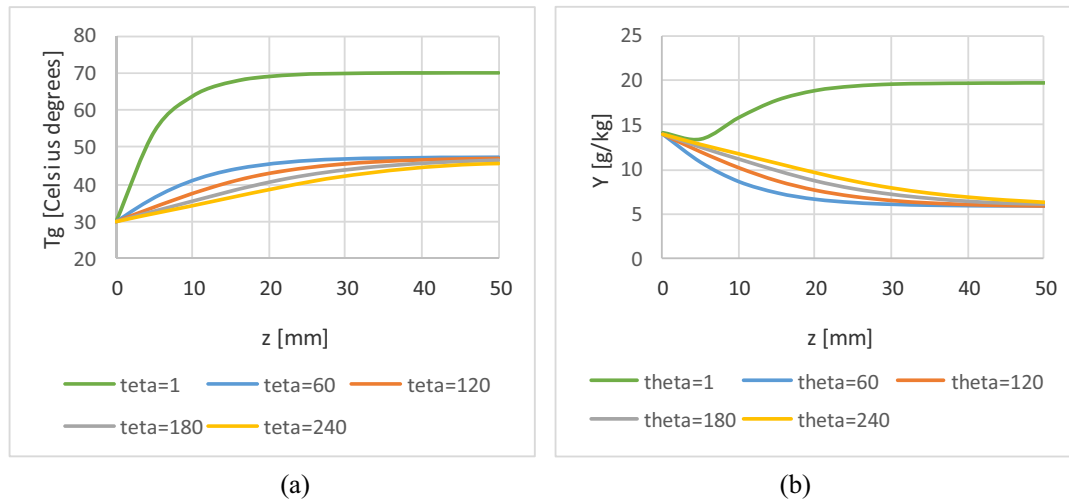
The input data that are used for the simulation are listed in table 1.

Table 1. Operational and design conditions.

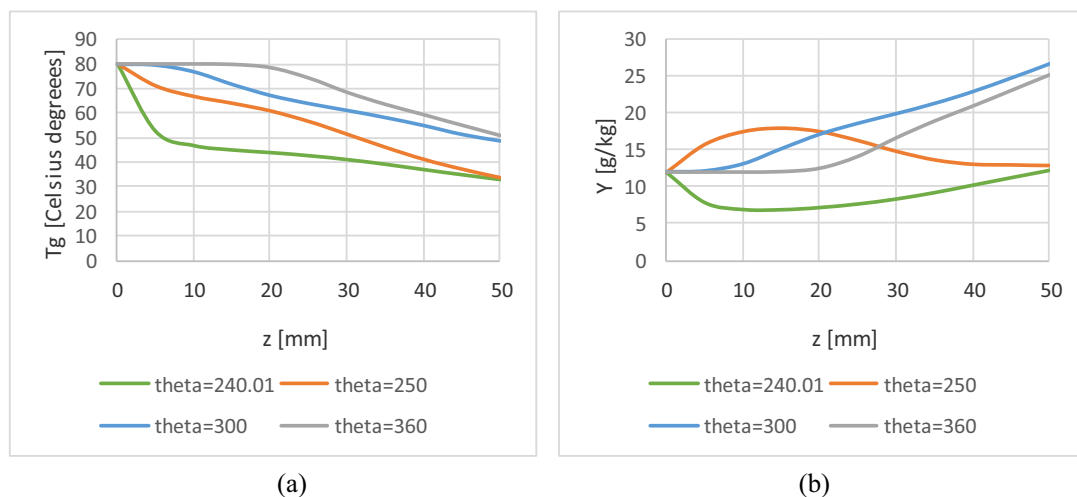
Y_0	Inlet humidity ratio of supply air [g/kg]	14
T_{g0}	Inlet temperature of supply air [°C]	30
$Y_{0,reg}$	Inlet humidity ratio of regeneration air [g/kg]	12
$T_{g0,reg}$	Inlet temperature of supply air [°C]	80
T_{s0}	Inlet temperature of solid desiccant [°C]	70
W_0	Inlet water content of solid desiccant [g/kg]	40
d_p	Desiccant particle diameter [mm]	3
c_{ps}	Isobaric specific heat capacity of solid desiccant [J/kgK]	921
u_{g0}	Inlet velocity of supply/regeneration air [m/s]	0.9
ω	Rotational speed [rph]	5
R	Bed/Desiccant wheel radius [mm]	100
L	Bed/Desiccant wheel thickness [mm]	50
θ_{ads}	Supply air sector [degrees]	240

The output of the theoretical model is, mainly, the desiccant parameters (water content and temperature) and the air conditions (humidity ratio and temperature). Using these properties, the performance of the desiccant wheel can be evaluated and an up-close look into the mass and transfer processes can be conducted.

Figure 3 and figure 4 depict the air temperature and humidity ratio profiles along the desiccant bed for different positions on the process and regeneration sectors. There is a certain instability that manifests itself when passing from one sector to another. Thus, the evolution in the process sector begins with a brief desorption (figure 3b) which takes place along the whole thickness of the desiccant bed. The phenomenon is determined by the relative high temperatures within the layer (figure 3a). As the angular position in the adsorption sector changes, the drying ability of the desiccant material is regained in the profoundness of the bed.



(a) (b)
Figure 3. Air temperature and humidity ratio profiles – adsorption/process stage.



(a) (b)
Figure 4. Air temperature and humidity ratio profiles – desorption/regeneration stage.

The initial reversed process (of adsorption type) is also revealed in the regeneration sector, but it is restricted only to the first 10 mm in thickness, in airflow direction (figure 4 b). The dynamics of the transfer processes depends on the axial position within the rotary bed and the angular/circumferential position on the regeneration sector. Locations where desiccant material is already regenerated (temperature and humidity ratio of the regeneration air are constant) are displayed in figures 4a and 4b.

Figures 5a and 5b show the water content and temperature profiles obtained for the counter flow desiccant wheel at different positions within the rotary bed/layer.

High-intensity heat transfer prior to the dehumidifier entry into the operating regime (adsorption or desorption) as well as inactivity ranges of the desiccant material due to reaching the equilibrium humidity at the gas-solid interface are clearly highlighted in these diagrams. As previously mentioned, the dynamics of the transfer processes depends on the axial position, z .

The numerical results obtained for rotary layer can be translated for the similar fixed layer using the following correlation: $\theta = \omega \cdot \tau$, where τ is time passed from the beginning of the adsorption/desorption operations.

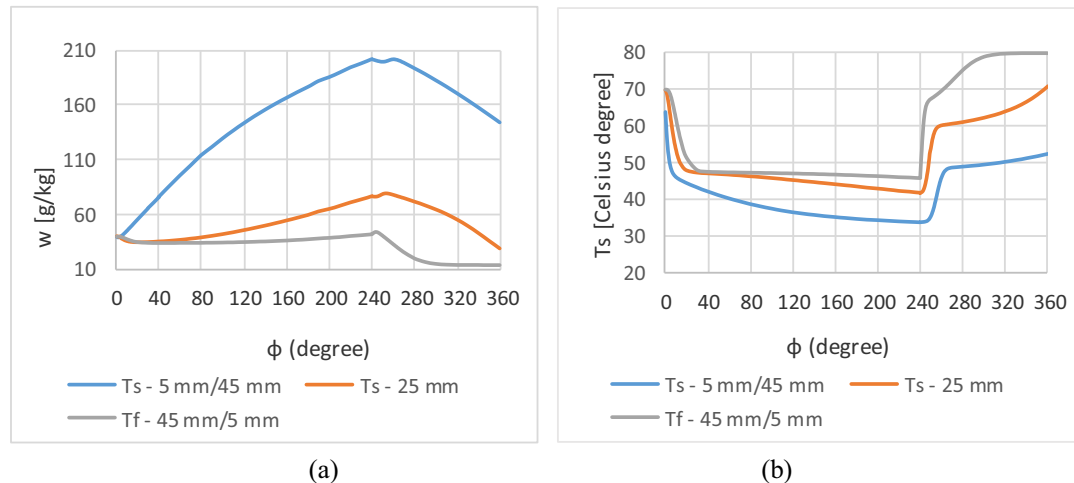


Figure 5. Solid temperature and water content profiles.

5. Conclusions

In the present paper a new one-dimensional mathematical model that describes the combined heat and mass transfer processes within the packed bed dehumidifier is proposed. The model is of gas side resistance type and provides an effective tool for analysing the performance of rotary or fixed dehumidifier.

To improve the accuracy of the numerical results, the thermo-physical properties of air and solid desiccant (specific heat, viscosity, thermal conductivity) were evaluated in terms of local temperature and humidity/water content. Also, the local axial air velocity and the pressure drop across the layer were considered.

The present study focuses on the evolutions of the air and solid desiccant parameters (temperature and humidity ratio/water content) in axial direction, at different angular positions in both process and regeneration sectors and in circumferential direction, at different profoundness within the rotary layer.

The mathematical model can generate a large number of numerical results that can be used to analyze the impact of design and operating conditions on the desiccant wheel performance. Air streams temperature, humidity and velocity, type of desiccant material (in homogeneous and combined bed), desiccant wheel thickness and radius, rotational speed, angular size of process and regeneration sectors are the most important analysis criteria. Finally, an optimized desiccant wheel in terms of moisture removal capacity, heat supplied for the regeneration stage and pressure drop across the rotary layer can be obtained.

6. References

- [1] European Commission 2016 Communication on an EU Strategy on Heating and Cooling, COM(2016)51
- [2] European Commission 2016 Overview of support activities and projects of the European Union on energy efficiency and renewable energy in the heating and cooling sector
- [3] Yang L, Yan H and Lam, J 2014 Thermal comfort and building energy consumption implications - A review *Applied Energy* **115** pp 164-173
- [4] REN 21 2018 Renewables 2018: Global status report
- [5] Lundgren K, Kjellstrom T 2013 Sustainability Challenges from Climate Change and Air Conditioning Use in Urban Areas *Sustainability* **5** pp 3116-3128
- [6] Jani, D B, Mishra, M and Sahoo, P K 2016 Solid desiccant air conditioning - A state of the art review *Renewable and Sustainable Energy Reviews* **60** pp 1451-1469
- [7] Henning H M 2007 Solar assisted air conditioning of buildings – an overview *Applied Thermal*

- Engineering* **27** pp 1734-1749
- [8] Fan Y, Luo L and Souyri B 2007 Review of solar sorption refrigeration technologies: Development and applications *Renewable and Sustainable Energy Reviews* **11** pp 1758-1775
 - [9] IEA, Solar Heating and Cooling Programme. Task 53, New Generation Solar Cooling & Heating Systems, 2014–2018 (<http://task53.iea-shc.org/>)
 - [10] IEA 2012 Technology Roadmap: Solar Heating and Cooling
 - [11] Prieto A, Knaack U, Auer T and Klein T 2017 Solar coolfacades: Framework for the integration of solar cooling technologies in the building envelope *Energy* **137** pp 353-368
 - [12] Ghafoor A, Munir A 2015 Worldwide overview of solar thermal cooling technologies, *Renewable and Sustainable Energy Reviews* **44** pp 763-774
 - [13] Ma Y, Saha S, Miller W, Guan L 2017 Comparison of different solar-assisted air-conditioning systems for Australian office buildings *Energies* **10** pp 1-27
 - [14] Rafique M, Gandhidasan P, Rehman S and Al-Hadhrami L 2015 A review on desiccant based evaporative cooling systems, *Renewable and Sustainable Energy Reviews* **45** pp 145-159
 - [15] Baniyounes A M, Ghadi Y Y, Rasul M G and Khan M K 2013 An overview of solar assisted air conditioning in Queensland's subtropical regions, Australia *Renewable and Sustainable Energy Reviews* **26** pp 781–804
 - [16] Allouhi A and van Vuuren D P 2015 Solar driven cooling systems: An update review, *Renewable and Sustainable Energy Reviews* **44** pp 159-181
 - [17] Ge T S, Li Y, Wang R Z and Dai Y J 2008 A review of the mathematical models for predicting rotary desiccant wheel *Renewable and Sustainable Energy Reviews* **12** pp 1485-1528
 - [18] La D, Dai Y J, Li Y, Wang T S and Ge T S 2010 Technical development of rotary desiccant dehumidification and air conditioning: A review *Renewable and Sustainable Energy Reviews* **14** pp 130-147
 - [19] Majumdar P and Worek W M 1989 Combined heat and mass transfer in a porous adsorbent *Energy* **14**(3) pp 161-175
 - [20] Awad M M, Ramzy K, Hamed A M and Bekheit M M 2008 Theoretical and experimental investigation on the radial flow desiccant dehumidifier bed *Applied Thermal Engineering* **28** pp 75-85
 - [21] Rady M A, Huzayyin A S, Arquís E, Monneyron P, Lebot C and Palomob E 2009 Study of heat and mass transfer in a dehumidifying desiccant bed with macro-encapsulated phase change materials *Renewable Energy* **34** pp 718-726
 - [22] Aristov Yu I, Parmon V N, Restuccia G and Cacciola G 2002 A family of new working materials for solid sorption air conditioning systems *Applied Thermal Engineering* **22** pp 191-204
 - [23] Ramzy K, Kadoli R and Ashok Babu T P 2011 Improved utilization of desiccant material in packed bed dehumidifier using composite particles *Renewable Energy* **36** pp 732-742
 - [24] Kravchik T, Korin E and Borde I 1990 Influence of material properties and heat removal on mass and heat transfer in a solid desiccant dehumidifier *Chemical Engineering Processes* **27** pp 19-25
 - [25] Kanoldt W W 1992 *Nicht-isotherme Adsorption am Adorbenseinzelkorn*, Dissertation (TU Munchen)

Acknowledgments

This work was partly supported by the project POSCCE-A2-O2.2.1-2009-4-ENERED, ID nr. 911, co-financed by the European Social Fund within the Sectoral Operational Program “Increase of Economic Competitiveness”.