

Preliminary experimental and numerical analysis of a silica gel packed bed humidification system

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Abstract. In this research, an innovative air humidification system based on two packed beds made of silica gel spherical particles is developed. Each bed is alternatively crossed by two airflows (regeneration and process). The first air stream, at outdoor conditions, after being heated and humidified through the sorption material, is supplied to the building. The second one, also at outdoor conditions, provides water vapour to the desiccant packed bed. The system design and optimization is carried out through a numerical and experimental approach. The developed phenomenological model is based on the Pseudo Gas-side Controlled (PGC) method, considering only the gas side resistance and, therefore, assuming uniform water content and temperature distribution within the desiccant particles. A test rig to evaluate performance of the packed bed humidification system has been realized and experimental results have been used to validate the model. Obtained performance highlights that the proposed system can provide adequate humidification and that it can be a valid alternative to conventional adiabatic and steam humidifiers.

1 Introduction

Air humidification during cold climates is a crucial point to achieve occupants' thermal comfort in buildings. Two types of humidification technologies are commonly adopted: adiabatic humidifiers and steam humidifiers. Although the former is particularly diffused due to its simplicity, efficiency and reduced costs, it cannot be integrated in several applications, such as in case of hospitals, because of the risk of bacterial growth due to the presence of liquid water. This issue is solved using steam humidifiers, which however require a high temperature heat source or electricity.

At present there is a great interest in innovative humidification systems based on sorption materials, which can at the same time minimize air contamination risk, due to the absence of liquid water, and exploit low temperature heat sources. La et al. [1] investigated the heating-humidification performance of a solar heating and humidification system with a one-rotor two-stage desiccant air-conditioning unit and estimated the optimal solar collector area. Solar heating with desiccant humidification could improve the indoor thermal comfort significantly, though in this case return air was used as humidification source.

Wada et al. [2] developed a prototype desiccant humidity control system that makes use of the waste heat generated by a water source heat pump. It was outlined that humidification without the use of water was feasible, even in the absence of an indoor latent heat load, when the outdoor humidity ratio was equal to 3.5 g/kg or more.

De Antonellis et al. [3] studied the humidification process for a hospital application during winter period through an experimental and numerical investigation. Water vapour is adsorbed from outdoor air and it is released through a desiccant wheel to the supply air.

Kawamoto et al. [4] experimentally investigated a desiccant based Air Handling Unit (AHU) for humidification purpose during winter period, showing that it is possible to increase the outdoor humidity ratio from 1.8 - 2.3 g/kg to 5.8 g/kg.

In this paper, an innovative air humidification system based on two packed beds of silica gel spherical particles is analysed through an experimental and numerical approach. This technology is particularly interesting because it exploits desiccant materials, which can adsorb water vapour from outdoor air and release it to the air stream supplied to the building. In this way, this system is suitable for hospital applications, because the presence of water droplets is avoided and there is not contamination with exhausted air from the building. Furthermore, it can exploit low temperature heat sources with primary energy savings. Compared to other technologies based on desiccant materials, such as desiccant wheels, packed beds do not present moving parts besides valves, and the investment cost is lower since typical sorption materials, such as silica gel, are widely available and cheap. The main disadvantage of this technology is that the humidification process becomes difficult in case of low outdoor humidity ratio and high outdoor temperature.

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2 Description of the packed bed humidification system

In Fig. 1 (a), it is shown the proposed humidification system. Conventional terminology used in Desiccant Evaporative Cooling (DEC) cycles is applied for the investigated humidification device: process air is heated and dehumidified (red line), while regeneration air is cooled and humidified (blue line). Therefore, according to this adopted nomenclature, the air stream supplied to the building is the regeneration one.

The system consists of the following components:

- Two silica gel packed beds.
- A heating coil.
- Two fans.

Figure 1 illustrates the processes undergone by the two air streams, which are both assumed initially at outdoor conditions. Process air provides water vapour to the bed, in order to obtain adequate humidification of the regeneration air stream. Process airflow is dehumidified and exhausted to the atmosphere. On the other side, the regeneration stream is first heated by a heating coil, then it passes through the bed, where it is cooled and humidified, and, finally, it is supplied to the building (if necessary after a further heating process to reach adequate temperature). The two flows are periodically inverted through the valves represented in Fig. 1, in order to continuously provide a humidified air stream (Fig. 1 (b)).

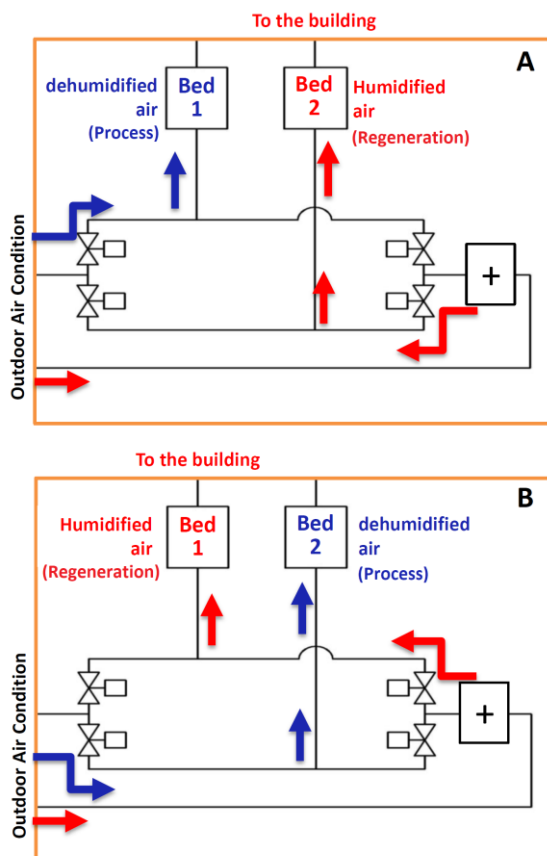


Fig. 1. Scheme of the proposed humidification system.

3 Experimental setup and methodology

The test rig mainly consists of two Air Handling Units (AHUs), two silica gel packed beds and four commutation valves. The AHUs control the inlet airflows temperature and humidity ratio through heating coils, cooling coils and adiabatic humidifiers. In order to reach high temperatures, in the regeneration side it is installed an additional electrical heater. Air mass flow rates are controlled by variable speed fans. Further details of the test rig are available in a previous work of the authors [3].

Packed beds inlet and outlet air conditions are measured with coupled PT100 Class A thermo-resistances and relative humidity capacitive sensors (± 0.2 °C and 1% at 20 °C). Airflow rates are measured through orifices plates and piezo-resistive pressure transmitters [6].

Finally, main characteristics of packed beds are hereinafter reported:

- Bed length: 4 cm.
- Bed diameter: 25 cm.
- Adsorption material type: silica gel RD.
- Average particle diameter: 4.2 mm.

The test facility is used to carry out experimental measurement in representative humidification conditions, in order to validate the model describe in Section 4.

Finally, water adsorption isotherm of silica gel RD has been measured by a gravimetric adsorption apparatus (Aquadyne DVS).

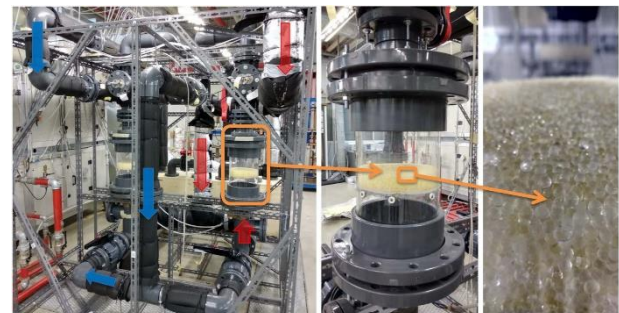


Fig. 2. Picture of the experimental setup and detail of the silica gel packed bed.

4 Model description

The present study has been carried out through a Pseudo Gas-Side Controlled (PGC) model based on the work of Pesaran and Mills [5]. In the developed phenomenological PGC model only the gas side resistance is considered: therefore, uniform water content and temperature distribution within desiccant particles are assumed.

Main adopted assumptions are:

- i. Uniform air flow along axial direction.
- ii. Heat and mass transfer only due to forced convection.
- iii. Negligible axial heat conduction along the bed.
- iv. No heat losses to the surrounding.

- v. Uniform temperature and water content within desiccant particles.
- vi. Stationary wet air in the silica gel pores in local equilibrium with the adsorbed phase.
- vii. No hysteresis effects in the desiccant material.

The following equations are applied to a longitudinal element of the bed:

$$\dot{m}_a c_{p,a} \frac{\partial T_a}{\partial z} = P[h_c + h_m(X_s - X_a)](T_a - T_s) \quad (1)$$

$$(\rho c_p A)_{bed} \frac{\partial T_s}{\partial t} = P[h_c(T_a - T_s) - h_m(X_s - X_a)H_{ads}] \quad (2)$$

$$\dot{m}_a \frac{\partial X_a}{\partial z} = P h_m (X_s - X_a)(1 - X_s) \quad (3)$$

$$(\rho A)_{bed} \frac{\partial W}{\partial t} = P h_m (X_s - X_a) \quad (4)$$

Where specific quantities are referred to humid air mass, according to the approach adopted in the reference study [5].

Heat and mass transfer coefficients are related in the following form:

$$h_c = Le h_m c_{p,a} \quad (5)$$

With h_c calculated as [4]:

$$h_c = 0.683 G_a Re_p^{-0.51} c_{p,a} \quad (6)$$

Boundary conditions are referred to inlet air states in the following way: $X(z=0, \tau) = X_{a,in}$ and $T_a(z=0, \tau) = T_{a,in}$. The initial conditions are referred to the initial state of the packed bed, hence $W(z, t=0) = W_0$ and $T_s(z, t=0) = T_{s,0}$. Finally, air outlet conditions are: $T_a(z=L, \tau) = T_{a,out}$ and $X(z=L, \tau) = X_{a,out}$. Other correlations and thermo-physical properties required to solve the equations are reported in literature [4].

The differential equations (1) – (4) are coupled with the adsorption isotherm of adopted silica gel. The equilibrium isotherm is experimentally obtained measuring the uptake curves of two desiccant samples at 40°C. Obtained data are fitted with the polynomial equation:

$$RH(W) = -0.07 + 5.81 W - 87.49 W^2 + 823.36 W^3 - 3230.41 W^4 + 4436.70 W^5 \quad (7)$$

The pressure drop along each bed is calculated through Ergun's correlation:

$$\frac{\Delta p}{L} = 150 \frac{\rho_a v(1-\varepsilon)^2 v}{\varepsilon^2 d_p^2} + 1.75 \frac{1-\varepsilon}{\varepsilon^3} \frac{\rho_a}{d_p} v^2 \quad (8)$$

Note that in this study, the average RD silica gel grain diameter d_p is 4.2 mm and the bed void fraction is $\varepsilon = 0.372$.

Finally, the system performance is evaluated using the conventional humidity ratio x referred to the dry air mass, which is related to the one based on total mass (humid air) as:

$$x = X/(1 - X) \quad (9)$$

In addition, average regeneration air temperature and humidity ratio variations are calculated as:

$$\Delta T_{reg} = \bar{T}_{reg,out} - T_{reg,in} \quad (10)$$

$$\Delta x_{reg} = \bar{x}_{reg,out} - x_{reg,in} \quad (11)$$

Where $\bar{T}_{reg,out}$ and $\bar{x}_{reg,out}$ are respectively the outlet average regeneration air temperature and humidity ratio (based on dry mass) during the half cycle period.

5 Results and discussion

After implementation and calibration of the numerical model, simulation results have been validated with experimental data. Secondly, the model has been used to analyse and design the humidification system.

5.1 Model validation

After model calibration, simulation results have been compared with further experimental data collected during two complete humidification - dehumidification cycles (Tab. 1).

Table 1. Test conditions adopted in the model validation

Half cycle duration	τ	1800 s
Bed length	L	4 cm
Inlet regeneration air temperature	$T_{reg,in}$	54 °C
Inlet process air temperature	$T_{pro,in}$	27 °C
Inlet regeneration air humidity ratio	$x_{reg,in}$	5.8 g/kg
Inlet process air humidity ratio	$x_{pro,in}$	14.5 g/kg
Inlet regeneration air velocity	$v_{reg,in}$	0.55 m/s
Inlet process air velocity	$v_{pro,in}$	0.55 m/s

As shown in Fig. 3, numerical results of the calibrated model are in very good agreement with experimental data. In particular, $x_{reg,out}$ values predicted with the PGC model are very close to the ones obtained with experimental tests. Instead, $T_{reg,out}$ is slightly overestimated (up to 5°C) by the model mainly due heat losses occurring in the rig during experimental tests.

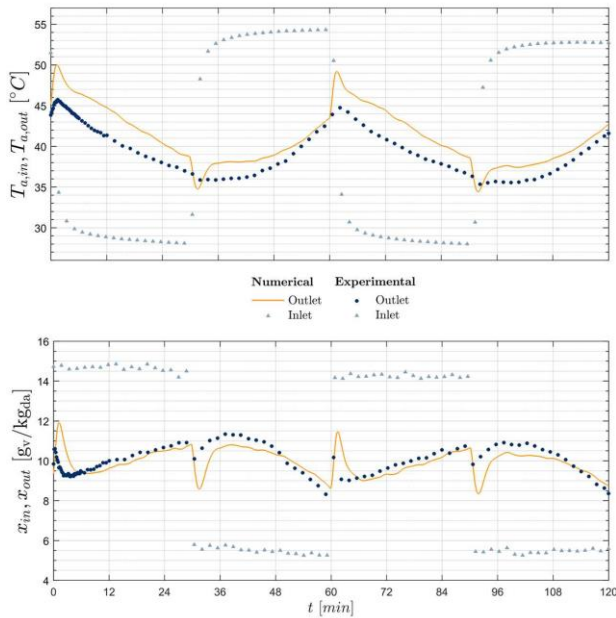


Fig. 3. Comparison between experimental data and numerical results (test conditions of Tab. 1).

5.2 Humidification system design

In this section appropriate bed length and regeneration airflow velocity are evaluated. The following outdoor air winter conditions, which are representative of Southern Europe regions, have been considered:

- $T_e = T_{pro,in} = 0^\circ\text{C}$.
- $x_e = x_{pro,in} = x_{reg,in} = 3.4 \text{ g/kg}$.

In addition, it is assumed the humidification system must supply fresh air to the building (i.e. a hospital), ambient conditions being, according to technical guidelines UNI 10339 [7]:

- $T_{amb} = 22^\circ\text{C}$.
- $x_{amb} = 6.6 \text{ g/kg}$.

Negligible latent load is assumed, hence the humidity ratio of the supplied air stream is:

- $x_s = x_{reg,out} = 6.6 \text{ g/kg}$.

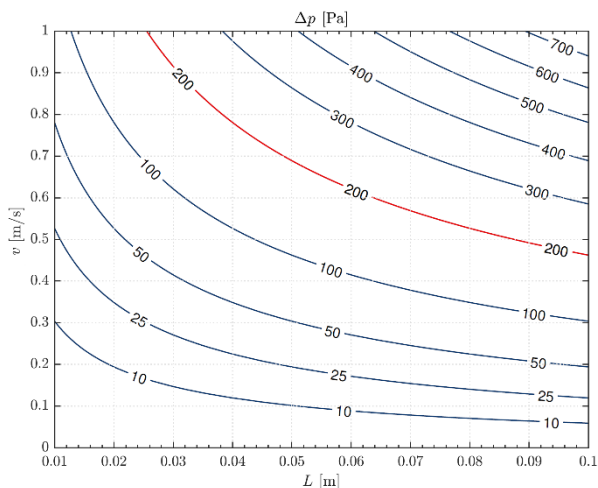


Fig. 4. Effect of velocity and bed length on pressure drop.

Consequently, the variation of humidity ratio of the regeneration airflow, which is assumed to be delivered to the building, is $\Delta x_{reg,out} = 3.2 \text{ g/kg}$.

Further constraints are set to design the humidification system: the maximum pressure drop of each bed is set to 200 Pa, since this is a typical value for desiccant wheels and heat exchangers. It is evaluated with Ergun's Equation (8), as shown in Fig. 4.

From preliminary sensitivity analysis, the effect of temperature and velocity variations upon outlet average regeneration humidity ratio has been evaluated. It is outlined that the system could reach the desired humidity ratio only if $v_{reg} < v_{pro}$. In order to reduce the degrees of freedom in the system design process, regeneration air velocity v_{reg} has been set constant and equal to 0.5 m/s.

The design process is an iterative procedure used in order to individuate the optimal configuration of the humidification system. The numerical simulations are performed varying process air velocity, regeneration temperature, bed length and commutation time providing the desired humidification capacity ($\Delta x_{reg,out} = 3.2 \text{ g/kg}$). Due to the high number of degrees of freedom, the following simplified procedure has been adopted:

1. $x_{pro,in}$, $x_{reg,in}$, $x_{reg,out}$ and $T_{pro,in}$ are fixed, according to aforementioned boundary conditions.
2. $T_{reg,in}$ is alternatively set to the following values: 30, 35 and 40°C .

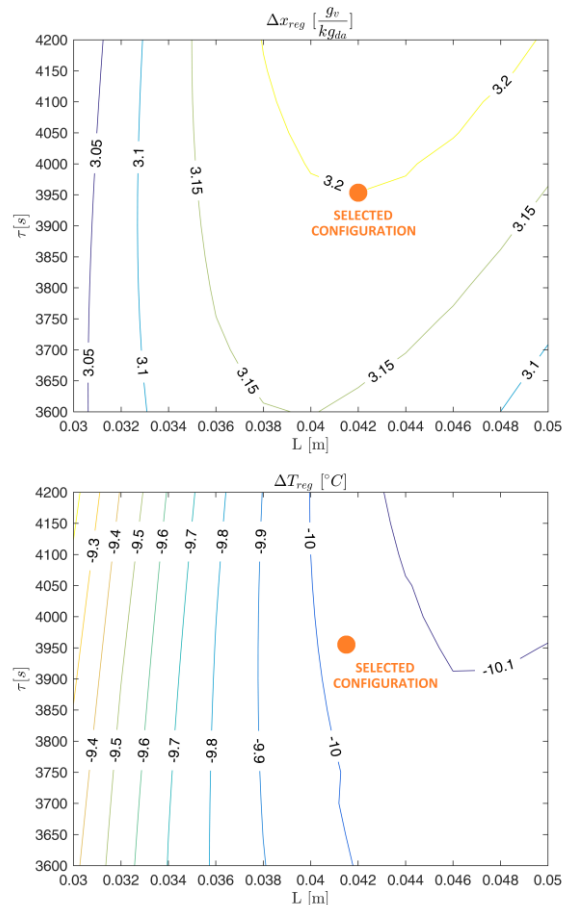


Fig. 5. Design of the humidification system: selected configuration ($T_{reg,in} = 35^\circ\text{C}$, $v_{pro} = 0.75 \text{ m/s}$).

3. v_{pro} is alternatively set to the following values: 0.5, 0.75 and 1 m/s.
4. The humidity ratio difference $\Delta X_{reg,out}$ is calculated as a function of L and τ . The iso-humidity line representing the desired $\Delta X_{reg,out}$ is plotted and the corresponding time τ is found.

Among these nine different main configurations (arising from the three different regeneration air temperatures and the three different process air velocities), only four configurations satisfy the pressure loss constraints (lower than 200 Pa). The configuration with the thicker bed, lower regeneration temperature and velocity is chosen. Therefore, as shown in Fig. 5, the optimal configuration is:

- $T_{reg,in} = 35^\circ\text{C}$.
- $v_{pro} = 0.75$ m/s.
- $L = 4.2$ cm.
- $\tau = 3950$ s.

5.3 Numerical analysis

In this section, the effect of the variation of process and regeneration inlet air conditions is evaluated. Humidification system bed length and half-cycle time are kept constant and equal to the values obtained at the end of the design process of section 5.2.

In all cases it is assumed that the inlet humidity ratio of process and regeneration airflows is equal to the one of outdoor air ($X_{in} = X_{pro,in} = X_{reg,in} = X_e$).

In Fig. 6, ΔT_{reg} and ΔX_{reg} are reported for different values of x_{in} and $T_{pro,in}$ at constant regeneration temperature ($T_{reg,in} = 35^\circ\text{C}$). The higher X_{in} , the higher ΔX_{reg} mainly due to the increase in the inlet relative humidity of the process airflow that enhances the water vapour adsorption process [8]. Similarly, the higher $T_{pro,in}$, the lower ΔX_{reg} because of the decrease in $RH_{pro,in}$. This effect should be properly considered in actual applications operating at outdoor temperature higher than the one adopted in the design process. In fact, in this case $\bar{x}_{reg,out}$ is limited and the humidification capacity decreases. Regarding ΔT_{reg} , it is shown that the regeneration air temperature reduction is mainly related to the released heat of adsorption. In fact, at a given ΔX_{reg} , ΔT_{reg} is only slightly affected by process air inlet temperature.

Finally, in Fig. 7 the effect of the variation of regeneration temperature and outdoor humidity ratio on ΔX_{reg} and ΔT_{reg} is investigated. An increase in $T_{reg,in}$ leads to an increase in the device humidification - dehumidification capacity and, consequently, in the temperature variation. This effect is mainly due to the lower relative humidity of the regeneration airflow. Therefore, the variation of $T_{reg,in}$ can be an effective way

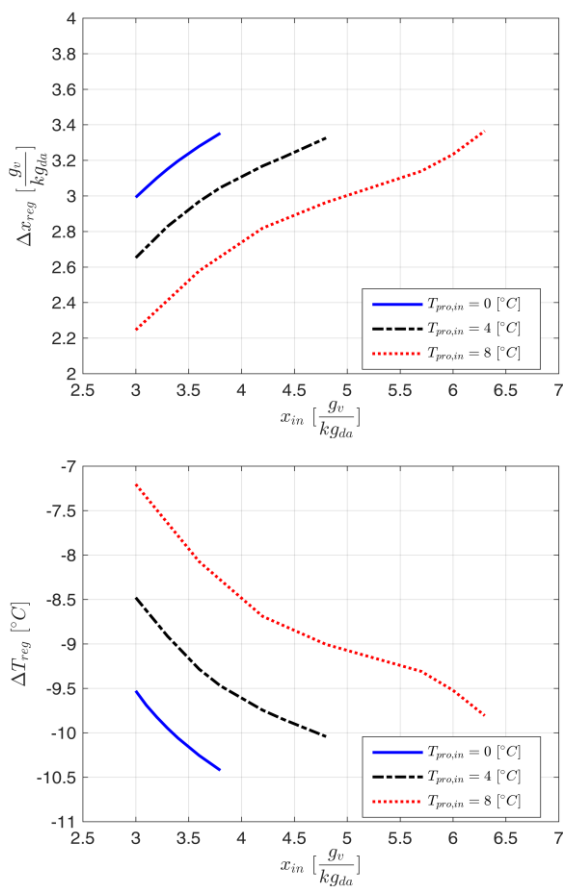


Fig. 6. Off-design performance of the humidification system (Selected configuration operating at $T_{reg,in} = 35^\circ\text{C}$, $v_{pro} = 0.75$ m/s).

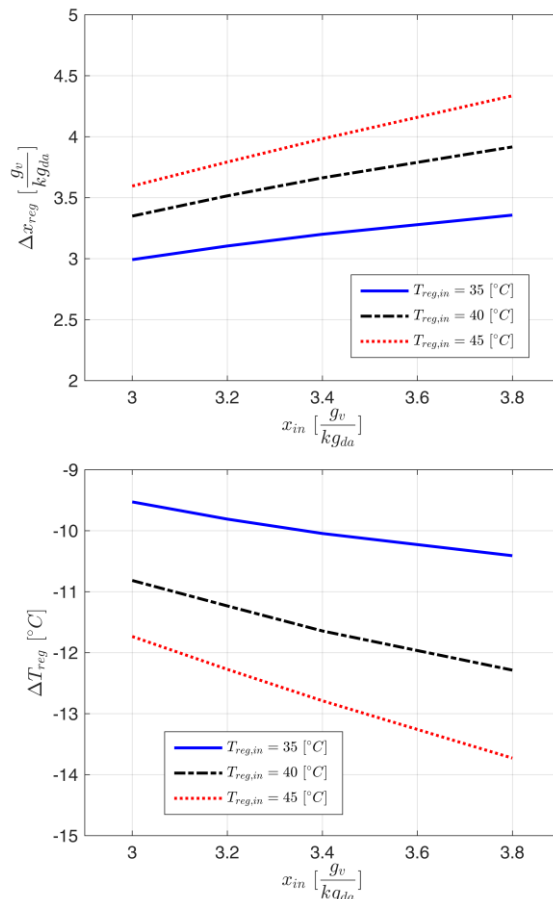


Fig. 7. Off-design performance of the humidification system (Selected configuration operating at $T_{pro,in} = 0^\circ\text{C}$, $v_{pro} = 0.75$ m/s).

to control the humidity ratio of the air stream supplied to the building. Anyway, it should be considered that, in case of high regeneration temperature, $T_{reg,out}$ can easily reach 30-35 °C. Consequently, a cooling process of the airflow before it is supplied to the building could be necessary.

6 Conclusions

In this work, a humidification system based on two silica gel packed beds is proposed and it is investigated through a numerical and experimental approach. It is shown that the humidification system operating with air at outdoor conditions ($T_e=0$ °C, $x_e=3.4$ g/kg) can provide the desired humidification capacity ($\Delta x_{reg,out}=3.2$ g/kg).

A critical issue is the outlet regeneration temperature, which might be too high ($T_{reg,out}>30$ °C) if the outdoor conditions get worse (high temperature and low humidity ratio) and, consequently, might lead to occupants' discomfort or require a cooling process.

Overall, the proposed system can be driven by low grade heat, being an interesting alternative to conventional steam humidifiers.

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Nomenclature

Acronyms

AHU	Air Handling Unit
DEC	Desiccant Evaporative Cooling
PGC	Pseudo Gas-side Controlled

Symbols

A	cross section area [m ²]
c_p	specific heat [J/(kg K)]
d_p	particle diameter [m]
G	mass flux [kg/(m ² s)]
h_c	heat transfer coefficient [W/(m ² K)]
h_m	mass transfer coefficient [kg/(m ² /s)]
L	bed length [m]
Le	Lewis number [-]
\dot{m}	mass airflow rate [kg/s]
P	perimeter [m]
p	pressure [Pa]
RH	relative humidity [-]
T	temperature [°C – K]
\bar{T}	time average temperature [°C or K]
t	time coordinate [s]
Re	Reynolds number [-]
v	superficial air velocity [m/s]
W	average water content [kg _v /kg _{dec}]
x	humidity ratio (dry air based) [kg _v /kg _{da}]
X	humidity ratio (humid air based) [kg _v /kg _a]
z	axial coordinate [m]

Greek Symbols

ϵ	void fraction [-]
ν	kinematic viscosity [m ² /s]
ρ	density [kg/m ³]
τ	half cycle duration [s]

Subscripts

a	humid air
amb	ambient air
bed	solid phase
da	dry air
dec	desiccant
e	external (outdoor) air
in	inlet
out	outlet
p	particle
pro	process air
reg	regeneration air
v	water vapour

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