21th International Conference on Renewable Energies and Power Quality (ICREPQ'23)
Madrid (Spain), 24th to 26th May 2023
Renewable Energy and Power Quality Journal (RE&PQJ)
ISSN 2172-038 X, Volume No.21, July 2023

Optimal placement of heat exchangers in a carbon capture-based ventilation system

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MOTIVES AND INCENTIVES

Providing occupant in indoor spaces with acceptable air quality is of crucial importance to protect their physical and cognitive heath. Such breathable indoor quality (IAQ) is ensured by maintaining the indoor generated species such as H_2O, CO_2 and VOCs to below their healthy thresholds. Among these species, a special attention must be given to the levels of indoor humidity and CO_2 due to their high generation rates from the occupants compared to the other species. Conventionally, ventilation systems are used to pump dehumidified outdoor to dilute these species, where the outdoor air is dried using vapor compression cooling, either as standalone system or integrated with desiccant dehumidifiers. However, this technique is energy intensive and bulky, especially in hot and humid climates. Recent efforts are directed to reduce the reliance on outdoor air and increase the fractions of recirculated indoor air. This is especially beneficial since the indoor air is characterized by lower temperature and humidity levels than that of the outdoor air. Nonetheless, this comes at the expense of jeopardizing the air quality by increasing the indoor CO_2 levels. Accordingly, indoor air treatment for carbon removal by adsorption has been suggested as a novel ventilation strategy. However, most adsorbents suffer from reduced and even abolished capacity for CO_2 in presence of H_2O, especially at the dilute levels. Accordingly, the indoor air requires dehumidification prior to the carbon removal, which can be achieved using the thermally driven desiccant dehumidifiers. Such system has been proposed by many researchers. Nonetheless, no one considered the effect of the pre-dehumidification on the carbon capture performance. This is especially critical since the dehumidified air is characterized by an elevated temperature that could hinder or reduce the CO2 adsorbent temperature. However, heat recovery units can be used to benefit from the energy of the discharged cool air to enhance the system performance. Therefore, this work evaluates the placement of the heat recovery unit to determine the optimal location that minimized the system size and energy consumption.

SYSTEM DESCRIPTION

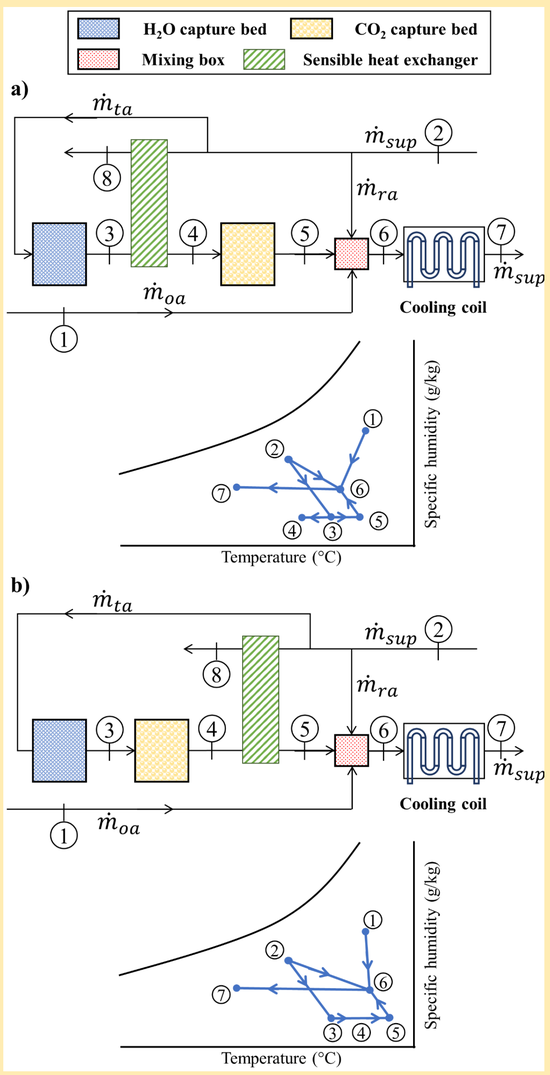

Fig. 1. Schematic of the proposed system with the two considered placement of the heat recovery unit a) before and b) after the decarbonization bed.

The proposed system is implemented in a classroom space located in a hot and humid climate. The proposed system is operated to meet the students thermal comfort conditions by maintaining a temperature of 24 °C and a relative humidity ranging between 40 % and 60 %. To provide the student with acceptable IAQ, the system should maintain CO_2 levels below 1000 ppm, O_2 concentration above 19.5 % and formaldehyde level below 8 ppb. Formaldehyde is chosen as the representative species of the VOCs family since it is commonly found in occupied spaces. The proposed system consists of two packed beds in series that handle the dehumidification and decarbonization of the indoor air, an air-to-air heat exchanger that acts as a heat recovery unit from the discharged indoor air and a cooling coil of a vapor compression cooling systems to provide the space with the necessary supply temperature. Two configurations of this system are proposed in this work where the placement of the heat recovery unit is varied with respect to the decarbonization bed as shown in Fig. 1.

In the first configuration (Fig. 1(a)), the heat exchanger is placed before the decarbonization bed. The indoor air at state (2) is extracted form the classroom at a flowrate $\dot{m}_{sa}$ and divided into airstreams. One airstream at flowrate $\dot{m}_{ta}$ is first dehumidified to state (3) before it is cooled in the heat recovery unit to state (4). The air is then decarbonized to state (5) before it is mixed with the recirculated indoor air $\dot{m}_{ra}$ and outdoor air at flowrate $\dot{m}_{oa}$ and state (1). The resulting air at state (6) is sensibly cooled to state (7) before it is resupplied back to the space. The corresponding states of the ventilation air are shown on the psychrometric chart (Fig. 1(a)).

In the second configuration (Fig. 1(b)), the heat recovery unit is placed after the decarbonization bed. In this case, the indoor air at state (2) is dehumidified to state (3), decarbonized to state (4) before it is sensibly cooled in the heat recovery unit to state (5). The remainder of the process is as in the previous configuration.

CASE STUDY

The developed models are applied for a case study of a typical high occupancy classroom of dimensions (5 m × 10 m × 3.5 m) with 33 students, located in Doha, characterized by extreme hot and humid conditions. The sizing of the different beds depends on the amount of the water vapor and CO_2 to be removed from the treated indoor air flowrate $\dot{m}_{ta}$. For the proper operation of the proposed system, the adsorbing material should be adequately chosen. For indoor air dehumidification, the conventional silica gel is selected due to its commercial availability at low prices, moderate regeneration temperature (<80 °C). For the carbon capture, MOF-74-Mg is selected since it presents high capacity at dilute levels of CO_2, moderate selectivity of H_2O over CO_2, low regeneration temperature, hygrothermal and cycling stability. For this reason, since it exhibits a CO_2 capacity of 6.3 g/kg at outdoor CO_2 levels and a regeneration temperature of 60 °C. Moreover, for the selected adsorbent MOF-74-Mg, an inlet humidity must not exceed 4 g/kg.

METHODOLOGY

Mathematical models are developed for the heat and mass transfers in the packed adsorption beds as well as inside the classroom space. These models are used to size and operate the packed beds for the two different configurations of the proposed system to determine the optimal placement of the heat exchanger with respect to the decarbonization bed.

The adsorbent for both H2O and CO2 are loosely packed in fixed beds to selectively capture the targeted species. In addition, heat is also exchanged between the adsorbent and the air due to the exo- and endothermic nature of the adsorption/desorption processes. Accordingly, coupled heat and mass transfers are needed to accurately model the packed bed operation and accurately determine the outlet air conditions. For this reason, a transient one-dimensional model is adopted with axially dispersed flow and constant velocity along the airflow direction. Moreover, the adsorption rate (mass exchange) between the adsorbent and air is modelled using the linear driving force assumption, where the mass exchange is proportional to the difference between the equilibrium uptake $\overline{q_i^*}$ and the actual uptake $\overline{q}_i$. The equilibrium uptake is calculated at the adsorbent temperature and the species concentration in the air using the isotherm model of the adsorbent. The species mass balance on the air side is thus given by

$$\frac{\partial C_i}{\partial t} + u\frac{\partial C_i}{\partial z} - D_{zi}\frac{\partial^2 C_i}{\partial z^2} + \left(\frac{1-\varepsilon_t}{\varepsilon_t}\right)\rho_s k_i(\overline{q}_i^* - \overline{q}_i) = 0$$

To determine the air conditions of temperature and species concentrations in terms of the supplied air conditions, heat and mass balances were developed for the inside of the classroom space, which is assumed to be well mixed (homogeneous). The lumped transient heat and mass balances are thus given by

$$\rho_a V_{CR} C_{p,a}\frac{dT_{CR}}{dt} = \dot{m}_{sup} C_{p,a}(T_{sup} - T_{CR}) + Q_e + Q_i$$

$$\rho_a V_{CR}\frac{dy_{CR,i}}{dt} = \dot{m}_{sup}(y_{i,sup} - y_{CR,i}) \pm \dot{Y}_i$$

The different models of heat and mass balances are solved using the finite volume approach with implicit scheme for the time derivative (transient term) and the first order upwind scheme for the first order spatial derivative (convection term) while the second order spatial derivative (diffusion term) was solved using the central difference scheme. A time step independence test yielded a time step of 10-3 s that is adopted to ensure accurate results with acceptable computational time. The convergence reached when the difference in the calculated parameters between two consecutive iterations is smaller than 10-8.

RESULTS AND DISCUSSION

The developed mathematical models were validated for the adsorption of both water vapor on silica gel and CO_2 on MOF-74-Mg with published data in literature. It was found that the maximum discrepancy on the calculated outlet air temperature and species concentration from the packed bed model were less than 9.5 %.

Using the classroom space model, it was found that the supply, treated and outdoor air flowrates needed at the peak humidity conditions were found to be 1.02 kg/s, 0.26 kg/s, and 0.078, respectively. These flowrates were required to offset the different species generation/consumption rates and ensure acceptable IAQ in the classroom and meet the required indoor air temperature that provided thermal comfort for the students.

The validated packed bed model was then used to properly size the packed beds for dehumidification and decarbonization following the approach presented in the previous section for both system configurations. Since the heat exchanger location was varies with respect to the decarbonization bed only, the size of the dehumidification bed remained the same. It was found that 15 kg of silica gel were needed in each bed to handle the peak latent load of the indoor air that ensure an airstream that is dry enough before entering the CO_2 capture bed. On the other hand, the size of decarbonization bed was drastically changed as can be seen in Table 1. For the first configuration, the heat exchanger was placed before the carbon capture bed. This resulted in pre-cooling of the dehumidified air that reduced the temperature of the MOFs and increased its CO_2 capacity to 10 g/kg at 34 °C. This resulted in peak thermal and electrical energy consumption of 0.39 kWh and 0.88 kWh, respectively.

For the second configuration, the decarbonization bed was sized while considering the temperature of the dehumidified air leaving the dehumidification bed. For the current case study, it was found that the dried air reached a temperature of 39 °C at the outlet of the bed. At this temperature, the MOFs capacity for CO_2 was reduced to 7.5 g/kg. This caused the increase of the needed adsorbent mass to 38.6 kg for each bed, which was equivalent to a rise of 33 % in the system size with respect to the first configuration. Consequently, the thermal energy for the adsorbent regeneration and the electrical energy required for the fan operation increased to 0.56 kWh and 1.1 kWh, respectively. This configuration resulted thus in 43.5 % and 25 % lower thermal and electrical energy consumption compared to the previous configuration. Therefore, the placement of the heat exchanger before the decarbonization bed was the optimal solution for energy efficient operation of the proposed system.

Table 1. The obtained packed bed dimensions for the considered system configurations.

Heat exchanger placement	*Before decarbonization bed*	*After decarbonization be*
Treated indoor air flowrate (kg/s)	*0.26*	
Dehumidification bed		
Silica gel mass (kg)	14.6	
Bed height (m)	0.20	
Bed diameter (m)	0.35	
Carbon capture bed		
MOF-74-Mg mass (kg)	29.0	38.6
Bed height (m)	0.30	0.40
Bed diameter (m)	0.60	0.60

CONCLUSION

For the peak outdoor air conditions of humidity, the configuration with the heat recovery preceding the decarbonization bed showed a reduction of 33 % in the carbon capture bed size along with 43.5 % and 25 % lower thermal and electrical energy consumption.