Theoretical modelling and experimental study of air thermal conditioning process of a heat pump assisted solid desiccant cooling system

Nie, Jinzhe; Li, Zan; Hu, Wenju; Fang, Lei; Zhang, Qunli

Published in:
Energy and Buildings

Link to article, DOI:
10.1016/j.enbuild.2017.07.075

Publication date:
2017

Document Version
Peer reviewed version

Citation (APA):
Nie, J., Li, Z., Hu, W., Fang, L., & Zhang, Q. (2017). Theoretical modelling and experimental study of air thermal conditioning process of a heat pump assisted solid desiccant cooling system. *Energy and Buildings*, *153*, 31-40. https://doi.org/10.1016/j.enbuild.2017.07.075

Theoretical modelling and experimental study of air thermal conditioning process of a heat pump assisted solid desiccant cooling system

Jinzhe Nie[a,b], Zan Li[a], Wenju Hu[a,b,*], Lei Fang[a,c], Qunli Zhang[a,b]

[a] Beijing Key Lab of Heating, Gas Supply, Ventilating and Air Conditioning Engineering, School of Environment and Energy Engineering, Beijing University of Civil Engineering and Architecture, Beijing 100044, P.R. China

[b] Beijing Advanced Innovation Center for Future Urban Design, Beijing University of Civil Engineering and Architecture, Beijing 100044, P.R. China

[c] International Centre for Indoor Environment and Energy, Department of Civil Engineering, Technical University of Denmark, Kgs. Lyngby 2800, Denmark

*Corresponding author. Tel: +8610 68331450. E-mail address: huwenju@bucea.edu.cn

Highlights

- A HP-SDC was proposed based on the VOCs and moisture adsorption of desiccant.
- A theoretical model of air thermal process in desiccant rotor was modelled.
- A prototype unit of the HP-SDC was developed and measured.
- The HP-SDC has high energy efficiency in tropical climates.
- The energy efficiency of the HP-SDC is sensitive to outdoor humidity ratio.

Abstract

Taking the integrated gaseous contaminants and moisture adsorption potential of desiccant material, a new heat pump assisted solid desiccant cooling system (HP-SDC) was proposed based on the combination of desiccant rotor with heat pump. The HP-SDC was designed for dehumidification, cooling and air purification aimed at improving indoor air quality and reducing building energy consumption. The heat and moisture transfer in adsorption desiccant rotor was theoretical modelled with one-dimensional partial differential equations. The theoretical model was validated with experimental measurements, and the results showed the model could be used to predict the heat and moisture transfer in desiccant rotor. The air thermal conditioning process and energy consumption of HP-SDC was then experimental measured under varied outdoor thermal environments. Results showed that compared to conventional ventilation system, the energy performance of HP-SDC was more efficient mainly due to high efficient air purification capacity, reduction of cooling load and raised evaporation temperature. The energy performance of HP-SDC was sensitive to outdoor humidity ratio. Further improvement of HP-SDC energy efficiency is suggested to be focused on low regeneration temperature desiccant rotor and more efficient high temperature refrigerant.

Key words: Solid desiccant cooling; Dehumidification; Sensible cooling; Heat pump; Desiccant rotor.

1 Introduction

To reduce energy consumption of ventilation and air-conditioning in buildings, solid desiccant cooling system has been proposed for years. Within a typical solid desiccant cooling system, the dehumidification for ventilation air is managed by desiccant material, and the sensible cooling is handled by cooling coil. Regenerative desiccant rotor is commonly used as the desiccant component in solid desiccant cooling system due to its continuous dehumidification function. Regenerative desiccant rotor has been commonly used as the dehumidifier in ventilation and air-conditioning system since Pennington patented the first solid desiccant cooling system-Pennington cycle in 1955 [1]. In the Pennington cycle, outdoor air supply firstly transferred moisture to desiccant wheel, and then transferred sensible heat to indoor exhaust air through a rotary sensible heat exchanger. Before delivered to indoors, the outdoor air supply was cooled by cooling coil. Thus, the outdoor air delivered to indoor was dehumidified and cooled to keep a comfortable indoor thermal environment. On the other side, indoor exhaust air was cooled by cooling coil and then got heat from outdoor air supply through the sensible rotary heat exchanger. Then indoor exhaust was heated by heating coil and used to regenerate the wheel which was statured with moisture adsorbed from outdoor air supply. Since the dehumidification for outdoor air supply was conducted by desiccant wheel, the liquid temperature in the cooling coil of Pennington cycle could be higher than that in conventional vapor-compression refrigeration cycle, and thus the energy efficiency could be improved. The main weak points of Pennington cycle include: the heating for regeneration air was conducted by electrical power which would decrease the energy efficiency of the system, no attention has been paid on the contaminant transfer from indoor exhaust air to outdoor air supply through the adsorption wheel and it may constitute a severe pollutant load for outdoor air supply.

Great efforts have been taken on the research and development of new solid desiccant cooling systems based on Pennington cycle. Considering energy consumption in HVAC systems, solid desiccant cooling system was energy efficient only when the desiccant rotor was regenerated with sustainable thermal energy such as solar energy [2]- [4], energy from co-generators [5] [6], or waste heat [7] which are normally limited by regional or climate factors [8]. Heat pump assisted solid desiccant cooling system which used condensing heat of heat pump to regenerate the desiccant rotor was proposed in several studies [9]- [12]. However, there has not been a consistent conclusion on the energy efficiency of the proposal due to the large variety in system configurations and operating conditions. Subramanyam et al. [13] integrated a desiccant rotor into a traditional refrigeration air conditioning device. They concluded that the coefficient of performance (COP) of the integrated system was 5% lower than the traditional system in the case where supply air was not reheated and delivered into conditioned space at dew point, and the COP of the integrated system was nearly double of the traditional air conditioning system if the supply air was reheated. Jurinak et al. [14] used a direct evaporative cooling device instead of a plate heat exchanger or cooling coil to take the air sensible cooling load after the desiccant dehumidification. They indicated that systems with improved dehumidifier, rotary sensible heat exchanger and direct evaporative cooling device could achieve seasonal COPs in the order of 1.1. Due to the direct evaporative cooling after dehumidification, the outlet air humidity ratio from desiccant rotor should be lower that the supply air, which resulted in much higher regeneration air temperature. Sheridan and Mitchell [15] proposed a solid desiccant cooling system in which outdoor air at ambient state was mixed with return air from the air-conditioned space, and the mixed air was dehumidified using a rotary desiccant dehumidifier and subsequently cooled by an indirect evaporative cooler which was a plate heat exchanger device. Further cooling to the required entry was performed by the evaporator coil of a vapor compression unit. The desiccant dehumidifier was regenerated by heat from the condenser of the vapor compression unit plus a solar cycle auxiliary heater. This meant that the condenser operated at a lower temperature than in the case where regeneration heat was fully from the condensing heat, and thus the vapor compression unit operated more efficiently. They reported that such a system saved energy compared with a vapor compression unit when the load had a high sensible fraction but vice-versa when the load had a high latent fraction. On the regeneration air side of heat pump assisted solid desiccant cooling system, Jia et al. [9] and Hao et al. [12] used additional electrical heater to achieve high regeneration air temperature, but the additional electrical heater decreased the overall energy efficiency of the system. The studies of Zhang et al. [10] and Sheng et al. [11] used high temperature refrigerants to achieve high condensing and regeneration temperature, but the COP of heat pump were relative low in their studies. A multi-stage solid desiccant system was proposed in the study of Tu et al. [16]. In the system, multi-stage solid desiccant rotors were used to dehumidify the ventilation air, the evaporator of a heat pump was used to sensibly cool the ventilation air, and the condenser of heat pump was used to heat up the regeneration air. Through the multi-stage desiccant rotors, the regeneration air temperature was significantly decreased. Thus, the COP of the heat pump which provided sensible cooling and regeneration heat was increased and the energy efficiency of the system was improved. The multi-stage desiccant rotor also increased the initial and maintenance investment which should be taken into consideration and the humidity-temperature independent control system in the multi-stage solid desiccant cooling system still needed further research.

These previous studies on solid desiccant cooling system showed a fact that the desiccant cooling technique was still not ideal due to either inconsistent energy efficiency or complex configuration and high initial investment. Besides the research on dehumidification, the studies of Fang et al. [17] and Zhang et al. [18] brought new vitality for solid desiccant cooling system since they found the silica gel rotor commonly used as desiccant wheel could remove indoor air pollutants such as VOCs with a high efficiency. The VOCs removal efficiency could reach more than 80% when the silica gel desiccant rotor was operated as a normal dehumidifier. However, the high indoor air cleaning performance of desiccant rotor has not been integrated into ventilation system.

In this article, a new type of heat pump assisted solid desiccant cooling system was proposed to optimize the air conditioning performance of desiccant rotor. The proposed heat pump assisted solid desiccant cooling system (HP-SDC) integrated air purification, dehumidification, cooling in one unit. A theoretical model based on one-dimensional partial differential equations was established to reveal the heat and moisture transfer between the ventilation and regeneration air in the system. A prototype unit of the proposed HP-SDC was developed. Humidity and temperature independent control system was designed and developed in the optimized HP-SDC. Theoretical simulations and experimental measurements were then conducted to investigate and evaluate the air thermal conditioning and energy performance of the new HP-SDC. The key factors which influenced the performance of the HP-SDC were thus explored.

2 Methodologies

As mentioned above, a novel heat pump assisted solid desiccant cooling system was designed, theoretically modeled, physically developed and experimental measured. Figure 1 shows the schematic diagram of the air system designed in the HP-SDC. In the proposed system, due to 80% of the cleaned recirculation air by the desiccant rotor could be used as outdoor fresh air [17],ventilation system supplies minimum levels of outdoor airflow and recirculates a large quantity of indoor air. The mixed ventilation air is dehumidified, cleaned by the solid desiccant rotor and cooled by the evaporator of the heat pump. The cool, dry and clean ventilation air is then delivered into the ventilated room. On the other side, the condensing heat from the heat pump is used to heat up the outdoor air which would regenerate the desiccant rotor. Due to that the condensing heat is always more than the regeneration heat requirement, another condenser was designed to take away the surplus condensing heat. The dual condensers design is also used to realize humidity and temperature independent control in the solid desiccant cooling system. Finally, the regeneration air, indoor exhaust air and the air taking surplus condensing heat is mixed and expelled to outdoor. In this way, the sensible heat, latent heat and air pollutants of indoor climate are removed to outdoor environment. From this design, energy on both sides of the heat pump is used for thermal conditioning and cleaning of ventilation air. The condensing heat is used to regenerate the rotor (removing latent heat and pollutants from the rotor). The evaporation cooling is used to cool the process air (removing sensible heat).

Figure 1.

Different from conventional heat pump assisted solid desiccant cooling system, the air cleaning performance of solid desiccant rotor is applied in the HP-SDC. The ventilation air which need to be dehumidified is from a large portion of indoor recirculation air mixing with a necessary amount of outdoor air, and this could effectively decrease the dehumidification load of the desiccant rotor and thus reduce the regeneration air temperature. Two condensers with electronic valves fitted on refrigerant outlets of the condensers were designed in the HP-SDC. The dual condensers and electronic valves design was aimed to distribute the refrigerant flow in each condenser and to control the heat for regeneration air exactly match which is required by the dehumidification load of the rotor.

To get the operation characteristics of the HP-SDC system under global thermal climates, a theoretical model to simulate the moisture and heat transfer in the desiccant rotor was developed. A prototype unit of the HP-SDC was designed and developed to experimentally measure its air thermal conditioning and energy performance.

2.1 Theoretical modeling of HP-SDC

To predict the air thermal conditioning process in HP-SDC, the theoretical model of desiccant rotor is the key where combined heat and moisture transfer occur predominantly in the system.

In previous modeling works, the studies of Nia et al. [19] and Zhang et al. [20] modeled the heat and moisture transfer in silica gel rotors. The study of Nia et al. [19] simulated the combined heat and mass transfer process that occurred in a solid desiccant wheel having balanced regeneration and adsorption air streams with a transient and one-dimensional model. In the modeling work of Zhang et al. [20], the partial differential equations were converted into finite differential equations and were solved with Gauss-Jordan elimination method. The study of Ge et al. [21] proposed a novel silica gel haloid compound desiccant wheel. The heat and mass tranfer between the air and the compound desiccant wheel was theretically modelled considering both the gas side and solid side resistance, and relative better agreements with experiment results were found in the study [21]. In the study of Koronaki et al. [22], a one dimensional GSR model which predicted the behavior of desiccant wheel was presented, and experimental data from two real desiccant wheels were used in order to validate the model which showed the simulation results are in reasonable agreement with the experimental data. In the study of Yadav [23], a mathematical model for predicting the performance of a desiccant wheel was proposed and used to evaluate the infleunce of effective regeneration sector.

From the previous studies, the key factors for modelling the heat and moisture transfer in desiccant rotors were summarized to be: the heat and mass transfer equations, the heat and moisture diffusion velocity between the air and adsorbent, and the equilibrium relation of mass concentration between the air and the adsorbent. In the theoretical model presented in this article, a one-dimensional non-steady-state model was established with a set of one-dimensional partial differential equations.

The physical model of the desiccant rotor is defined in Figure 2. In the theoretical model, following assumptions of heat and moisture transfer between the air and desiccant rotor were given.

(1) The air acceleration, heat conduction and mass diffusion in the solid materials and air stream along the axial direction of the rotor was neglected.

(2) The air density was assumed to be constant in the system.

(3) The sorption of moisture on the substrate material of the desiccant rotor was neglected.

(4) The adsorption heat of indoor air pollutions and sorption competition between pollution and moisture was neglected due to the minor proportion of indoor air pollution compared to moisture content.

Figure 2.

The heat and moisture transfer model was written based on the following equations.

The moisture conservation equation was written as:

$$\frac{\partial Y}{\partial t} dt \times (A\rho_A dx) + \frac{\partial Y}{\partial x} dx \times (VA\rho_A dt) + \frac{\partial W}{\partial t} dt \times (f_d dx) = 0$$

(1)

Where, $\frac{\partial Y}{\partial t} dt \times (A\rho_A dx)$ is the moisture weight change in the air inside the computation domain after a dt period of adsorption/desorption.

$\frac{\partial Y}{\partial x} dx \times (VA\rho_A dt)$ is the moisture weight difference between the inlet and outlet air of the computation domain.

$\frac{\partial W}{\partial t} dt \times (f_d dx)$ is the moisture content change in the adsorbent of the computation domain after a dt period of adsorption/desorption.

Y is the moisture content in the air, g / kg.

A is the cross section area of a computation domain, m^2.

ρ_A is the density of the air, kg / m^3.

V is the air speed in the flute section of the silica gel rotor, m / s.

W is the moisture adsorbed on the adsorbent, g/kg.

f_d is the weight of sorption material in the direction along the airflow, kg/m.

Dividing $dt \times (A\rho_A dx)$ in each section of Equation 1, the moisture conservation equation was written as:

$$\frac{\partial Y}{\partial t} + V\frac{\partial Y}{\partial x} + \frac{f_d}{A\rho_A} \times \frac{\partial W}{\partial t} = 0$$

(2)

The moisture transfer between the air and adsorbent was written as :

$$\frac{\partial W}{\partial t} dt \times (f_d dx) = K_Y(Y - Y_W)\ Pdtdx$$

(3)

Where, $K_Y(Y - Y_W)\ Pdtdx$ is the moisture tranfered bwteen the air and the adsorbent in the computaion domain during a dt period of adsorption/desorption.

Dividing $dt \times (f_d dx)$ in each section and considering the condition in which the adsorbent is saturated with moisture, Equation 3 was further written to be：

$$\frac{\partial W}{\partial t} = \begin{cases} \frac{K_Y P}{f_d}(Y - Y_w), & when\ W < W_{\max},\ or\ when W = W_{\max}\ and\ Y < Y_w \\ 0, & when\ W = W_{\max}\ and\ Y > Y_w \end{cases}$$

(4)

K_Y is the mass transfer coefficient, $kg/m^2/s$.

P is the section perimeter of a computation domain, m.

$W_{\max}$ is the maximum moisture adsorbed on the adsorbent, g/kg.

Y_w is the moisture content in the air which reached equilibrium state with adsorbent, g/kg.

The energy conservation equation was written as:

$$(YC_{pv} + C_{pa})\frac{\partial T_{k-air}}{\partial t}dt \times A\rho_A dx + (YC_{pv} + C_{pa})\frac{\partial T_{k-air}}{\partial x}dx \times VA\rho_A dt + (C_{pd} + WC_{p1})\frac{\partial T_{k-adsorbent}}{\partial t}dt \times f_d dx + C_{pm}\frac{\partial T_{k-adsorbent}}{\partial t}dt \times f_m dx = q\frac{\partial W}{\partial t}dt \times f_d dx$$

(5)

Where, $(YC_{pv} + C_{pa})\frac{\partial T_{k-air}}{\partial t}dt \times A\rho_A dx$ is the energy change of the air inside the computation domain after a dt period of adsorption/desorption.

$(YC_{pv} + C_{pa})\frac{\partial T_{k-air}}{\partial x}dx \times VA\rho_A dt$ is the energy difference between the inlet and outlet air of the computation domain.

$(C_{pd} + WC_{p1})\frac{\partial T_{k-adsorbent}}{\partial t}dt \times f_d dx$ is the energy change of the adsorbent in the computation domain after a dt period of adsorption/desorption.

$C_{pm}\frac{\partial T_{k-adsorbent}}{\partial t}dt \times f_m dx$ is the energy change of the substrate material in the computation domain after a dt period of adsorption/desorption.

$q\frac{\partial W}{\partial t}dt \times f_d dx$ is the adsorpton heat realsed during the moisture sorption on the adsorption material.

Dividing （$YC_{pv} + C_{pa}$） $dt \times dxA\rho_A$ in each section, Equation 5 could be further written to be：

$$\frac{\partial T_{k-air}}{\partial t} + V\frac{\partial T_{k-air}}{\partial x} + \frac{(C_{pd} + WC_{pl})f_d + C_{pm}f_m}{(YC_{pv} + C_{pa})A\rho_A}\frac{\partial T_{k-adsorbent}}{\partial t} = \frac{qf_d}{(YC_{pv} + C_{pa})A\rho_A}\frac{\partial W}{\partial t} \quad (6)$$

T_{k-air} is the absolute temperature of air, K .

$T_{k-adsorbent}$ is the absolute temperature of adsorbent, K .

f_m is the weight of substrate material in the direction along the airflow, kg/m.

C_{pa} is the specific heat at constant pressure of the air, $J/kg/k$.

C_{pd} is the specific heat at constant pressure of the air, $J/kg/k$.

c_{pl} is the specific heat at constant pressure of moisture in the adsorbent, $J/kg/k$.

c_{pm} is the specific heat at constant pressure of substrate material, $J/kg/k$.

c_{pv} is the specific heat at constant pressure of moisture in the air, $J/kg/k$.

q is the adsorption heat of moisture, J/kg.

The heat transfer between the air and the adsorbent was written as:

$$\frac{\partial T_{k-adsorbent}}{\partial t} dt[(C_{pd} + WC_{p1})f_d dx + C_{pm} f_m dx] + \alpha P(T_{k-adsorbent} - T_{k-air}) dt dx + qK_Y P(Y_w - Y) dt dx = 0$$

(7)

Where, $\alpha P(T_{k-adsorbent} - T_{k-air}) dt dx$ is the heat transferred between the air and the adsorbent in the computation domain during a *dt* period of adsorption/desorption.

$qK_Y (Y_w - Y) P dt dx$ is the adsorpton heat realsed in the computation domain during a *dt* period of moisture sorption on the adsorption material.

α is the heat transfer coefficient, $W / m^2 / k$.

Diving $dt[(C_{pd} + WC_{p1})f_d dx + C_{pm} f_m dx]$ in each section, Equation 7 could be further written to be：

$$\frac{\partial T_{k-adsorbent}}{\partial t} + \frac{\alpha P}{(C_{pd} + WC_{p1})f_d + C_{pm} f_m}(T_{k-adsorbent} - T_{k-air}) + \frac{qK_Y P}{(C_{pd} + WC_{p1})f_d + C_{pm} f_m}(Y_w - Y) = 0$$

(8)

The Equation 2, 4 give the moisture and energy conservation in the air and the adsorbent in a computation domain, and the Equation 6, 8 give the moisture and heat transfer in the air and the adsorbent in a computation domain.

Referred to the study of San [24], the adsorption heat of moisture could be calculated with equation 9.

$$q = h_v(1.0 + 0.2843e^{-10.28W})$$

(9)

Where,

h_v is the latent heat of water vaporization, J/kg.

According to the study of Zhang et al. [20], the saturated vapor pressure in the air which achieved equilibrium with the adsorbent be calculated with following equation.

$$\ln p_{ws} = 23.196 - \frac{3816.44}{T_w - 46.13}$$

(10)

The equations 1-10 form the integrated heat and moisture transfer in the silica gel rotor. The heat and moisture transfer equations were coded in MATLAB to solve the proposed model.

2.2 Experimental system of HP-SDC

Besides the theoretical modeling, a prototype unit of the HP-SDC was designed and developed to experimentally measure the performance of the proposed system. The development of a real prototype unit could also be useful to discover the practical problems and solutions of the HP-SDC application. The airflow diagram shown in Figure 1 gives the air system of the HP-SDC. Figure 3 shows the heat pump system of the HP-SDC.

Figure 3.

The detailed information of the key components including the silica gel rotor, compressor, expansion valve, evaporator and condensers in the HP-SDC prototype are given in Table 1. The models and accuracies of the measuring instruments in the prototype are shown in Table 2.

Table 1.

To control the refrigerant flow rate to the two condensers, electronic control valves “EX5-U21” produced by “Emerson” were selected and connected to the outlets of the condensers.

Table 2.

Regarding the refrigerant for the heat pump in HP-SDC, different refrigerants were compared with their COP, environmental friendliness and safety performance. R134 was finally chosen to be the candidate according to the environmental protection criteria in Europe and the un-combustibility requirement of laboratory operation in Denmark. The Ozone Depletion Potential (ODP) of R134a is 0 and its Global Warming Potential (GWP) is 1300.

With all the key components selected, the prototype unit of HP-SDC was constructed. Figure 4 shows photos of the heat pump and silica gel rotor in the prototype unit.

Figure 4.

Experimental measurements were then conducted with the prototype in a field lab in Technical University of Denmark. With the purpose of measuring the HP-SDC operating in different outdoor climates, the field lab was equipped with a ventilation system which can mimic different outdoor climate conditions. An air handling unit was used to simulate the cooling load in the field lab. Several electric humidifiers were used to simulate the latent load in the field lab.

Geometry and thermal-physical properties of the field lab are list in Table 3. Indoor thermal environment set-points and interior heat conduction in the field lab are listed in Table 4. Based on the thermal-physical information of the field lab, the sensible and latent cooling load of the field lab in different outdoor thermal environments can be calculated.

Table 3.

Table 4.

For the HP-SDC operation in summer climates, the supply airflow rate for the field lab was designed to be fixed at 230L/s but with variable supply air temperature and humidity ratio, i.e. constant air volume air-conditioning system (CAV).

3 Simulation and experiment case design

With the theoretical model and the prototype unit, theoretical simulations and experimental measurements were conducted to inquire into the air thermal conditioning process and energy performance of the HP-SDC under different outdoor climate conditions, i.e. warm, hot and extremely hot and humid climates. Using the climate data of temperature and humidity for each hour of year 2002 in Colombo, Sri Lanka which represent tropical climate, five typical outdoor climate classes plus one extreme condition were categorized. They represent the most probable outdoor climate conditions in which the HP-SDC could work during the whole year. Five climate classes were categorized with outdoor air temperatures, and the temperature representing each class is the mean value in the range of the class. The corresponding moisture content representing the class is the mean value of outdoor air humidity ratio in the temperature classes.

Table 5.

With the outdoor thermal climate conditions and the thermal physical properties of the field lab, the thermal loads of the field lab and the required thermal conditions of ventilation air delivered to the field lab were calculated and summarized in Table 6.

Table 6.

As mentioned, the airflow delivered to the field lab was designed to be 230L/s which was mixed by indoor recirculation air and outdoor air. Due to the strong indoor air purification ability of silica gel rotor, and according to the EU standard for ventilation [25], the outdoor air delivered into the field lab in the HP-SDC system was designed to be 4L/s per person. Table 7 gives the rates of different airflows in the HP-SDC.

Table 7.

Under the designed simulation and experiment cases, the thermal conditions of airflows at different locations of HP-SDC were simulated and experimental measured. The locations and corresponding implications of airflows are listed in Table 8. The heat pump operation conditions including the evaporating pressure, condensing pressure and power consumption under different cases were measured as well.

Table 8.

4 Results

Theoretical simulations and experimental measurements were conducted to investigate the air thermal conditioning process and energy consumption of the HP-SDC, and the results are given as followings.

4.1 Model validation

Due to that the theoretical model was aimed to reveal the heat and mass transfer in the desiccant rotor, during the theoretical simulation, the thermal conditions of process and regeneration inlet air were got from the measured values. The simulated thermal conditions of process and regeneration outlet air are compared with experimental measurement results in Table 9.

Table 9.

The simulated temperature and humidity ratio of dehumidified air and regeneration outlet air consist with the measured values. The relative deviations of temperature are from 0.2% to 3%, and the humidity ratio deviations are in the range from 9% to 14% under the designed cases. The deviation of humidity ration is speculated to be caused by the air leakage between dehumidification section and regeneration section in the desiccant rotor. In the theoretical model, the dehumidification section was assumed to be completely isolated from regeneration section, but air leakage and carrying over occurred in the prototype unit. Over all, the validation results show that the theoretical model could be used to predict the heat and moisture transfer between process air and regeneration through the desiccant rotor.

4.2 Air thermal conditions in HP-SDC

Under the designed experimental cases, the thermal conditions of process air and regeneration air at different locations of the HP-SDC were measured, and the results are shown below in Table 10.

Table 10.

To better show the air conditioning process of HP-SDC, the thermal conditions of ventilation air and regeneration air under the summer extreme class of Colombo are given on psychrometric chart as Figure 5.

Figure 5.

From the figure, it can be seen that the total cooling load of HP-SDC could be much lower than the traditional air conditioning system due to the strong air cleaning potential of silica gel rotor and the non-dew point cooling dehumidification design. Besides the energy saving of dehumidification and sensible cooling, the energy consumption for regenerating the desiccant rotor become a big concern of the HP-SDC application. However, Table 10 shows that there is always excess heat which should be taken away from the heat pump. It means the condensing heat from heat pump is always more than regeneration requirement. In this way, decreasing the regeneration temperature become the key point to further improve the COP and energy performance of the HP-SDC.

4.3 Energy performance of HP-SDC

Under the designed experimental cases, the power consumption of the heat pump in HP-SDC was measured with “Danfoss ADAP-KOLL Drive” frequency controller and shown in Table 11. With thermal conditions before and after the evaporator, the cooling load of heat pump could be calculated, and thus the COP of heat pump was calculated and listed in Table 11.

Table 11.

The energy consumption of HP-SDC increased with the increasing of cooling load as expected. However, the COP decreasd with the increasing of outdoor humidity ratio. This could be mainly due to that higher outdoor humidity ratio require higher regeneration temperaure which lead to higher condensing temperature and lower COP of the heat pump.

5 Discussions

From the theoretical modelling and experimental validation of heat and moisture transfer in the desiccant rotor, the following parameters are summarized to be key issues for the accuracy of the prediction model.

1) The maximum moisture content on the solid adsorbent.

2) The mass transfer velocity between the air and the solid adsorbent.

3) The adsorption heat released from moisture adsorption on the adsorbent.

During modelling the silica gel desiccant rotor in this article, the adsorption heat of moisture on silica gel was derived from the study of San [24]. The saturated vapor pressure in the air which achieve equilibrium state with silica gel was derived from the study of Zhang [20]. The maximum moisture content in the silica gel was got from the product manual of solid desiccant rotor manufacture [26]. The key parameters given above are strongly suggested to be experimental measured in advance to model the heat and moisture transfer in other adsorbent desiccant rotors.

The energy efficiency of the proposed HP-SDC could benefit from the novel combination of desiccant rotor with heat pump taking the advantage of high efficient air purification performance, the reduced cooling load and increased evaporating temperature. The air purification performance of the HP-SDC could keep indoor air quality in equal level with traditional ventilation system which should have a much larger outdoor air supply rate, and thus the energy consumption for cooling and dehumidifying outdoor air in the HP-SDC could be reduced. The HP-SDC use solid desiccant material to dehumidify ventilation air and sensible cool the ventilation air with evaporator to realize non-dew point cooling. During the dehumidification process, the enthalpy of ventilation air is constant. During the cooling process, the moisture of ventilation air is constant. Through this way, the cooling load could be lower than traditional air conditioning system which should cool and dehumidify the ventilation air to the dew point of supply air, and the evaporation temperature of heat pump in the HP-SDC could be thus increased. The deceased cooling load and increased evaporation temperature improved the COP and energy efficiency of the HP-SDC.

In the location where outdoor air humidity ratio was higher, the COP of HP-SDC became lower. It was due to higher regeneration temperature was required when outdoor air humidity ratio increased. Increasing regeneration temperature required higher condensing temperature of the heat pump which reduce COP of the system. If the dehumidification capacity of the desiccant rotor could be reactivated by lower regeneration temperature, the energy efficiency of the HP-SDC could be improved significantly. Researches on metal-organic framework materials used as desiccant materials were conducted in the study of Seo [27], and the results showed that MIL-100 (Cr) had excellent desorption property under low regeneration temperature. Positive results have been gotten as well in the studies of Ge et al. [21] [28], Jia et al.

[29] and Tokarev et al. [30] which use composite desiccant material to decrease regeneration temperature. Jeong et al. [31] provided a detailed evaluation of a four-partition desiccant wheel to make low-temperature driving heat source possible and achieve considerable energy saving with a four-partition desiccant wheel hybrid air-conditioning system. In the simulation study of Jeong et al. [31], the hybrid air-conditioning system improves COP by approximately 94% as compared to the conventional vapour compression-type refrigerator. Further research could be conducted on applicating these low temperature regenerative desiccant materials or wheel on solid desiccant cooling system.

Another solution which may further improve the energy efficiency of HP-SDC is to develop refrigerant which has higher COP than R134a in the dynamic thermal process of HP-SDC. The refrigerant used in the prototype HP-SDC is R134a which is commonly used in normal conventional air conditioner. The COP of the HP-SDC would be further improved if refrigerant which has stronger property in the required evaporating-condensing temperature range can be found or synthesized.

The proposed HP-SDC use the indoor air cleaning potential of desiccant rotor to realize equal indoor air quality with less outdoor air supply. This application is especially suitable for regions facing atmospheric pollution challenges where the outdoor air is not lean anymore. In these places, outdoor air pollutions such as particles and oxynitride makes traditional ventilation invalid or even harmful to indoor environment. High efficiency particulate air filters should be integrated in building ventilation systems of these areas which lead to higher fan power consumption and pollution accumulation on the filters. Ventilation systems which have energy efficient indoor air cleaning and thermal conditioning performance may be a solution to improve indoor air quality and decrease building energy consumption in these areas.

6 Conclusions

Based on the dehumidification and air cleaning potential of desiccant rotor, a novel heat pump assisted solid desiccant cooling system was proposed aimed at improving indoor air quality and decreasing building energy consumption. The heat and moisture transfer in the desiccant rotor of the HP-SDC was theoretical modelled. A prototype unit of the HP-SDC was designed, developed. Theoretical simulations and experimental measurements were conducted to investigate the air thermal conditioning process and energy consumption of the HP-SDC, and following conclusions are given.

Experimental validation results showed that the theoretical model established could be used to predict the heat and moisture transfer in the desiccant rotor of the HP-SDC. The relative difference of temperature between simulated and measured results vary from 0.2% to 3%, and the humidity ratio deviation between simulated and measured results vary from 9% to 14% under the designed cases.

From the air thermal conditioning and energy measurements, the proposed HP-SDC has shown efficient energy performance for building ventilation and air conditioning. The energy performance of HP-SDC could benefit from high efficient air purification potential, reduced cooling load and increased evaporation temperature. The energy efficiency of the HP-SDC is sensitive to outdoor humidity ratio. In the outdoor environment where has higher humidity ratio, the COP of the HP-SDC decrease.

Further improvement of the HP-SDC could be from low regeneration temperature desiccant rotor, high efficiency refrigerant and desiccant material which has low temperature rise during moisture adsorption.

7 Acknowledgments

This work was funded by the China Postdoctoral Science Foundation in the research project: “Theoretical and experimental research on a solid desiccant air-conditioning system which integrates the function of indoor air cleaning” (Project no. 2016M590036).

8 References

[1] N.A. Pennington, Humidity changer for air conditioning. U.S. Patent No.2700537, January 1955.

[2] P. Mavroudaki, C.B. Beggs, P.A. Sleigh, The potential for solar powered single-stage desiccant cooling in southern Europe, Applied Thermal Engineering, 22(3)(2002)1129–1140.

[3] P. Bourdoukan, E. Wurtz, P. Joubert, M. Sperandio, Potential of solar heat pipevacuum collectors in the desiccant cooling process: modeling and experimental results, Solar Energy, 82(2008)1209–1219.

[4] N. Enteria, H. Yoshino, A. Mochida, R. Takaki, A. Satake, R. Yoshie, T. Mita-mura, S. Baba, Construction and initial operation of the combined solar thermal and electric desiccant cooling system, Solar Energy, 83(2009)1300–1311.

[5] G. Angrisani, F. Minichiello, C. Roselli, M. Sasso, Desiccant HVAC systemdriven by a micro-CHP: experimental analysis, Energy and Buildings, 42(2010)2028–2035.

[6] G. Angrisani, F. Minichiello, C. Roselli, M. Sasso, Experimental investigation tooptimise a desiccant HVAC system coupled to a small size co-generator, Applied Thermal Engineering, 31(2011)506–512.

[7] J. Fischer, K. Mescher, B. Elkin, S.M. McCune, J. Gresham, High-performance schools, ASHRAE Journal, 49(5)(2007)30–46.

[8] Y. Xiong, J Shi, Y Wu, C Ma, H Chen, Preparation and investigation on density and surface tension of quaternary bromides for concentrating solar power, Solar Energy Materials and Solar Cells, 157(2016)709–715.

[9] C.X. Jia, Y.J. Dai, J.Y. Wu, R.Z. Wang, Analysis on a hybrid desiccant air conditioning systems, Applied Thermal Engineering, 26(2006)2393–2400.

[10] Y.F. Zhang, X.W. Hu, Z.S. Mao, L.L. Wei, C.M. Chen, H. Hao, Cycle performance of heat pump in desiccant cooling system, Journal of Chemical industry and Engineering, 20(2009) 2177–2182.

[11] Y. Sheng, Y. Zhang, N. Deng, L. Fang, J. Nie, L. Ma, Experimental analysis on performance of high temperature heat pump and desiccant wheel system, Energy and Buildings, 66(2013)505–513.

[12] H. Hao, Y.F. Zhang, G.B. Zhou, Feasibility study of moderate or high temperature heat pump coupling to desiccant wheel hybrid system, Journal of Shenyang Jianzhu University, 24(4)(2008)675–678.

[13] N. Subramanyam, M.P. Maiya, and M. Srinivasa, Application of desiccant rotor to control humidity in air-conditioning systems, Applied Thermal Engineering, 24(2004) 2777–2788.

[14] J.J. Jurinak, J.W. Mitchell, W.A. Beckman, Open-cycle desiccant air conditioning as an alternative to vapor compression cooling in residential applications, Jounal of Solar Energy Engineering, 106(1984)252–260.

[15] J.C. Sheridan, J.W. Mitchell, A hybrid solar desiccant cooling system, Solar Energy, 34(1985)187–193.

[16] R. Tu, X. Liu, Y. Jiang, Performance analysis of a two-stage desiccant cooling system, Applied energy, 113(2014), 1562–1574.

[17] L. Fang, G. Zhang, A. Wisthaler, Desiccant wheels as gas-phase absorption (GPA) air cleaners: evaluation by PTR-MS and sensory assessment, Indoor Air, 18(5)(2008) 375–385.

[18] G. Zhang, Y.F. Zhang, L. Fang, Theoretical Study of Simultaneous Water and VOCs Adsorption and Desorption in a Silica Gel Rotor, Indoor Air, 18(1)(2008) 37–43.

[19] F.E. Nia, D. Paassen, M.H. Saidi, Modeling and simulation of desiccant wheel for air conditioning, Energy and Buildings, 38(10)(2006)1230–1239.

[20] X.J. Zhang, Y.J. Dai, R.Z. Wang, A simulation study of heat and mass transfer in a honeycombed rotary desiccant dehumidifier, Applied Thermal Engineering, 23(2003)989–1003.

[21] T.S. Ge, F. Ziegler, R.Z. Wang, A mathematical model for predicting the performance of a compound desiccant wheel (A model of compound desiccant wheel), Applied Thermal Engineering, 30(2010)1005–1015.

[22] I.P. Koronaki, E. Papoutsis, V. Papaefthimiou, E. Rogdakis, Numerical and experimental analysis of a solid desiccant wheel, Thermal Science, 20(2)(2016)613–621.

[23] A. Yadav, L. Yadav, Comparative performance of desiccant wheel with effective and ordinary regeneration sector using mathematical model, Heat and Mass Transfer, 50(10)(2014)1465-1478.

[24] J.Y. San, Heat and mass transfer in a two-dimensional cross-flow regenerator with a solid conduction effect, International Journal of Heat and Mass Transfer, 36(3)(1993)633–643.

[25] CEN Report, CR 1752, "Ventilation for Buildings-Design Criteria for the Indoor Environment", European Committee for Standardization, 2001.

[26] Munters Air Treatment Equipment Co., Ltd, Manual of ML400-MLT1400 series dehumidifiers, 2014.

[27] You-Kyong Seo, J.W. Yoon, J.S. Lee, Y.K. Hwang, Chul-Ho Jun, Jong-San Chang, S. Wuttke, P. Bazin, A. Vimont, M. Daturi, S. Bourrelly, P.L. Llewellyn, P. Horcajada, C. Serre, G. Férey, Energy-efficient dehumidification over hierachically porous metal-organic frameworks as advanced water adsorbents, Advance material, 24(2012)806‑810.

[28] T.S. Ge, Y. Li, Y.J. Dai, R.Z. Wang. Investigation on a novel two-stage solar driven rotary desiccant cooling system using composite desiccant materials, Solar Energy, 84(2)(2010)157-159.

[29] C.X. Jia, Y.J. Dai, J.Y .Wu, R.Z. Wang, Use of compound desiccant to develop high performance desiccant cooling system, International journal of refrigeration, 30(2007), 345-353.

[30] M. Tokarev, L. Gordeeva, V. Romannikov, I. Glaznev, Y. Aristov, New composite sorbent $CaCl_2$ in mesopores for sorption cooling/heating, International Journal of Thermal Sciences, 41(5)(2002)470-474.

[31] J. Jeong, S. Yamaguchi, K. Saito and S. Kawai, Performance analysis of four-partition desiccant wheel and hybrid dehumidification air-conditioning system, International Journal of Refrigeration, 33(3)(2010)496-509.

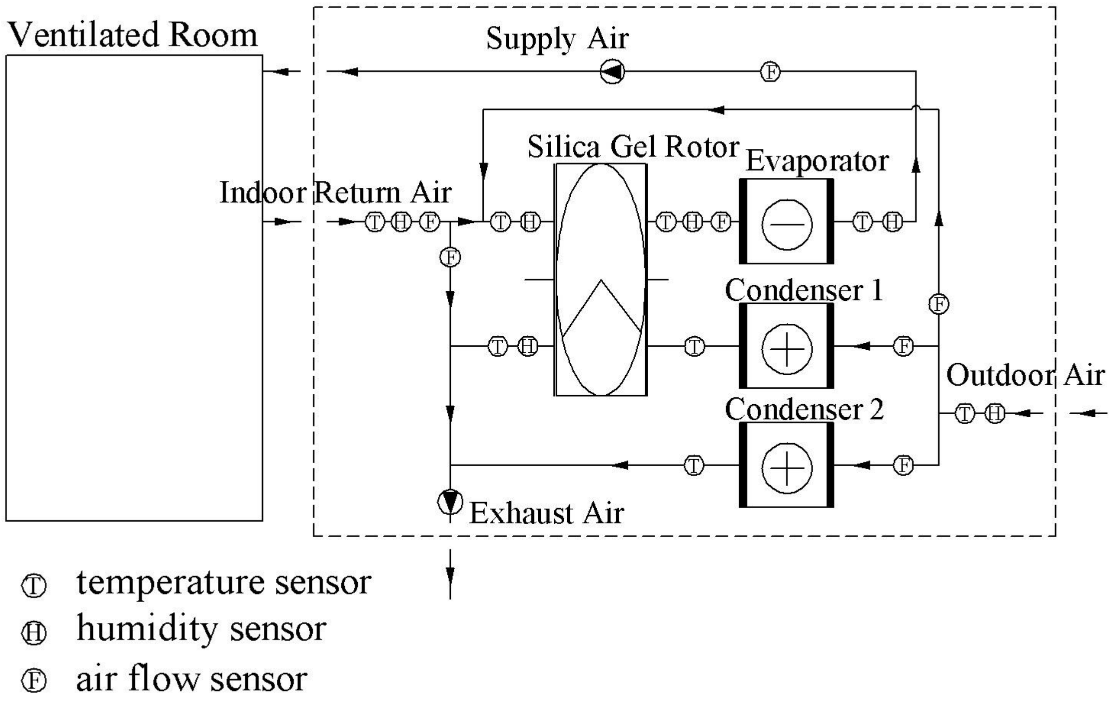
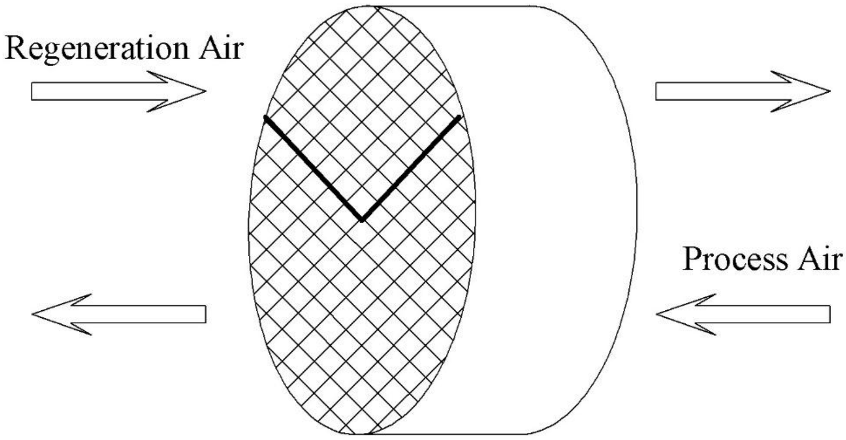
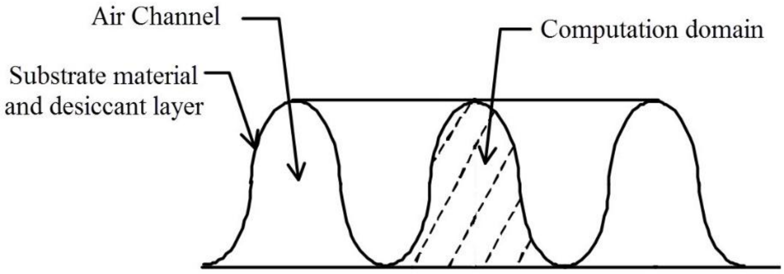


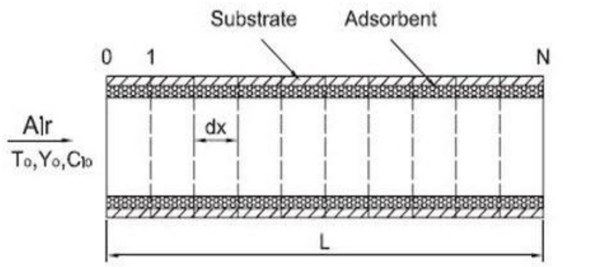
Figure 1. Schematic diagram of HP-SDC designed for summer operation mode



2-A. Desiccant rotor with counter flow



2-B. Computation domain of desiccant rotor



2-C. View along the air flow of one computation domain

Figure 2. Physical model of one desiccant rotor

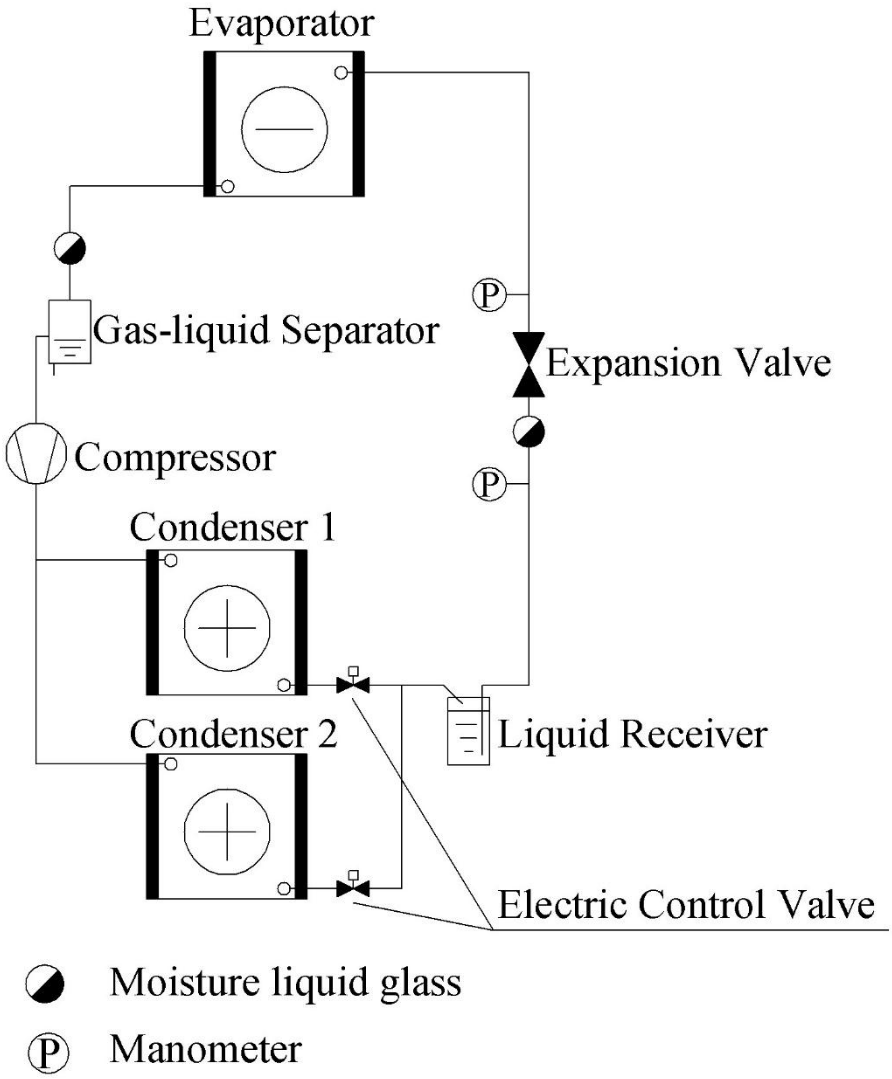


Figure 3. Heat pump designed for the HP-SDC

heat pump silica gel rotor

Figure 4. Pictures of the heat pump and the silica gel rotor for prototype HP-SDC

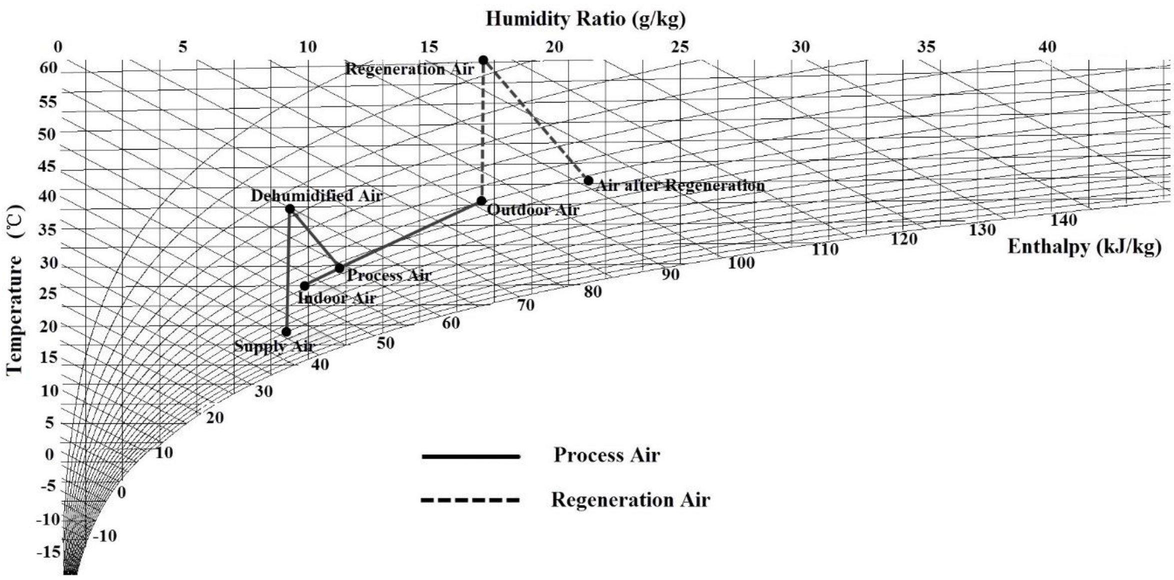


Figure 5. Thermal conditions of ventilation air and regenration air under Colombo extreme summer class

Tables

Table 1. Key components of the HP-SDC prototype unit

Components	Producer	Model	Parameters
Silica gel rotor	Munters	MLT800	Diameter:454mm, Depth:200mm
Compressor	Bizer	2GC-2.2	Maximum refrigerant displacement: 9.15m^3/h
Expansion valve	Danfoss	AKV10-6	MOPD 18bar
Evaporator	Roenest	-	500mm*500mm*8Rows
Condenser 1	Roenest	-	300mm*250mm*5Rows
Condenser 2	Roenest	-	300mm*250mm*5Rows

Table 2. Models and accuracies of the measuring instruments in the prototype HP-SDC

Measured objects	Measuring equipment	Measuring ranges	Accuracies
Temperature and Humidity Ratio	Carel DPDT010000 Duct Probe thermal meters	-20°C to 70°C, 10% to 90% RH	+/-0.5°C at 25°C, +/-0.9°C at -20°C to70°C; +/-3% RH at 25°C/50%RH, +/-6% RH at -20°C to 70°C
Airflow Rates	IRIS damper and Huba Control AG 669 pressure transmitters	From 0 m^3/h -288 m^3/h to 0 m^3/h -1260 m^3/h	10% of the measuring range

Table 3. Geometry and thermal-physical properties of the field lab

Parts of Envelope	Area(m^2)	Heat transfer coefficient(W/m^2*K)
Roof	72	0.20
External wall	21.6	0.25
External window	14.4	1.50
Interior walls and doors	72	2.00
Interior floor	72	0.20

Table 4. Indoor thermal environment set-points and inner heat conduction

Parameters		Value
Indoor temperature (°C)	Summer	25
Indoor relative humidity (%)	Summer	50
Occupants (persons)		10
Heat from lights (W)		43
Heat from computer (W)		210
Heat from projector (W)		250

Table 5. Thermal climate data of Colombo

Summer climate	Temperature (°C)	Humidity Ratio (g/kg)
Class 1	20.9	14.3
Class 2	24.7	17.6
Class 3	28.5	18.5
Class 4	32.3	18.0
Class 5	36.1	15.1
Extreme case	38.0	17.1

Table 6. Hygrothermal loads and supply air thermal conditions calculated for Colombo summer climates

Cities and Climate Classes	Indoor climate		Outdoor climate		Indoor hygrothermal load		Air delivered to room	
	Temperature (°C)	Humidity Ratio(g/kg)	Temperature (°C)	Humidity Ratio(g/kg)	Sensible Load(kW)	Latent Load(kg/h)	Temperature (°C)	Humidity Ratio(g/kg)
Colombo Summer Class 1	25	9.85	20.9	14.3	1.34	0.68	20.18	9.17
Colombo Summer Class 2	25	9.85	24.7	17.6	1.50	0.68	19.62	9.17
Colombo Summer Class 3	25	9.85	28.5	18.5	1.66	0.68	19.06	9.17
Colombo Summer Class 4	25	9.85	32.3	18	1.81	0.68	18.49	9.17
Colombo Summer Class 5	25	9.85	36.1	15.1	1.97	0.68	17.93	9.17
Colombo Summer Extreme Class	25	9.85	38	17.1	2.05	0.68	17.65	9.17

Table 7. Airflow rates of the HP-SDC prototype unit

Airflow	Flow rates(L/s)
Recirculation air	190
Outdoor air to field lab	40
Regeneration air	115
Air for excess condensing heat	130
Exhaust air from room	40

Table 8. Implications of airflow simulated and measured

Airflow	Implication
Outdoor air	Air taken from outdoor climates
Indoor air	Air taken from the ventilated room
Process air	Indoor recirculation air mixed with outdoor air entering desiccant rotor for dehumidification and cleaning
Dehumidified air	Ventilation air dehumidified by desiccant rotor
Sensible cooled air	Air cooled by evaporator and delivered to ventilation room
Regeneration air	Outdoor air heated by condenser used to regenerate desiccant rotor
Air after regeneration	Air has regenerated desiccant rotor at outlet of desiccant rotor
Excess heat air	Air used to take excess heat from condensing heat

Table 9. Theoretical simulated and experimental measured thermal conditions of outlet process air and regeneration air （T is the temperature, HR is the humidity ratio）

Colombo Thermal Environments	Process air		Regeneration air		Dehumidified air				Air after regeneration			
	T (ºC)	HR(g/kg)	T (ºC)	HR(g/kg)	T (ºC)	T (ºC)	HR (g/kg)	HR (g/kg)	T (ºC)	T (ºC)	HR (g/kg)	HR (g/kg)
	--	--	--	--	simulated	measured	simulated	measured	simulated	measured	simulated	measured
Summer Class 1	24.37	10.65	49.74	14.01	31.89	31.61	8.37	9.22	34.25	34.52	18.79	16.99
Summer Class 2	25.24	11.29	60.16	17.49	35.47	34.64	8.20	9.43	38.94	39.01	24.08	21.58
Summer Class 3	26.07	11.52	64.33	18.18	37.37	36.26	8.08	9.41	40.84	41.1	25.54	23.11
Summer Class 4	26.48	11.34	61.1	18.3	36.37	35.68	8.39	9.45	40.57	40.47	24.60	21.96
Summer Class 5	27.89	10.89	57.62	15.09	36.57	36.05	8.25	9.07	39.68	39.54	20.67	18.63
Summer Extreme Class	28.15	11.29	62.17	17.04	38.01	37.1	8.30	9.29	41.73	41.54	23.39	21.25

Table 10. Experimental measured thermal conditions of process air and regeneration air at different locations of HP-SDC（T is the temperature, HR is the humidity ratio）

Colombo Thermal Climates	Outdoor air		Indoor air		Process air		Dehumidified air		Sensible cooled air		Regeneration air		Air after regeneration		Excess heat air	
	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)	T (ºC)	HR (g/kg)
Summer Class 1	20.63	14.01	25.08	9.84	24.37	10.65	31.61	9.22	19.63	9.24	49.74	14.01	34.52	16.99	23.12	14.01
Summer Class 2	24.98	17.49	25.06	9.79	25.24	11.29	34.64	9.43	17.97	9.39	60.16	17.49	39.01	21.58	28.55	17.49
Summer Class 3	28.37	18.18	25.36	9.9	26.07	11.52	36.26	9.41	17.96	9.4	64.33	18.18	41.1	23.11	32.95	18.18
Summer Class 4	32.52	18.3	24.99	9.8	26.48	11.34	35.68	9.45	18.25	9.42	61.1	18.3	40.47	21.96	37.2	18.3
Summer Class 5	36.03	15.09	25.7	9.98	27.89	10.89	36.05	9.07	18.13	9.16	57.62	15.09	39.54	18.63	55.92	15.09
Summer Extreme Class	38.28	17.04	25.46	9.93	28.15	11.29	37.1	9.29	18.3	9.15	62.17	17.04	41.54	21.25	53.16	17.04

Table 11. Energy consumption and COP of the heat pump in HP-SDC

Colombo Thermal Climates	Cooling load (kW)	Power input (kW)	Coefficient of Performance
Summer Class 1	3.42	0.99	3.47
Summer Class 2	4.78	1.77	2.70
Summer Class 3	5.25	2.08	2.53
Summer Class 4	5.04	1.72	2.92
Summer Class 5	5.13	1.66	3.09
Summer Extreme Class	5.39	1.96	2.75