Effect of changes in the regeneration temperature and the regeneration airflow velocity on the system performance of the solid desiccant air conditioning

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Abstract. This paper presents the effect of changes in operating parameters on the solid desiccant air conditioning system performance. It has consisted of the regeneration temperature and the regeneration airflow velocity parameters. These parameters were examined in two operation modes, the former ranging from 2 m/s–0.83 m/s at constant regeneration temperature, and the latter ranging from 75°C–46.4°C at constant regeneration airflow velocity. The results indicate a smaller decline in dehumidification performance for the constant regeneration temperature operating mode than the constant airflow velocity operating mode. On the contrary, the system performance and the recovery energy ratio at the latter operating mode gives a higher value than the former operating mode. These results suggest that when the dehumidification performance becomes the main function in the system, the regeneration airflow velocity should not be too low to avoid a significant reduction in system performance.

1. Introduction

Since the 1930s, solid desiccant technology has been used in humidity control for industrial applications in the United States [1]. Unfortunately, the use of this technology for heating, ventilation, and air conditioning (HVAC) applications is not feasible because performance and the additional cost do not scale. In the early 1960s, a new method for improving the performance of solid desiccant systems was invented by N.A. Pennington [2] and Carl. G. Munters [3]. Introduced by Pennington, a rotary heat exchanger matrix in the form of a wheel was impregnated with a desiccant material to make an adiabatic regenerative dehumidifier when coupled to a heat source and a double evaporative cooler. This operating cycle is widely known as the ventilation cycle. Up to now, several studies on the solid desiccant air conditioning (SDAC) have been carried out focusing on system performance [4], desiccant materials [5], and strategy to implement.

One of the highlights of this technology is its ability to reduce energy consumption by utilizing a low-grade heat source. Besides, the advantage of SDAC is to reduce the initial equipment cost because this system can separate the handling of latent heat loads and sensible heat loads. Even the desiccant material able to absorb the undesirable particle, and thus increase indoor air quality. Therefore, the
SDAC system is simpler, safer, more durable, and easier to implement for air conditioning applications [6,7]. SDAC comprises a rotor that impregnated with desiccant material, a heater, and a sensible heat exchanger. The rotor needs to be regenerated by flowing the hot air with a specific temperature range. This process defines as the regeneration process, which is intended to break the bond between moisture and desiccant material in the rotor. With the presence of this process, the desiccant wheel can operate continuously and keep its performance.

In our previous work, we have investigated the effect of changes in the regeneration heat input on the performance of the desiccant wheel. The regeneration heat input is considered to rise and drop based on changing the load factor. The changing load factors are represented by a variation of the regeneration temperature \(T_{\text{reg}}\) and the regeneration airflow velocity \(V_{\text{reg}}\) that examined in two operation modes [8]. The result showed that the changes in heat input are influencing the decreases in the desiccant wheel performance. Comparison among two operating modes, the constant regeneration air temperature operation had a smaller drop in dehumidifying performance when the amount of heat input to the regeneration air decreased. It was also found that the optimal rotor rotation speed is proportional to the regeneration airflow rate that can be proposed as a solution for recovering the decline of the dehumidification performance.

In this paper, we present the further investigation of the performance of SDAC with a combination of a sensible heat exchanger (SHE). We want to clarify the effect of changes in the regeneration temperature \(T_{\text{reg}}\) and the regeneration airflow velocity \(V_{\text{reg}}\) on the system performance of SDAC. We expected that the existence of SHE is not only to reduce the outlet temperature of the process air but also preheats the regeneration airflow that can reduce the external energy input for the regeneration process and improve the system performance. Two operating modes, a constant \(T_{\text{reg}}\) mode with changing of \(V_{\text{reg}}\) from 2 m/s to 0.83 m/s, and a constant \(V_{\text{reg}}\) mode at various \(T_{\text{reg}}\) from 75°C–46.4°C, were used as the operating parameters to examine the effectiveness of SHE on the system performance. Finally, this investigation can use as a reference for implementing the SDAC in real applications.

2. System description
The experimental apparatus (figure 1) consists of a desiccant rotor, a sensible heat exchanger, a set of control devices, and a measurement instrument. The system was designed with two flow paths: the regeneration side and the adsorption side. Each flow path has a heater, an air control box, a blower, and a regeneration hot air. The desiccant rotor has a diameter of 0.32 m and a width of 0.2 m with an effective cross-sectional area of 0.066 m². It is formed from a fiberglass-based honeycomb matrix impregnated with silica gel of about 50 wt%. The crossflow heat exchanger type plate was adopted as the sensible heat equipment to further remove more sensible load of the outlet process air.

The desiccant wheel was divided into two chambers. In both chambers, six thermocouples (Type-T, accuracy = ±0.5°C) were installed for air temperature measurements at the inlet and outlet of the desiccant rotor and distributed uniformly at angular intervals of 30°. The thermocouples were also installed at the inlet and outlet of the hot and fresh air that flows in the sensible heat exchanger. A chilled mirror dew-point hygrometer (Optica, GE Sensing & Inspection Technologies and GE Optimization and Control Co., Ltd., Boston, United States., accuracy = ±0.2°C in dew point) was used to measure the absolute humidity at the inlet and outlet of the wheel. The airflow rate of the system was monitored using a pitot tube-type flow meter (New Aero Eye, Wetmaster Co., Ltd., Tokyo, Japan., accuracy = ±3%RD). From the measured airflow, the surface air velocity was calculated from the cross-sectional area of the adsorption (or regeneration) zone of the desiccant wheel.
Figure 1. A schematic diagram of the experimental apparatus.

2.1 Experimental conditions
The inlet condition of the outside air (OA) and the regeneration air (RA) were represented by air states 1 and 4 respectively, as shown in figure 2. The inlet OA condition was fixed at 30°C and 15 g/kgDA, while the RA condition was set at 26°C and 10.5 g/kgDA. To reach the temperature and humidity targets on both conditions, the air control box that equipped with the humidifier and heater was controlled separately in each air handling unit. The adsorption process can be seen on the airflow from the air state 1 to the air state 3, and the regeneration process is shown at the airflow state conditions of 4-7. The outside air at state point 1 enters the desiccant wheel (DW) with a constant process airflow rate \( V_{pro} \) of 2 m/s, in which it is dehumidified and heated by the regeneration air at state point 6. At state point 2 to 3, the hot dry air is then cooled in the heat exchanger, using a crossflow type sensible heat exchanger.

Figure 2. The inlet conditions for the adsorption and regeneration air.
Meanwhile, the regeneration air at state point 4 is preheated in the sensible heat exchanger, using the waste heat air at state point 2. Further, the preheated regeneration air at state point 5 is heated until the predetermined regeneration temperature through a heater. Then, the air state at point 6 is passed through the desiccant wheel at which desiccant material dries up moisture from the adsorption process air and transferred to the hot regenerated air. Finally, the leaving air at the desiccant wheel (the air state at point 7) is exhausted to the ambient air. The regeneration air temperature \((T_{\text{reg}})\) and the regeneration airflow velocity \((V_{\text{reg}})\) fitted to a specified of values that depend on the operating modes.

In the constant regeneration air temperature operating mode, \(V_{\text{reg}}\) has a varied value from 2 m/s to 0.83 m/s, while \(T_{\text{reg}}\) was kept constant at 75°C. For the constant regeneration airflow velocity operating mode, \(T_{\text{reg}}\) was varied from 75°C to 46.4°C with maintaining \(V_{\text{reg}}\) at 2 m/s. In addition, the rotor rotation speed was fixed at 35 rph.

2.2 Performance indexes

In order to investigate the effect of changes in \(T_{\text{reg}}\) and \(V_{\text{reg}}\) on the system performance of SDAC, the following indexes were analyzed.

1. **Dehumidification performance** \((\Delta X)\)

   The dehumidification capacity represents the ability of the desiccant wheel to remove moisture from processed air; here, \(\Delta X\) is defined as the absolute humidity difference between the inlet \((X_{\text{OA}})\) and outlet \((X_{\text{SA}})\) of the processed air. Specifically, we obtain
   \[
   \Delta X = X_{\text{OA}} - X_{\text{SA}} \, [\text{g/kgDA}],
   \]  
   (1)

2. **Efficiency of the sensible heat exchanger** \((\varepsilon_{\text{SHE}})\)

   The heat transfer efficiency of the sensible heat exchanger refers to the air heating through the wheel on the adsorption process and the thermal energy supplied for the regeneration process [9] that can be calculated as follows:
   \[
   \varepsilon_{\text{SHE}} = \frac{(T_{\text{SA}} - T_{\text{SA}'})(T_{\text{SA}} - T_{\text{RA}})}{\text{Q}_{\text{tot}}}
   \]  
   (2)

   where \(T_{\text{SA}}, T_{\text{SA}'}, \) and \(T_{\text{RA}}\) are the outlet temperature of the process air at state point 2 (°C), the outlet temperature of the process air at state point 3 or the air state after the SHE (°C), and the inlet temperature of the regeneration air at state point 4 (°C), respectively.

3. **Recovery energy ratio** \((\text{RER})\)

   The recovery energy ratio defined as the ratio between the heat recovery \((\text{Q}_{\text{rec}}, \text{kw})\) and the total energy \((\text{Q}_{\text{tot}}, \text{kw})\) that used in the regeneration process. It was estimated as follows:
   \[
   \text{RER} = \frac{\text{Q}_{\text{rec}}}{\text{Q}_{\text{in}}}
   \]  
   (3)

   \[
   \text{Q}_{\text{rec}} = m_{\text{reg}}(h_{\text{RA'}} - h_{\text{RA}})
   \]  
   (4)

   \[
   \text{Q}_{\text{in}} = m_{\text{reg}}(h_{\text{RA}} - h_{\text{RA'}})
   \]  
   (5)

   where, \(h_{\text{RA}}, h_{\text{RA'}}, h_{\text{RA}}\) are the enthalpy of the regeneration air at state point 6 or the air state after the heater (kJ/kg), and the enthalpy of the preheated regeneration air at state point 5 or the air state before entering heater (kJ/kg), and the enthalpy of the inlet regeneration air or the air state at point 4 (kJ/kg), respectively.

4. **Coefficient of performance** \((\text{COP})\)

   COP is the ratio between the cooling capacity \((Q_{\text{a}})\) and the total heat input \((Q_{\text{in}})\). COP represents the overall performance of the desiccant system that calculated from the following equation.
   \[
   \text{COP} = \frac{Q_{\text{a}}}{Q_{\text{in}}}
   \]  
   (6)

   \[
   Q_{\text{a}} = m_{\text{ads}}(h_{\text{OA}} - h_{\text{SA}})
   \]  
   (7)
where \( h_{OA} \) and \( h_{SA} \) are the enthalpy of the outside air (OA) at state point 1 (kJ/kg) and the supply air (SA) at state point 3 (kJ/kg), respectively.

This experiment was substantiated by verifying the mass conservation (\( \Delta M \)) between the adsorption and regeneration sides using the estimates of airflow rates and the difference in the absolute humidity between inlet and outlet. The mass conservation equation reads [6]

\[
\Delta M = \frac{m_{ads}(x_{OA} - x_{SA}) - m_{reg}(x_{FA} - x_{RA})}{m_{ads}(x_{OA} - x_{SA})},
\]

(8)

where \( m_{ads} \) and \( m_{reg} \) are the mass flow rates [kg-DA/s] in the adsorption and the regeneration sides, \( x_{OA} \), \( x_{SA} \), \( x_{EA} \), and \( x_{RA} \) are the absolute humidity of the outside air, the supply air, the exhaust air, and the regeneration air, respectively. In all experiments, it was confirmed that \( \Delta M \) was less than 5%. Taking into account the experimental uncertainty for the measuring instrument, the maximum amounts of relative uncertainty obtained for \( \Delta X \), \( \varepsilon_{SHE} \), RER, and \( COP \) are 2.7%, 1.0%, 3.1% and 2.2%, respectively.

3. Result & Discussions

3.1 Effect of the constant regeneration temperature operating mode

In the first experiment, the regeneration temperature (\( T_{reg} \)) was set to 75°C while the regeneration airflow velocity (\( V_{reg} \)) was varied from 2 m/s to 0.83 m/s (figure 3). The rotor rotation speed was fixed at 35 rph. As shown in figure 3, the dehumidification performance (\( \Delta X \)) of the desiccant declined along with a reduction in the mass flow rate of the regeneration stream. This condition diminishes the capacity of the desiccant wheel to take more moisture from processed air. The decline in dehumidification performance is not so high as long as the airflow velocity is between 2 m/s and 1.25 m/s or about 1.8 g/kgDA in absolute humidity. However, when the regeneration airflow velocity is 0.83 m/s, the air state of the exhaust after regeneration reaches a limit value, and the dehumidification performance is constrained to maintain the material balance. This phenomenon causes a significant decline in dehumidification performance of 3.7 g/kgDA. This result suggests that the dehumidification performance declines sharply when the regeneration airflow rate is too low. A similar trend is also seen in COP, which increases as the \( V_{reg} \) increase. This means that the rise in \( V_{reg} \) causes the ratio between the regeneration heat input (\( Q_{in} \)) and the cooling capacity (\( Q_c \)) to increase.

Furthermore, the ratio between the heat recovery and the total energy input, the recovery energy ratio (RER) of the system, is shown in figure 4 at the left vertical axis. The result shows that the changes in \( V_{reg} \) from 2 m/s to 0.83 m/s affect the RER drops from 53% to 42% or a drop of 21%. It can be explained that the reduction in the amount of regeneration air that less than the amount of adsorption air (process air) leads to an imbalance in performance between the adsorption process and the regeneration process. This means that the rotor on the adsorption side could not extract more moisture from the processed air because of the rotor starting to saturate. Consequently, the amount of regeneration heat is more used to heat the rotor rather than heating the regeneration air that causes the supply air temperature at point 2 to decrease. This condition directly declines the effectiveness of SHE, as shown in Figure 4 (on the right vertical axis). Thus, we can conclude that the increment in \( V_{reg} \) affects a rise in the energy recovery ratio (RER) and the effectiveness of SHE, and conversely.
**Figure 3.** Effect of changes in $V_{reg}$ on the dehumidification performance ($\Delta X$) and COP at the constant $T_{reg}$ operating mode.

**Figure 4.** Effect of $V_{reg}$ changes on the recovery energy ratio (RER) and the effectiveness of the sensible heat exchanger (SHE) at the constant $T_{reg}$ operating mode.
Figure 5. Effect of changes in $V_{\text{reg}}$ on the regeneration heat input ($Q_{\text{in}}$), the heat recovery ($Q_{\text{rec}}$), and the cooling capacity ($Q_a$).

Figure 5 shows that a rise in $V_{\text{reg}}$ from 0.83 m/s to 2 m/s leads to an increase in recovering waste heat from the rotor from 0.16 kW to 0.5 kW. The presence of the sensible heat exchanger is not only to recover the waste heat but also to increase the cooling capacity. This condition can be seen in Figure 5 (in the right vertical axis) that presents the cooling capacity ($Q_a$) enhancement along with the rise in $V_{\text{reg}}$. As regards the regeneration heat input ($Q_{\text{in}}$), the rise in $V_{\text{reg}}$ causes an increase in the enthalpy of the regeneration air that affects the regeneration heat input to increase. Thus, a rise in $V_{\text{reg}}$ at the constant $T_{\text{reg}}$ causes an increase in $Q_{\text{rec}}$, $Q_{\text{in}}$, and $Q_a$.

3.2 Effect of the constant regeneration airflow velocity operating mode

Figure 6 shows the dehumidification performance ($\Delta X$) of the SDAC system when the regeneration air temperature ($T_{\text{reg}}$) varies between 75°C to 46.4°C. This operating mode was examined at a constant regeneration airflow velocity of 2 m/s. The absolute humidity of the regeneration air was fixed at 10.5 g/kgDA, while the regeneration temperature was varied. The result shows that the amount of dehumidification tends to drop of 4 g/kgDA as the temperature of regeneration is decreased from 75°C to 46.4°C. It can be explained that a decrease in regeneration temperature is followed by a rise in the dehumidification limit. This means that there is a decrease in the adsorption thrust force that causes the dehumidification performance to decline. However, there is a decrease in system performance (COP) as an increased regeneration temperature. This is due to an increase in the amount of heat to achieve a higher $T_{\text{reg}}$ that leads to an increase in the heat input of the system.
Figure 6. Effect of changes in $T_{\text{reg}}$ on the dehumidification performance ($\Delta X$) and COP at the constant $V_{\text{reg}}$ operating mode

In figure 7, the effectiveness of SHE is almost not affected by changes in $T_{\text{reg}}$. This result indicates that the effectiveness of SHE is more influenced by $V_{\text{reg}}$ than $T_{\text{reg}}$. However, the recovery energy ratio (RER) increases when the regeneration temperature decreases. The reduction in $T_{\text{reg}}$ from 75°C to 46.4°C causes an increase in RER from 53% to 61% or about 16%. Furthermore, figure 8 shows that the increase in $T_{\text{reg}}$ affects the heat recovery ($Q_{\text{rec}}$), the regeneration heat input ($Q_{\text{in}}$), and the cooling capacity ($Q_{a}$) to increase. However, it was found that at the lowest $T_{\text{reg}}$ of 46.4°C, the cooling capacity ($Q_{a}$) is higher than the regeneration heat input ($Q_{\text{in}}$). As a result, there is a significant increase in system performance (COP) and recovery energy ratio (RER) when the lowest of $T_{\text{reg}}$ is used. However, it should be noted that the dehumidification performance gives the most deficient performance. Therefore, it is necessary to find out the optimal operating mode by considering the changes in $V_{\text{reg}}$ and $T_{\text{reg}}$ at various rotor rotation speed.

Figure 7. Effect of $T_{\text{reg}}$ changes on the recovery energy ratio (RER) and the effectiveness of the sensible heat exchanger (SHE) at the constant $V_{\text{reg}}$ operating mode
4. Conclusions

In this present work, the effect of changes in the regeneration airflow velocity ($V_{\text{reg}}$) and the regeneration temperature ($T_{\text{reg}}$) on the system performance of the solid desiccant air conditioning has been obtained in two operating modes, i.e., the constant regeneration temperature mode and the constant regeneration airflow velocity mode. Regarding the dehumidification performance, the constant regeneration temperature ($T_{\text{reg}}$) operating mode offers better performance than that for the constant airflow velocity ($V_{\text{reg}}$) operating mode. On the contrary, the system performance (COP) and the recovery energy ratio (RER) at the constant $V_{\text{reg}}$ operating mode gives a higher value than the constant $T_{\text{reg}}$ operating mode. These results suggest that when the dehumidification performance becomes the main function of the system, the reduction in $V_{\text{reg}}$ should be considered to avoid a significant reduction in the system performance. Further investigations are needed to find out the optimal operating mode by considering the changes in $V_{\text{reg}}$ and $T_{\text{reg}}$ at various rotor rotation speed. By understanding its behavior, we can determine the effective configuration that provides the optimal performance of the solid desiccant air conditioning system.

### Nomenclature

| Symbol | Description | Unit |
|--------|-------------|------|
| $COP$  | coefficient of performance | [-] |
| $h$    | enthalpy | [kJ/kg] |
| $\Delta M$ | mass conservation | [%] |
| $\dot{m}$ | regeneration mass flow rate | [kg/s] |
| $\dot{m}_{\text{ads}}$ | adsorption mass flow rate | [kg-DA/s] |
| $\dot{m}_{\text{reg}}$ | regeneration mass flow rate | [kg-DA/s] |
| $Q_{\text{in}}$ | regeneration heat input | [kW] |
| $Q_{\text{rec}}$ | heat recovery | [kW] |
| $Q_{\text{tot}}$ | total regeneration heat input | [kW] |
| $RER$  | recovery energy ratio | [%] |
| $T$    | temperature | [°C] |

### Abbreviations

- DW: desiccant wheel
- EA: exhaust air
- OA: outside air
- RA: regeneration air
- RA’: regeneration air after pre-heat (the air state at point 5)
- rph: rotations per hour
- SA: supply air
- SA’: final supply air (the air state at point 3)
| Symbol | Description                              |
|--------|------------------------------------------|
| $T_{reg}$ | regeneration temperature [°C]             |
| $V_{pro}$ | adsorption inlet air velocity [m/s]       |
| $V_{reg}$ | regeneration inlet air velocity [m/s]     |
| $X$    | absolute humidity [g/kg-DA]              |
| $\Delta X$ | dehumidification capacity [g/kg-DA]     |

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