Feasibility study on an energy-saving desiccant wheel system with CO\(_2\) heat pump

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Abstract. In traditional desiccant wheel, air regeneration process occurs inside an open loop, and lots of energy is consumed. In this paper, an energy-saving desiccant wheel system with CO\(_2\) heat pump and closed loop air regeneration is proposed. The general theory and features of the desiccant wheel are analysed. The main feature of the proposed system is that the air regeneration process occurs inside a closed loop, and a CO\(_2\) heat pump is utilized inside this loop for the air regeneration process as well as supplying cooling for the process air. The simulation results show that the proposed system can save significant energy.

1. Introduction

Desiccant wheels are also known as dehumidification wheels, wheel dehumidifiers or rotary dehumidifiers in different literatures. With the requirements of energy saving and emission reduction in the world, desiccant wheels have been expanded greatly from special areas such as electronics, food, storage and pharmaceutical industries, to supermarkets, restaurants, schools, hospitals, and office buildings[1]. Most research works focus on the performance analysis of desiccant systems. Many works analyze the performance of hybrid vapor compression-desiccant wheel systems and show that this technology can save significant energy compared to the traditional systems where the vapor compression system is used for dehumidification [2-13]. The biggest advantage of hybrid vapor compression-desiccant wheels system lies in the elimination of overcooling and reheating.

In the above mentioned research studies, it is reported that the air regeneration process of the desiccant wheel occurs inside an open cycle, which needs lots of energy for regeneration.
In this paper, a new desiccant wheel system with closed-loop air regeneration is introduced and analyzed. In a transcritical CO₂ cycle heat pump, the cooling process in the supercritical region is a gas cooling process, which is particularly suitable for the regenerative heating of air. The regeneration air can be heated to the desired temperatures, and no auxiliary heater is needed. Therefore, a CO₂ transcritical cycle heat pump is utilized in the proposed system. In addition, the air regeneration process occurs inside a closed loop. The CO₂ transcritical cycle heat pump is utilized to not only regenerate the recycled air, but also to cool the supply air.

The aim of this work is to study the performance of the proposed system, as well as explore the energy saving potential.

2. Methodologies
In order to provide universality for the air dehumidification process, only fresh air processing is considered in the simulation, and the fresh air is from the ambient.

Figure 1 illustrates the schematic of a traditional desiccant wheel-cooling system. In the further analysis, this system is named System 1. As shown in Figure 1, the desiccant wheel is divided into two sections for the process air and the regeneration air. Water vapor is adsorbed by the desiccant when the moist air stream passes through the process air side. The wheel rotates then exposes the moisture-laden desiccant to the regeneration air stream, which is heated up by an electric heater before entering the desiccant wheel. Water is desorbed from the desiccant and the desiccant material is regenerated. The hot and humid regeneration air leaving the desiccant wheel is then discharged to the ambient. Significant heat energy is required for the regeneration process. After the process air leaves the desiccant wheel at Point B, it is cooled in Evaporator I of a vapor compression cycle, obtaining supply air with the required temperature and humidity ratio at Point C[2].

Figure 2 illustrates an advanced high temperature heat pump coupled to desiccant wheel-cooling system. In the further analysis, this system is named System 2. Compared to System 1, the difference is only on the regeneration air side: the airstream from the ambient is first pre-heated through the heat rejected from the condenser before being further heated to the desired temperature by an electric heater[2].
Figure 2. Schematic of advanced high temperature heat pump coupled to desiccant wheel-cooling system (System 2).

Figure 3. Schematic of proposed desiccant wheel system with closed loop air regeneration (System 3).
A schematic of the proposed system, the new desiccant wheel system with closed loop air regeneration, is illustrated schematically in Figure 3. In the further analysis, this system is named System 3. The air regeneration process occurs inside a closed loop. A transcritical CO₂ heat pump is utilized for the recycled air regeneration including dehumidification, cooling and heating process. In addition, the CO₂ heat pump supplies cooling for the supply air. In the CO₂ heat pump, there are two evaporators: Evaporator I is utilized to cool the supply air, and Evaporator II is utilized for cooling and dehumidification of the recycled air in the closed loop. The pressure-enthalpy diagram and temperature-entropy diagram of the CO₂ heat pump cycle are shown in Figure 4.

3. Mathematical model of desiccant wheels

According to Beccali et al.[9-10], the outlet humidity ratio (Xₐ) and temperature (Tₐ) of the process air in a desiccant wheel can be calculated by the following expressions:

\[
X_\text{out} = e^{0.053T_\text{out}} \left(0.9428 \text{UR}_{\text{reg}} + 0.0572 \text{UR}_{\text{in}}\right) - 1.7976 / 18.671 \tag{1}
\]

\[
\left(\text{UR}_{\text{out}} e^{0.053T_\text{out}} - 1.7976\right) / 1867.0 = (h_\text{out} - 1.006 T_\text{out}) / (2501 - 1.805 T_\text{out}) \tag{2}
\]

\[
\text{UR}_{\text{out}} = (0.9428 \text{UR}_{\text{reg}} + 0.0572 \text{UR}_{\text{in}}) \tag{3}
\]

A desiccant wheel type-I (Silica Gel) is selected for the three systems. The air enthalpy leaving the desiccant wheel, hₐ, is calculated by the following expression:

\[
h_\text{out} = (0.1312 h_{\text{reg}} + 0.8688 h_{\text{in}}) \tag{4}
\]

The enthalpy (hₐ) is a function of humidity ratio Xₐ and the regeneration temperature Tₐ. It can be expressed by the following well-established equation:

\[
h_{\text{reg}} = 1.01 T_{\text{reg}} + (2500 + C_p T_{\text{reg}}) X_{\text{reg}} / 1000 \tag{5}
\]

Where Cₚ is the specific heat of the moisture.

For the calculation of relative humidity URₐ, the following empirical relation has been developed:

\[
\text{UR}_{\text{reg}} = (18.6715 X_{\text{reg}} + 1.7976) e^{0.053T_{\text{reg}}} \tag{6}
\]

The desiccant wheel runs with identical volume air-flows on the air process side and the regeneration air side. The ranges of available field data for the given model are as follows:

\[X_{\text{reg}} = 10-16 \text{ g kg}^{-1}, T_{\text{reg}} = 40-80 ^\circ\text{C}, T_{\text{in}} = 20-35 ^\circ\text{C}, X_{\text{in}} = 8-15 \text{ g kg}^{-1}.\]
The established model consists of ten variables and six equations. Therefore, the values of four variables have to be given. Usually, the parameters \( h_{in}, U_{in}, \) and \( X_{out} \) are known based on the given operating conditions. In addition, the value of \( X_{reg} \) can be calculated when the evaporating temperature of the CO\(_2\) heat pump (i.e., \( T_e \)) and the heat transfer temperature difference are given.

In addition, the performance of CO\(_2\) heat pump (System 3) is: \[
\text{COP} = \frac{(h_2-h_5)}{(h_2-h_1)}
\]

### 4. Performance simulation and results

The performance and energy savings analysis are based on the supply air conditions: the dew point of the supply air is between 5 °C and 0 °C, and the dry-bulb temperature is 20 °C. In addition, the fresh air conditions from the ambient are based on ARI standard rating conditions and typical ambient conditions. The operating conditions of four typical case studies are listed in Table 1. The dew point of the supply air in Case 1 is the same as the one in Case 2 with different fresh air conditions. The mass flow rate of process air and regeneration air streams are both assumed as 1 kg s\(^{-1}\).

The variations of the regeneration temperature, \( T_{reg} \), of System 3 as a function of the evaporating temperature, \( T_e \), for the four cases are shown in Figure 5. It can be seen that \( T_{reg} \) increases with an increase of \( T_e \) for all cases.

| No. | Point A: Fresh air | Point B: Dew point of air after the desiccant wheel on process air side | Point C: Supply air |
|-----|-------------------|-------------------------------------------------|---------------------|
| Case 1 | ARI standard rating condition: \( T_{db} = 26.7 \) °C, \( X_{in} = 11.23 \) g kg\(^{-1} \) | \( T_{dp} = 5 \) °C (i.e., \( X_{out} = 5.46 \) g kg\(^{-1} \)) | \( T_{dp} = 5 \) °C, \( T_{db} = 20 \) °C |
| Case 2 | Typical outdoor air condition: \( T_{db} = 20 \) °C, \( X_{in} = 8.8 \) g kg\(^{-1} \) | \( T_{dp} = 5 \) °C (i.e., \( X_{out} = 5.46 \) g kg\(^{-1} \)) | \( T_{dp} = 5 \) °C, \( T_{db} = 20 \) °C |
| Case 3 | \( T_{db} = 20 \) °C, \( X_{in} = 8.8 \) g kg\(^{-1} \) | \( T_{dp} = 3 \) °C (i.e., \( X_{out} = 4.7 \) g kg\(^{-1} \)) | \( T_{dp} = 3 \) °C, \( T_{db} = 20 \) °C |
| Case 4 | \( T_{db} = 20 \) °C, \( X_{in} = 8.8 \) g kg\(^{-1} \) | \( T_{dp} = 0 \) °C (i.e., \( X_{out} = 3.8 \) g kg\(^{-1} \)) | \( T_{dp} = 0 \) °C, \( T_{db} = 20 \) °C |

**Figure 5.** Variations of required regeneration temperature \( T_{reg} \) with evaporating temperature \( T_e \) in System 3 in the four cases.
The variations of the COP and the discharge pressure, P, of the CO₂ heat pump of System 3 as a function of the evaporating temperature, Tₑ, for all Cases are shown in Figure 6. It can be found that the COP increases and then decreases with an increase of the evaporating temperature, Tₑ, in every case. Thus, there is an optimal COP when Tₑ varies from 1 °C to 15 °C.

![Figure 6. Variations of COP of the CO₂ heat pump with evaporating temperature Te in System 3.](image)

An energy saving comparison for Case 1 is listed in Table 2. According to the total cooling load and COP, the compressor power is calculated. Electric heater supplies supplementary heat in System 2. The energy saving rates of System 3 for all cases are listed in Table 3. It can be found that System 3 can save significant energy.

| Items                          | System 1 | System 2 | System 3 |
|-------------------------------|----------|----------|----------|
| Point A: Fresh air from ambient | Tᵢn = 26.7 °C, Tᵳ,in = 19.4 °C |
| Point B: Air at the outlet of desiccant wheel on air process side | Tₒut = 47.8 °C, Xₒut = 5.46 g kg⁻¹ | Tₒut = 46.3 °C, Xₒut = 5.46 g kg⁻¹ |
| Point D: Regeneration air | Tᵲᵣₑᵢₙ = 68.4 °C, Xᵲᵣₑᵢₙ = 11.23 g kg⁻¹ | Tᵲᵣₑᵢₙ = 63.5 °C, Xᵲᵣₑᵢₙ = 9.57 g kg⁻¹ |
| Cooling load of evaporator I, (kW) | 28.3 | 26.8 |
| Cooling load of evaporator II, (kW) | — | 33.0 |
| Heating load for regeneration, (kW) | 43.0 | 50.7 |
| COP | 5.2 | 4.1 | 6.3 |
| Power of compressor, (kW) | 5.4 | 6.9 | 9.5 |
| Heating capacity from condenser/gas coolers, (kW) | — | 35.2 | 50.7 |
| Power of the electric heater, (kW) | 43.0 | 7.8 | — |
| Total energy consumption, (kW) | 48.4 | 14.7 | 9.5 |
| Energy-saving rate compared to System 1 | — | 70% | 80% |
| Energy-saving rate compared to System 2 | -230% | — | 35% |

| Items                                      | Case 1 | Case 2 | Case 3 | Case 4 |
|--------------------------------------------|-------|-------|-------|-------|
| Compared to System 1, energy-saving rate   | 80%   | 84%   | 84%   | 84%   |
| Compared to System 2, energy-saving rate   | 35%   | 70%   | 66%   | 68%   |
5. Results and discussion

In order to reduce the energy consumption and greenhouse gas emissions of desiccant wheel systems, an energy-saving desiccant wheel system with closed loop air regeneration is proposed and analyzed. A transcritical CO$_2$ heat pump is recommended as a high temperature heat pump in the proposed system. The mathematical model of the system is developed, and the operating performance of the systems are simulated and analyzed. The results show that the proposed system can save 35%-84% energy than conventional systems and advanced high temperature heat pump systems.

Accurate control system is the key point for this kind of system.

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