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EXPERIMENTAL INVESTIGATIONS OF SOLAR-ASSISTED ADSORPTION DEHUMIDIFICATION

Summary

Air dehumidification is energy-intensive; adsorption dehumidifiers can use low-grade heat, making it viable to be used with solar thermal heat. This work experimentally investigates a commercially available desiccant wheel that is extended by a water-to-air heat exchanger for solar thermal heat integration. For the successful integration of solar thermal heat, the regeneration temperature of the desiccant wheel plays a major role. It can be reduced from 93 °C to 72 °C without losses in moisture removal rate or efficiency, when regenerating electrically and the regeneration air volume flow rate is increased to maintain a constant regeneration heat flow rate. Efficiency losses occur when heat is integrated with the water-to-air heat exchanger due to preheated purge air. These losses can be compensated by increasing the total regeneration heat flow rate. This way, solar thermal heat can substitute electricity with a factor of 0.71 to 0.84 on a commercially available desiccant wheel.

Keywords: Dehumidification, Desiccant Wheel, Solar Thermal, Experiments

Field of application for the findings: Solar Air Conditioning

Addressed audience: Researchers

1. Introduction

Air dehumidification is an important and energy intensive task to supply conditioned air. Dehumidified air is needed for indoor air-conditioning for occupant comfort or building preservation and for various industrial processes. Adsorption dehumidifiers are an alternative to conventional vapor compression condensing dehumidifiers, which suffer efficiency losses at low dew point temperatures and low relative humidity. Such applications are for example waterworks or gas pressure regulating stations where dew point temperatures between 0 and 10 °C are needed to prevent condensation on pipe surfaces carrying cold fluids. Desiccant wheels are divided into multiple sections; the process air section, the regeneration air section and often additional sections for heat recovery. In the regeneration air section, air is heated from ambient temperature to temperatures of 100 to 150 °C to release the humidity that was adsorbed by the desiccant. Therefore, the driving energy for adsorption dehumidifiers is low- to medium-grade heat, which is mostly provided electrically, for commercially available desiccant wheels with process air volume flow rates < 3000 $m^3\ h^{-1}$. In Central Europe, the dehumidification load aligns well with solar irradiation. Therefore, solar thermal heat is well suited to (partially) replace electrical energy for regeneration. To increase solar fractions and solar thermal efficiency, lower regeneration temperatures are crucial. As summarized by Ge et al. (2014) many simulation studies, but few experimental studies, have been performed on combined solar thermal systems and desiccant wheels. The regeneration temperature of desiccant wheels has been investigated in numerous experimental studies. A decrease in moisture removal rate (MRR), the dehumidified vapor mass flow rate of the process air, was observed when lowering the regeneration temperature with constant volume flow rates (Angrisani et al., 2012; Eicker et al., 2012; Zendehboudi et al., 2018; Su et al., 2020; Ma et al., 2023). The regeneration heat flow rate was thus also decreased. Angrisani et al. (2012) reported a decrease in MRR for constant heat flow rates and lower regeneration temperatures by increasing the volume flow rate. Eicker et al. (2012) demonstrated an increase in MRR when increasing the regeneration volume flow rate while maintaining a constant regeneration temperature. In contrast to the aforementioned studies, this work investigates the influence of different regeneration temperatures on the moisture removal rate for a constant regeneration heat flow rate using a commercially available desiccant wheel. The overall research goal is to investigate the integration of solar thermal heat into a commercially available desiccant wheel. Furthermore, the interaction between solar thermal preheating and additional electrical heating under consideration of a purge air section is assessed.

2. Experimental Setup and Measurements

Fig. 1 shows a diagram of the laboratory test rig including measurement devices. The test rig consists of a commercially available silica gel desiccant rotor with a frequency-controlled motor, an electrical heater and a purge air section to cool the wheel material between the desorption and adsorption processes. To integrate solar heat, a water-to-air heat exchanger is placed in the regeneration air stream. The desiccant wheel has nominal volume flow rates of 690 $m^3 h^{-1}$ and 254 $m^3 h^{-1}$ for process air and regeneration air respectively. The process air section has 70 %, the regeneration air section 25 % and the purge air section 5 % of the cross-sectional area. The regeneration (R,in) and process inlet air (P,in) are provided by the same air conditioning unit. The regeneration air is heated by the water-to-air heat exchanger that represents solar heating (R,sol). Then, part of the regeneration air passes through the purge air section, while the remaining air is heated electrically (R,heat). These air streams are mixed and pass through the regeneration section (R,out). The process air inlet air (P,in) passes directly through the desiccant wheel (P,out) and is then cooled by a water-to-air heat exchanger to increase measurement accuracy (P,cooled).

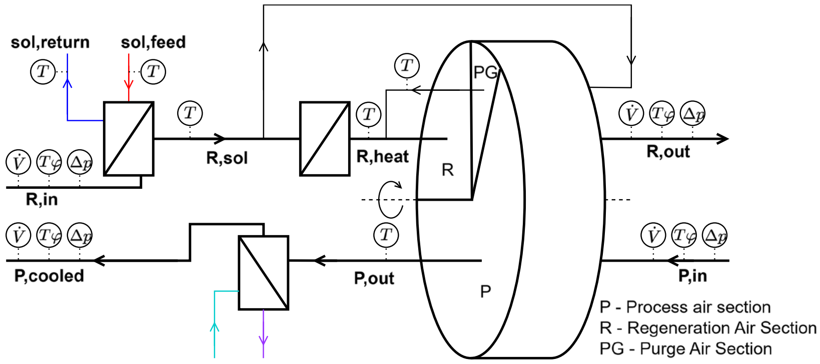

Fig. 1: Diagram of the solar desiccant wheel test rig with nomenclature and measurement points.

As a measure for the dehumidification performance, the MRR is evaluated, which is the vapor mass flow rate from the process air to the desiccant wheel.

$$\dot{m}_{\mathrm{vap,P}} = \dot{m}_{\mathrm{air,P}} \cdot (\omega_{\mathrm{P,in}} - \omega_{\mathrm{P,out}}) \qquad \text{(eq. 1)}$$

$\dot{m}_{\mathrm{air,P}}$ is the mass flow rate of the dry air of the process air stream and $\omega_{\mathrm{P,in}}$ and $\omega_{\mathrm{P,out}}$ are the humidity ratios of the process air before and after the desiccant wheel.

3. Results

To compare the performance of the desiccant wheel with constant heat flow rates, the regeneration temperature is decreased by increasing the volume flow rate of the regeneration air. To reduce the regeneration temperature from 93 °C to 65 °C, an increase of the regeneration volume flow rate from 254 $m^3 h^{-1}$ to 440 $m^3 h^{-1}$ is needed. Fig. 2 shows the resulting MRR for only electrical heating (blue) and for solar preheating with a constant solar feed temperature $T_{\mathrm{sol,feed}}$ of 70 °C (pink). For the electrical heating, $\dot{m}_{\mathrm{vap,P}}$ significantly decreases only for the lowest investigated $T_{\mathrm{R,heat}}$ of 64 °C. Above that, $\dot{m}_{\mathrm{vap,P}}$ remains almost constant. When the regeneration air is preheated by the water-to-air heat exchanger, $\dot{m}_{\mathrm{vap,P}}$ decreases by 18 % with lower regeneration temperatures (98 °C to 67 °C). The losses in dehumidification performance can be attributed to the reduced effect of the purge air due to the preheated regeneration air. This effect was clearly visible when keeping the total heat flow rate and air volume rates constant while varying the preheat temperature $T_{\mathrm{R,sol}}$. This way, the regeneration temperature $T_{\mathrm{R,heat}}$ remains the same while the air that enters the purge air section increases with rising $T_{\mathrm{R,sol}}$. The resulting $\dot{m}_{\mathrm{vap,P}}$ decreases by 10 % when increasing the solar preheat temperature $T_{\mathrm{R,sol}}$ from 40 °C to 85 °C. This reduction can be attributed to the preheated purge air. To compensate for the reduction in dehumidification when using solar heat with the commercially available desiccant wheel (with a purge air section), the total regeneration heat, the sum of $\dot{Q}_{\mathrm{sol}}$ and $\dot{Q}_{\mathrm{el}}$ must be increased to maintain the nominal dehumidification. For the desiccant wheel in the test rig, this leads to an increase of $T_{\mathrm{R,heat}}$ from 93 °C without preheating to 107 °C (with solar thermal preheating to 85 °C) to maintain the nominal MRR of around 5.4 kg h^{-1}. Fig. 3 shows the heat flow rates for a varying solar preheat temperature under nominal working conditions. With rising solar preheat temperatures, $\dot{Q}_{\mathrm{sol}}$ increases while $\dot{Q}_{\mathrm{el}}$ decreases. The ratio of substituted electrical energy (compared to pure electrical regeneration) to solar heat is between 0.71 and 0.84. The highest

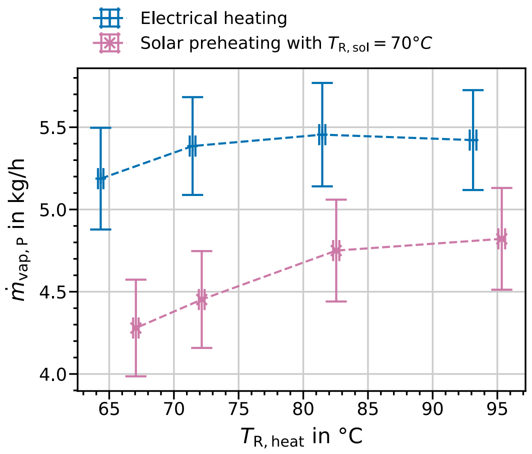

Fig. 2: Vapor mass flow rate for varying regeneration temperature $T_{\mathrm{R,heat}}$, keeping the heat flow rate constant by varying the regeneration air flow rate $\dot{V}_{\mathrm{R,in}}$

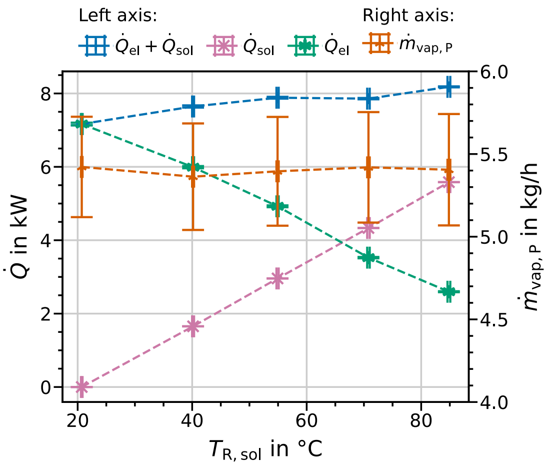

Fig. 3: Electrical, solar and total heat needed to achieve $\dot{m}_{\mathrm{vap,P}}$ of 5.4 kg h^{-1} with varying $T_{\mathrm{R,sol}}$ and nominal air volume flow rates.

ratio is achieved for a solar preheat temperature of 71 °C

4. Conclusion

Using a commercially available desiccant wheel with purge air section, the regeneration temperature can be reduced from 93 °C down to 72 °C without significant losses in MRR, when the heat for regeneration is kept constant by increasing the volume flow rate of the regeneration air. If in addition solar heat is used for preheating the air, efficiency losses occur due to the preheating of the purge air, a consequence of used desiccant wheels design, which is common in commercially available devices of that capacity range. To achieve the same MRR, solar heat can substitute electrical energy with a factor between 0.71 and 0.84 using nominal working conditions and a commercially available desiccant wheel complemented with a water-to-air heat exchanger in the regeneration air stream. To improve the use of solar heat in desiccant wheels, the solar heat should be integrated after the purge air is separated from the regeneration air stream. Additionally, when using lower regeneration temperatures, the segmentation of the desiccant wheel might be adjusted to account for the different ratio of volume flow rates.

5. References

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