

Desiccant Cooling with Solar Energy

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Desiccant cooling systems combine sorptive dehumidification, heat recovery, evaporation and heating to create a cooling process which can offer energy savings compared to conventional air conditioning systems. Waste heat or solar energy can be used for the required regeneration of the sorbents in the dehumidifier, leading to further energy savings. Parametric studies, particularly of the dehumidifier, have been undertaken. These show that it may be possible to reduce the regeneration air flow without a significant reduction in the dehumidification efficiency, enabling desiccant cooling systems to run with high COPs. The results of the measurements were used as input parameters for a new dynamic simulation program, which was specifically developed for the purpose of assessing the potential for desiccant cooling under different climatic conditions.

Introduction

Cooling is important for space conditioning of most buildings in warm climates and in non-residential buildings with high internal cooling loads in cooler climates. Along with the growth of cooling loads in modern buildings with low thermal inertia, the peak electricity demand increases during the day. This causes problems concerning the energy supply in some countries, as the demand for air conditioning occurs simultaneously with the maximum demand of industry and households. The inertia of energy plants leads to a high consumption of coal, oil, gas and nuclear energy by energy companies to ensure the provision of electricity. To depress the swing of energy demand during the day, solar energy for cooling purposes could make a significant contribution to lowering the energy consumption. The solar cooling technology presented in this paper is based on a process combining sorptive dehumidification, heat recovery, evaporation and heating. The so-called desiccant cooling system takes air from outside or from the building, dehumidifies it with a solid or liquid desiccant, cools it by heat exchange and then evaporatively cools it to the desired state of a maximum 100% relative humidity in the inlet air stream. The desiccant must be regenerated by heat. This can be achieved with solar energy from solar air collectors or liquid collectors.

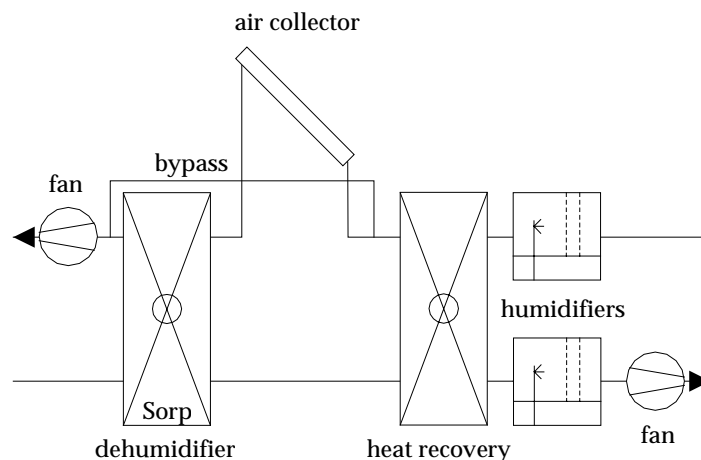


Figure 1: Desiccant cooling system with solar air collectors

Figure 1 shows how solar air collectors can be integrated. Adaptation to liquid collectors is also possible with an air/liquid heat exchanger. By using solar liquid collectors the storage possibility can increase the content of solar energy to the required auxiliary energy. The example in Figure 2 shows the desiccant cooling process under usual design conditions for middle Europe ($\vartheta_{\text{amb}} = 32^{\circ}\text{C}$, $\text{rh} = 40\%$; $\vartheta_{\text{in}} = 26^{\circ}\text{C}$, $\text{rh} = 55\%$). Figure 3 shows the process in a psychrometric chart, where the state points are numbered to correspond to the points on the process schematic. Ambient air is dried and heated by a dehumidifier from 1 to 2, regeneratively cooled by exhaust air from 2 to 3, evaporatively cooled from 3 to 4 and then brought into the building. The inlet humidifier allows control of the temperature and humidity. Exhaust air at state 5 from the inside of the building moves in the opposite direction and is evaporatively cooled to 6 up to saturation. At 7 the air is heated by the energy removed from the heat regenerator. From 7 to 8 solar or other heat must increase the temperature up to the regeneration level of the desiccant to regenerate the dehumidifier.

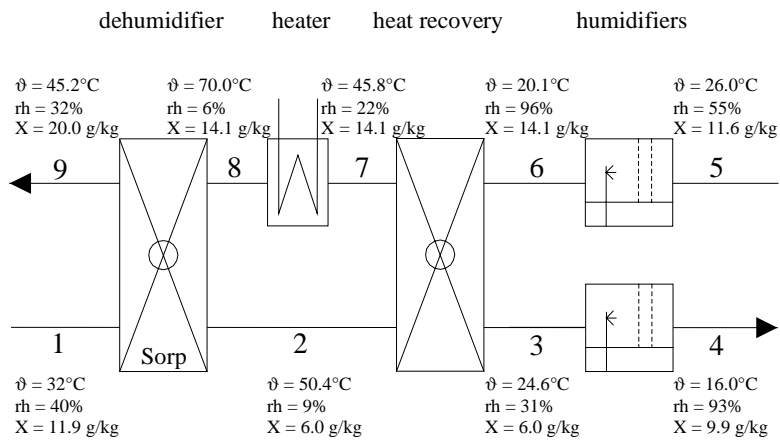


Figure 2: Conditions of air in a desiccant cooling system. A bypass flow of 25% is considered.

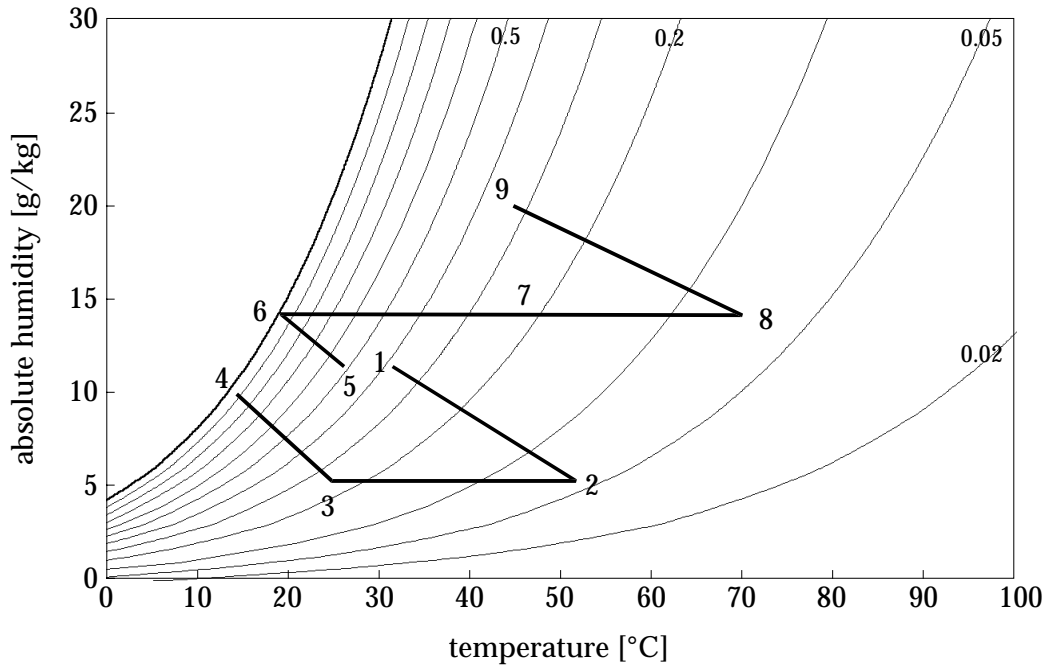


Figure 3: Process of a desiccant cooling system plotted in a psychrometric chart

Measurements

The world-wide first test plant for open cycle desiccant cooling systems with solar air collectors has been installed at the Hochschule für Technik in Stuttgart. With the modular setup, a quick change of components is possible. This enables parameter studies to be conducted easily. The test plant was developed under aspects of very precise measurement conditions. With regard to the choice of sensors, the measurement of temperatures and relative humidity do not cause problems. However, measuring air flow in compact plants is very complicated. A totally developed symmetrical flow profile is only possible in very long ductwork. Turbulences appear at bends and expansions of ducts and falsify the measurement results. This was the reason for using long tubes to connect the components of the desiccant cooling systems. Each component was connected to the tubes with special concentric adapters.

The measurements of the rotating dehumidifier were of greatest interest, and hence were investigated in detail. The dehumidification achieved depends on the inlet conditions of the entering air, on the velocities in the process and the regeneration section and the regeneration temperature and humidity. Furthermore, the rotation speed influences the performance significantly. Fifteen rounds per hour were determined as optimal for dehumidification. A faster rotation leads to incomplete regeneration and dehumidification. Lower rotation speeds lead to a lower dehumidification, as the maximum possible water content in the adsorbens is reached some time before entering the regeneration section. In the regeneration section the water is removed before entering the process section resulting in an unfavourable increase in the adsorbens temperature. This causes a reduced adsorption in the process section, and the adsorbens must first be cooled down by the inlet air until the adsorption process can restart.

For the measurements of the dehumidification, the process air flow was constant at $V_P = 2530$ m³/h while the regeneration air flow was changed from $V_R = 2530$ m³/h down to $V_R = 890$ m³/h. This corresponds to a ratio of $V_R/V_P = 1.00$ down to $V_R/V_P = 0.35$. To obtain relevant statements on the performance of the dehumidification with low regeneration air flows, it was necessary to improve the seals of the dehumidifier. With the non-improved seals, noticeable leakage air flows appeared due to the pressure difference in the different flow sections and valid measurement results were not achievable.

The maximum possible dehumidification depends on the thermodynamic properties of the entering air streams. The process air can be maximally dehumidified down to $rh = \text{constant}$, which is marked by the condition of the regeneration air. The ratio of the reached value for dehumidification to the ideal dehumidification is defined as the dehumidification efficiency. The results of the dehumidification efficiencies η_{sorp} of this study are

$$\begin{array}{ll} V_R/V_P = 0.35 & \eta_{\text{sorp}} = 0.62 \\ V_R/V_P = 0.50 & \eta_{\text{sorp}} = 0.75 \\ V_R/V_P = 0.75 & \eta_{\text{sorp}} = 0.82 \\ V_R/V_P = 1.00 & \eta_{\text{sorp}} = 0.87 \end{array}$$

Particularly interesting in these results is the fact that the dehumidification efficiency decreases only by 7% while the air flow is reduced from $V_R/V_P = 0.75$, which is the usual operating mode, to $V_R/V_P = 0.50$. An air flow ratio of $V_R/V_P = 0.35$ would decrease the dehumidification efficiency by 20%. A balanced air flow with $V_R/V_P = 1.00$ results in a dehumidification efficiency which is only 5% higher than in the usual operating mode. A reduction of the regeneration air flow is important for saving energy, as the energy input for regeneration is linearly dependent on the regeneration air flow.

The small reduction of the dehumidification efficiency compared to the reduction of air flow can be explained with the faster desorption velocity compared to the adsorption velocity. This can be justified with the assumption that capillary condensation takes place in micropores which are placed in front of macropores while these are still filled with vapour. So the adsorption process will be slowed down. During desorption the macropores are filled with liquid water at the same vapour pressure and the desorption process will not be delayed. A further justification is the fact that the heat and mass transfer coefficients only change with the square root of the air velocity.

Simulations

For the task of this work the program ESIMA, which is an abbreviation of "Energy Simulation in Mathematica", was developed. It is a modular simulation tool for dynamic thermal building simulations and was evaluated with the BESTEST procedure [1]. It was developed to calculate the performance of buildings and plants and their interaction. The simulations presented here were made with a room analogous to BESTEST case 940, which represents a heavyweight building. This building with a floor area of 48 m² was assumed to be occupied by 5 persons, of whom emit 100 W of sensible heat and 30 g/h of moisture. Furthermore, an internal cooling load of 20 W/m² was assumed during the time of occupancy between 8 a.m. and 6 p.m. A constant air change of 0.2 1/h was fixed to take building leakage into account. The indoor air temperatures were set to a minimum of 20°C during occupancy time and were set back to 15°C outside the occupancy time. The maximum permissible indoor air temperatures vary according to DIN 1946 depending on the ambient air temperature. A desiccant cooling system pro-

grammed with the algorithm provided in [2] was connected to this building. During the occupancy time a fresh air demand from the desiccant cooling system of a minimum of 150 m³/h and a maximum of 518 m³/h, which equals an air change of 4 1/h, was considered. The constant parameters of this system were a heat recovery efficiency of 0.85 and a humidifier efficiency of 0.90. For the dehumidifier the efficiencies measured at the test plant were used. The system can operate in different modes listed below depending on the indoor air conditions.

1. Heat recovery mode HR. Used for highly efficient heat recovery enabling the heat recovery and the dehumidifier to be used as an enthalpy exchanger. Enabled for $\vartheta_i < 22^\circ\text{C}$.
2. Ventilation mode V. Used only for ventilation without enabling any thermal component of the system. Enabled for $22^\circ\text{C} \leq \vartheta_i < 23^\circ\text{C}$.
3. Adiabatic cooling mode AC. Used for cooling if dehumidification is not necessary or inlet temperatures are low enough, enabling the outlet humidifier and the heat recovery wheel. Enabled for $23^\circ\text{C} \leq \vartheta_i < 24^\circ\text{C}$.
4. Desiccant cooling mode DCS. Classic operating mode enabling all components of the entire system. Enabled for $\vartheta_i \geq 24^\circ\text{C}$.
5. Desiccant cooling mode DCS*. Used if indoor relative humidity is over 60%. Classic operating mode enabling all components except the inlet humidifier. Enabled for $\vartheta_i \geq 24^\circ\text{C}$.

The heat supply of the desiccant cooling system is only achieved with solar air collectors. This signifies that no energy storage is available in either the desiccant cooling system or the solar system. A thermal inertia is only provided by the mass of the building. One of the main parameters describing the performance of an entire desiccant cooling system is the coefficient of performance, abbreviated as COP. For the calculation of the COP it is usual to use enthalpy differences instead of temperature differences to show the influence of dehumidification. To calculate the enthalpies required for the calculation of the COP, each condition of air along the system must be evaluated. These conditions can only be calculated with a knowledge of the components performance.

The described test building and the adapted desiccant cooling system with solar air collectors were simulated for the climatic situations of Djakarta in Indonesia, Phoenix (Arizona) in the United States, Seville in Spain and Stuttgart in Germany. The climatic data for each location were produced with the program Meteonorm 4.0 [3].

Table 1: Results of annual simulations for the performance of a desiccant cooling system adapted to a heavyweight building in different climates.

	Mean COP only for DCS mode DCS* mode [-]	Additional cooling demand [kWh/m ² a]	DCS system cooling energy [kWh/m ² a]	Peak cooling load [W/m ²]	Control mode during operation [-]	
Djakarta	4.04	218.4	2.6	67.8	HR V AC DCS DCS*	0.00 0.00 0.00 0.12 0.88
Phoenix	1.02	54.5	160.4	96.3	HR V AC	0.01 0.09 0.07

					DCS	0.70
					DCS*	0.13
Seville	1.12	34.9	121.7	89.1	HR	0.04
					V	0.11
					AC	0.07
					DCS	0.58
					DCS*	0.20
Stuttgart	0.57	6.9	58.0	40.8	HR	0.36
					V	0.16
					AC	0.09
					DCS	0.36
					DCS*	0.02

The simulation results show that in heavyweight buildings the use of solar energy alone without any storage system can cover 89% of the cooling demand in Stuttgart, 78% of the cooling demand in Seville, 75% of the cooling demand in Phoenix and only 1% of the cooling demand in Djakarta. In the tropical climate of Djakarta or during humid periods in Phoenix and Seville, it is necessary to disable the function of the inlet humidifier (control mode DCS*), resulting in an inlet air temperature close to the room conditions. Hence the cooling capacity of the system is very low. On the other hand, the simulations show that in humid climates the gained COPs of the desiccant cooling systems are high and reach values greater than 1. This shows that it is extremely useful to install desiccant cooling systems working only for sorptive dehumidification, which is less energy-consuming than dehumidification in common air conditioning systems, where the air must be cooled to below or close to the dew point temperature. For the requirements of deep dehumidification in climates with an ambient-air absolute humidity often greater than 15 g/kg, an additional component for dehumidification is necessary. Therefore, a second rotating dehumidifier and a second heat recovery can be used as shown in Figure 4. A desiccant cooling system with this configuration can cover 29% of the cooling demand.

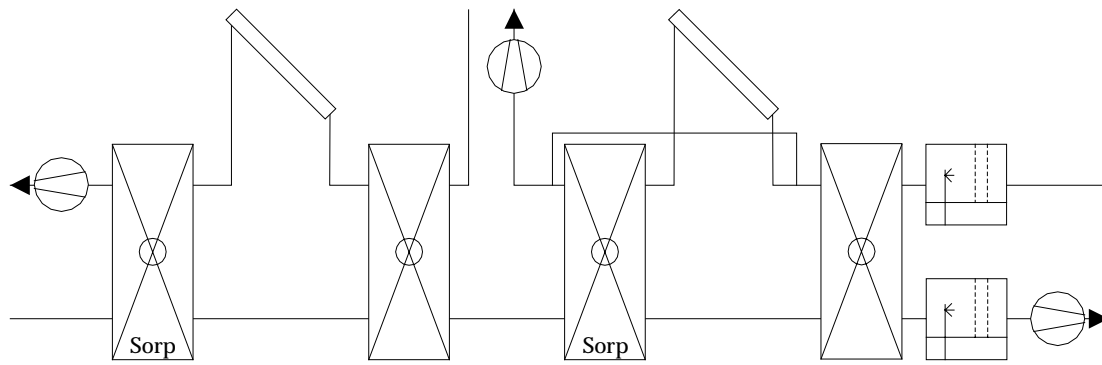


Figure 4: Desiccant cooling system with solar air collectors for deep dehumidification

Conclusions

Using solar energy for cooling purposes is an attractive idea with good prospects for conventional air conditioning systems. The replacement of compressor cooling systems by solar driven desiccant cooling systems or a combination of both could make an important contribution to environmental protection. The main argument for the applicability of solar energy is that cooling loads and solar availability are approximately in phase. Due to the combination of solar cooling and heating, solar collectors which are used only for heating purposes would become more economical. Parametric studies showed that it may be possible to reduce the regeneration air flow without a significant reduction in the dehumidification efficiency, enabling desiccant cooling systems to run with high COPs. Results of annual simulations showed the potential for desiccant cooling under different climatic conditions. The largest solar fractions were evaluated for climatic conditions of Central Europe. In climates with high ambient-air absolute humidities, additional dehumidification devices are necessary. Therefore, a sophisticated configuration of a desiccant cooling system was presented where a second heat recovery and a second dehumidifier is used.

References

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