THE POTENTIAL OF SOLAR ENERGY USE IN DESICCANT COOLING CYCLES

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ABSTRACT

The use of heat produced by solar thermal collectors is an interesting option for thermal driven air conditioning processes. A thermal driven cooling technique which fits well to non-tracking solar collectors is the desiccant cooling technique. Recently several projects have been carried out which focus on the connection of desiccant cooling systems with solar thermal energy for regeneration of the sorbents. This communication deals with three main topics: (1) Experiences achieved in a realized system which is coupled to a solar collector are discussed, (2) a new concept is presented, in which a solar air collector is integrated into the desiccant cooling cycle as only heat source and (3) a comparative study is presented which compares system performance for different system configurations and different climatic situations.

KEYWORDS

Desiccant cooling, solar thermal energy, combined systems, solar air collector

INTRODUCTION

Desiccant cooling with rotating dehumidification wheels has been discussed for a long time as an interesting option for solar driven air conditioning, see e.g. Robinson 1992; Lof 1992. During the last ten years this technique received increased attention, since it works without conventional refrigerants and it allows for the use of low temperature heat to drive the cooling cycle. The basic ventilation cycle, shown in Figure 1, produces the cooling effect by evaporitve cooling; however, the potential of using evaporative cooling is increased due to the dehumidification of air by the desiccant (e.g. silica gel). In this cycle, in general the dehumidification process has to suffice two purposes: dehumidification in order to match indoor comfort criteria and an 'extra' dehumidification in order to allow for subsequent humidification to produce the cooling effect. A combination of this cycle with a conventional vapour compression chiller can be a promising option. It allows to remove the latent load mainly with a thermal driven sorption wheel and to remove the sensible load mainly with the chiller. Since the chiller works at higher evaporator temperatures, if dehumidification is done by sorption, it works with higher efficieny. For both options - the pure ventilation cycle as well as combined systems - the use of solar thermal energy has been investigated by means of measurements and detailed simulations.

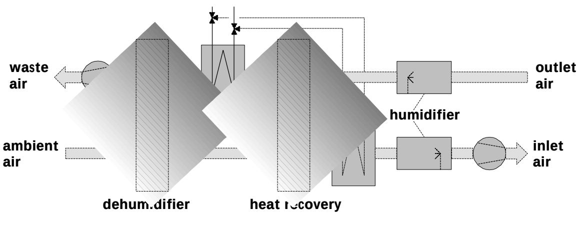

Figure 1: Ventilation cycle desiccant cooling system

EXPERIENCES WITH A REALIZED SYSTEM

A solar assisted desiccant cooling pilot plant has been installed at a technology center (TGZ Riesa) in Riesa/Saxony in 1996. Specificiations of this system are given in Table 1; the installation is shown in Figure 2. During the last 1 ½ years a monitoring program was carried out in order to optimize system control and to achieve practical experiences with this technology.

room volume seminar room	330 m^3
nominal air volume flow	2700 m^3/h
flat plate collector (liquid heat carrier)	20 m^2
hot water buffer storage	2 m^3

Table 1: Specifications of the Riesa system

Figure 2: Solar assisted desiccant cooling system Riesa: collector and air handling unit installed on the roof

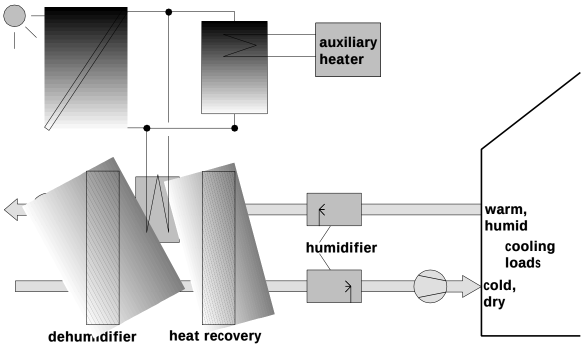

Figure 3: Scheme of the solar assisted desiccant cooling system in Riesa

System design

The system was designed using the simulation software TRNSYS (TRNSYS 1994). For the design of desiccant cooling systems two new subroutines (so called Types) were developed: Type 275 which models the desiccant cooling system consisting of the components as shown in Figure 1 and Type 276 which controls the desiccant cooling system. The overall simulation consists of the building (cooling load), the desiccant cooling system including control and the heat supply system which consists of the solar collector, a buffer storage and an auxiliary heater. The solar assisted desiccant cooling system is shown in Figure 3. In Figure 4 the dependence of the solar fraction of thermal energy required for cooling is given as a function of the collector area for different values of the buffer storage volume. The figure indicates that with the specifications choosen according to Table 1 a solar fraction for cooling of about 80 % can be expected.

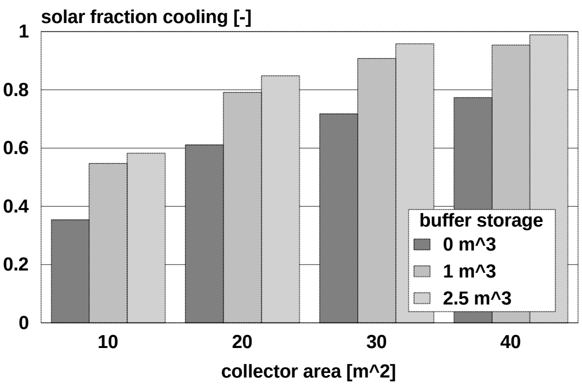

Figure 4: Solar fraction for cooling as function of collector and buffer storage size

Control of the desiccant cooling system

Two main issues have to be considered with regard to control the desiccant cooling system: Firstly, five operation modes of the system are possible depending on outdoor conditions and cooling/heating load. Those are: active heating, heat recovery, free ventilation, adiabatic cooling, desiccant cooling. Secondly, the ventilators of the desiccant cooling system can be controlled in order to get variable flow rates to match the cooling load, if the hygienic air change rate does not suffice. Detailed simulation calculations indicate that it is advantageous to operate the system with the minimum acceptable air flow rate in any operation mode and to increase the air flow rate only in the active modes, i.e. active heating or desiccant cooling. Operation according to these rules minimizes primary energy demand of the system remarkably. A typical time pattern which demonstrates the control of the desiccant cooling system is shown in Figure 5. The curves demonstrate that the room temperatures remained relatively constant at 19°C (inlet) and 25°C (outlet), though the ambient temperature was varying between 23°C and 31°C. In the late afternoon the system switched to adiabatic cooling due to reduced cooling loads.

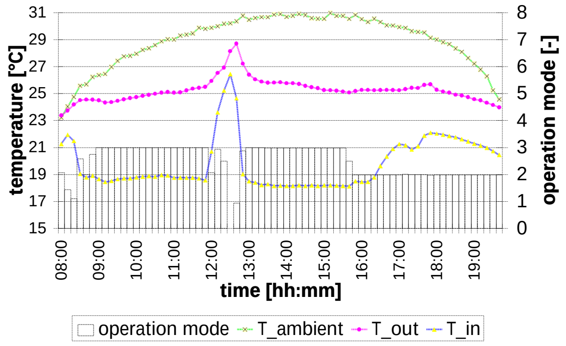

Figure 5: Time pattern of temperature and operation modes on August, 11, 1997. The system was switched off during lunch time. Operation modes: 1 = free ventilation, 2 = adiabatic cooling, 3 = desiccant cooling

Operation of the solar system

The heat supply system can also be driven in different operation modes (modes only given for cooling): (1) direct operation of the desiccant cooling system with the solar collector, (2) storage loading with solar energy (no cooling load occurs or adiabatic cooling sufficient), (3) preheating of the heating fluid by the solar collector and backup by the thermal buffer storage (solar radiation is not sufficient for cooling) and (4) driving heat comes completely from the buffer storage (no solar energy at required temperature level available). The upper part of the buffer storage is heated with an auxiliary heater such that the temperature does not fall below a minimum value (set temperature). Again detailed system simulations were used in order to determine the best value of this set temperature. The results are shown in Figure 6. The graphic shows the required primary energy demand for the system as function of the minimum regeneration temperature. This regeneration temperature is about 5 K below the set temperature of the storage due to heat exchanger temperature losses. The figure indicates that it is advantageous to keep the set temperature at the lowest acceptable value, i.e. to drive the system with the lowest acceptable temperature. This result can be interpreted as follows: purged sensible cooling loads are given as

$$P_{load,sensible} = \dot{m}_{air} \cdot c_{air} \cdot \left(T_{out} - T_{in}\right) \quad .$$

The air flow is realized with electricity (ventilators), the usable temperature difference by thermal energy (regeneration air). The average regeneration temperature increases, if a higher value of the set temperature is choosen. With increasing regeneration temperatures the primary energy demand for the ventilators decreases on the one hand but the required thermal energy for regeneration increases on the other hand. Since simulataneously the solar fraction decreases with increasing regeneration temperatures, the best result is achieved at lowest acceptable regeneration temperatures. The system in Riesa is operated with a set temperature of 60°C in the upper part of the buffer storage.

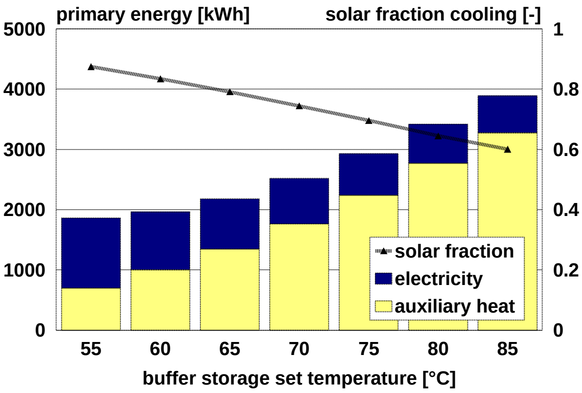

Figure 6: Primary energy consumption and solar fraction for cooling as function of minimum the regeneration temperature, which is defined by the set temperature in the buffer storage

Measured results

Results from measurements carried out during the year 1997 indicate that a solar fraction for cooling in the range of 76 % has been achieved. The overall collector efficiency during a week with tpyical summer conditions was about 54 % and the COP of the cooling system was in the range of 0.6 at the same time.

POTENTIAL OF NEW SYSTEM COMBINATIONS

Several reasons indicate that it seems to be reasonable to look for other system modifications. On the one hand simplifications in the field of the solar system would lead to cost reductions and on the other hand the integration of conventional chillers into desiccant cooling cycles might be an interesting concept for an energetic improved backup. Furthermore it turned out that the monovalent desiccant cooling cycle as presented in Figure 1 is not able to satisfy the indoor comfort criteria in warm and humid climates. For very high ambient humidities the system cannot provide humidities behind the dehumidifier low enough to achieve a low temperature *and* humidity after the fresh air humidifier.

In the further course of this paper two paths are described in order to evaluate the performance of new system combinations:

- The first path holds for applications with no requirement for fixed indoor conditions In this case the performance of a simple, purely solar (thermal) driven system can be determined by computing the hours where desired indoor temperature and humidity values are not achieved.
- The second path holds for applications with the requirement for fixed indoor conditions. In this case a backup has to be installed and the performance of the solar system can be measured by computing the solar fraction for the required thermal energy to drive the cooling process.

Non-fixed indoor conditions

Quite a simple configuration of a solar driven desiccant cooling system can be realized using solar air collectors. Two ways to integrate the solar air collector are given in Figure 7.

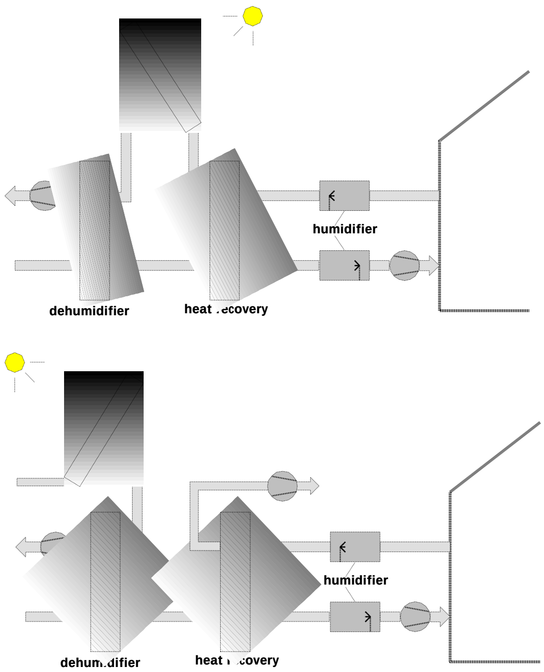

Figure 7: Desiccant cooling system with solar air collector as only heat source. System integrated mode (top) and ambient air mode (bottom)

Both types of systems require no buffer storage and no auxiliary heater for cooling. Which of the systems should be installed depends on the ambient air conditions. The variant with ambient air works better in case of low ambient air humidities; however, another ventilator is required in this case.

During a broad parametric study, summarized in C. Hindenburg 1998, it was shown, that systems like those given in Figure 7 can be realized under temperate climatic conditions such as given in central Europe. Figure 8 shows ambient air conditions for the summer period (May - October) for Freiburg/Germany and the respective indoor temperature and humidity values in a lecture room which is air conditioned with a solar driven desiccant cooling system. For the simulations the seminar room from TGZ Riesa was used and a market available solar air collector of 23 m^2 was assumed for heat supply. Figure 8 indicates that the temperature remains in the comfort region according to the german standard DIN 1946, 1994 over the whole period; only the indoor humidity exceeds the comfort region at about 20 hours. An analysis of the costs for the solar heat supplied to the desiccant cooling system resulted in values of 0.25 DEM per kWh. Of course, solar energy gained during the heating season can be supplied to the building as well.

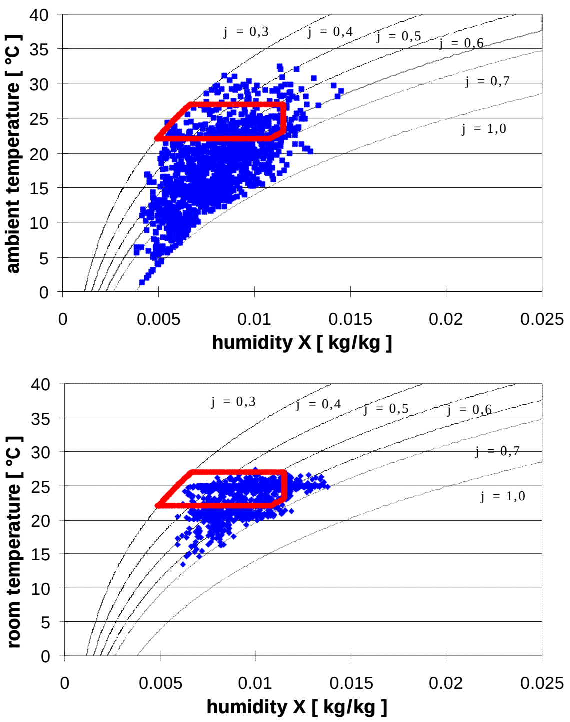

Figure 8: Ambient air conditions (top) and indoor conditions (bottom) of a seminar room air conditioned with a solar driven desiccant cooling system (May-October); the area in the lines characterizes the comfort region according to german standard DIN 1946/II

Fixed indoor conditions

In a second study different system combinations have been investigated for the case that strict indoor conditions are required; details of this study and the assumptions for the economic analysis can be found in Henning 1998. The systems were investigated for 3 different climates, characterized by Test Reference Years (see Table 2). A reference office building (room area 400 m^2) with a high fraction of glazings on the south facing wall has been taken as cooling/heating load; specifications of the building are given in Franzke, 1994.

location	Copenhagen/ Denmark	Freiburg/ Germany	Trapani/ Italy
collector radiation sum	1127 kWh	1196 kWh	1920 kWh
max. temper-ature, °C	29	33	36
max. humidity ratio, g/kg	13	14.5	24

Table 2: Climatic conditions of investigated sites

The following systems were compared with regard to their primary energy consumption and their costs:

- Reference system with electrically driven vapour compression chiller (figure 9, top)
- Desiccant cooling system without conventional chiller (figure 9, center) with flat plate collectors or solar air collectors and buffer storage sizes of 0, 5 and 10 m^3; this system type has been investigated for the climates of Copenhagen and Freiburg
- Desiccant cooling system with integrated conventional chiller (figure 9, bottom) with flat plate collectors or solar air collectors and buffer storage sizes of 0, 5 and 10 m^3; this system type has been investigated for the climate of Trapani, since the monovalent desiccant cooling cycle does not suffice under that conditions.

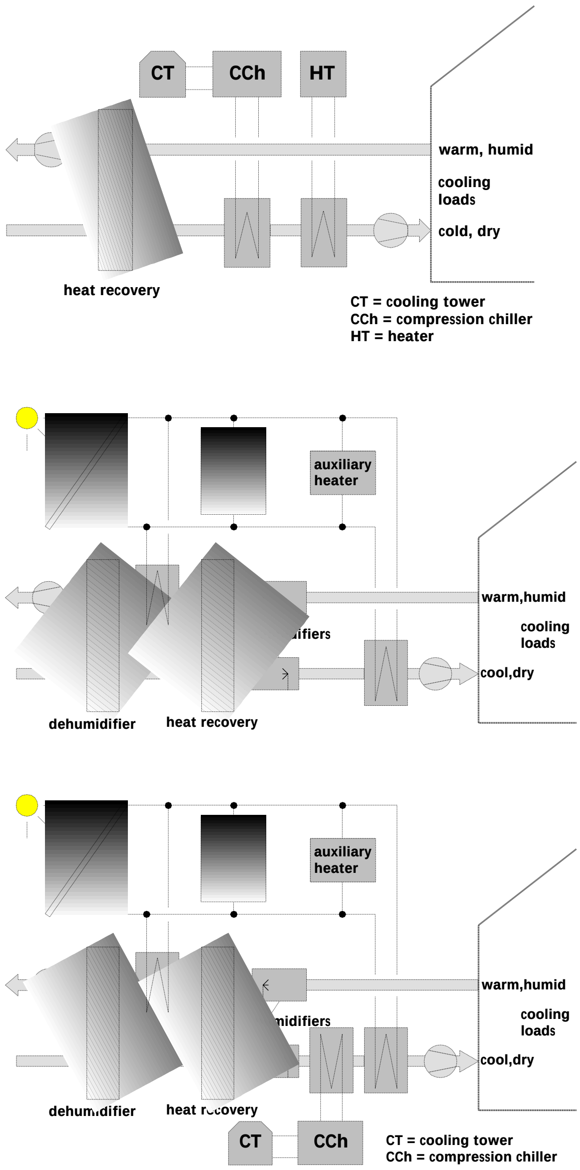

Figure 9: Compared systems; reference system (top), solar assisted desiccant cooling system without backup chiller (center) and solar assisted desiccant cooling system with backup chiller (bottom)

In order to compare the different systems with regard to energy and economy the following performance criteria have been defined:

- specific annual overall costs for building air conditioning, i.e. the amount of money that has to be spent for a comfortable indoor climate over the whole year (heating and cooling): US$ per m^2 and year

(including investitions and planing, capital costs, operation and maintenance costs, energy costs)

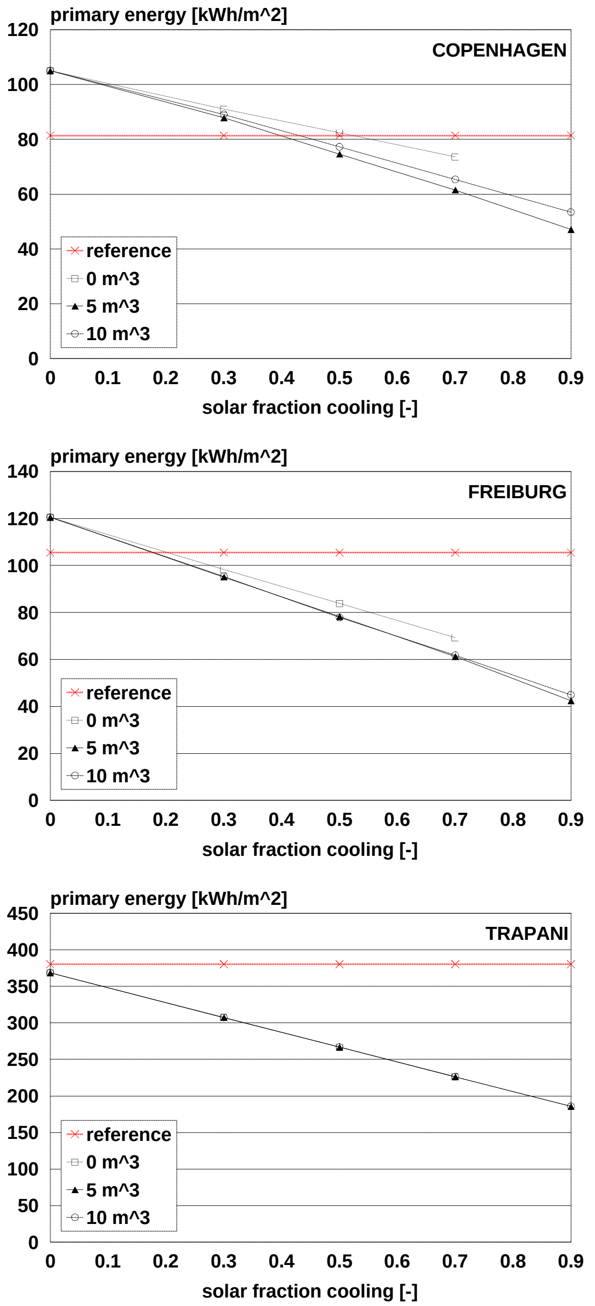

Figure 10: Annual specific primary energy consumption for air conditioning and heating as function of the solar fraction for cooling for the locations Copenhagen (top), Freiburg (center) and Trapani (bottom). Buffer storage sizes of 0, 5 and 10 m^3. Results apply for flat plate collecotrs; however, results for solar air collectors are very similar.

- annual amount of primary energy required for this purpose: kWh of primary energy per m^2 and year
- cost of saved primary energy in US$ per kWh defined as:

$$\frac{\text{(cost of solar system} - \text{cost of conventional system)}}{\text{(primary energy conv. system} - \text{primary energy solar system)}}$$

In Figure 10 the specific primary energy demand is shown for the locations Copenhagen, Freiburg and Trapani. It turns out that in Copenhagen and - less pronounced - in Freiburg ‘switching’ from conventional (electrically driven) cooling to desiccant (thermal driven) cooling results in an increase of the primary energy demand. Primary energy savings can only be achieved if a certain fraction of the required heat to drive the cooling process is yielded from the solar collector. The value of the solar fraction at the break-even point is in the range of 40 % in Copenhagen and in the range of 18 % in Freiburg; the exact value depends on the applied solar collector (water or air cooled) and the buffer storage size. However, in Trapani a primary energy saving is already achieved in case of a completely gas fired desiccant cooling system (solar fraction for cooling = 0).

Primary energy saving is dependent from the location, i.e. the climatic conditions, since sorptive dehumidification is more favourable compared to conventional de-humidification (condensation). The higher the humidity of ambient air the higher is the fraction of energy required for dehumidification purposes. Therefore, it turned out as a first result of this study that a combination of (solar driven) sorptive dehumidification with conventional (vapour compression chiller driven) temperature reduction is a promising combination for warm and humid climates.

A closer look to the system operation in Trapani is given in Table 3, listing the numbers of hours with the different operation modes.

cases with cooling	2427 hours
cases with adiabatic cooling	954 hours
cases with desiccant cooling	1473 hours
desiccant cooling without backup chiller	686 hours
desiccant with backup chiller for temperature decrease	787 hours
desiccant cooling with backup chiller for dehumidification and temperature decrease	519 hours

Table 3: Operation conditions for the combined system (Figure 9, bottom) in Trapani

In Figure 11 the specific annual overall costs for air conditioning and heating are shown. Following conclusions can be drawn form the figures:

- Costs are a bit lower for flat plate collectors with liquid heat carrier than for solar air collectors; the main reason is that at tempertaures required for air conditioning the flat plate collectors exhibit higher efficiencies.

In general the installation of a buffer storage shows benefits with relation to costs. This benefit is more pronounced in Copenhagen and Freiburg than in Trapani, i.e. in climates in which heating is similar or more important than cooling. In these climates high solar fractions (> 70 %) can not be achieved without a buffer storage without oversizing the collector field distinctly.

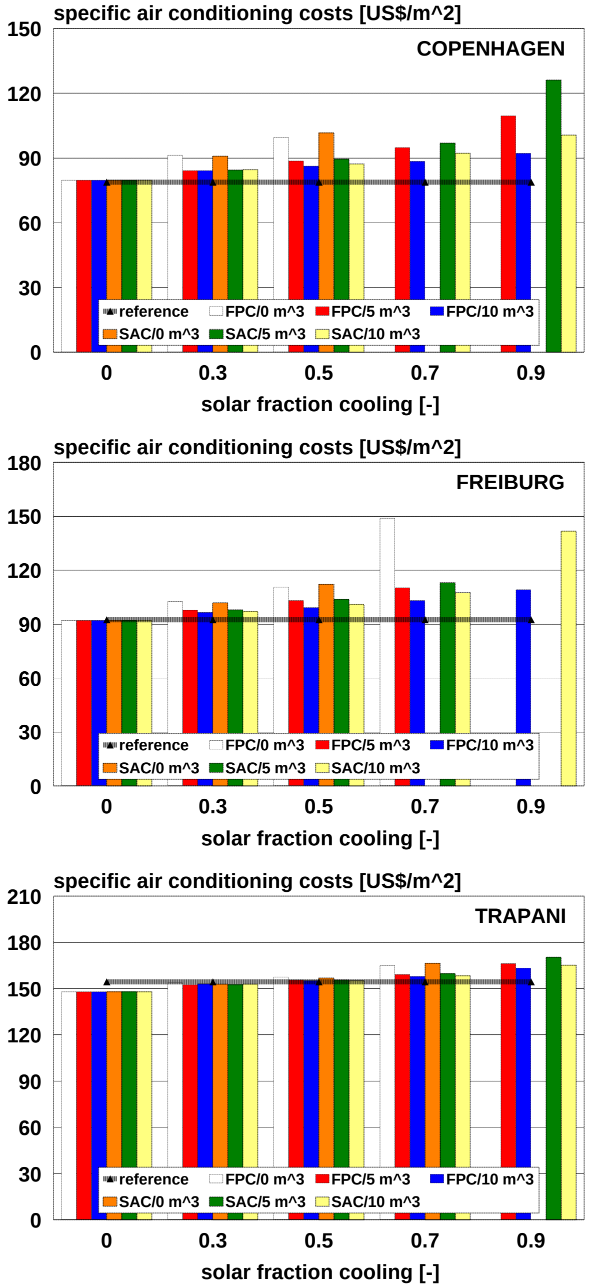

Figure 11: Annual specific costs for air conditioning and heating as function of the solar fraction for cooling for the locations Copenhagen (top), Freiburg (center) and Trapani (bottom). FPC = flat plate collector (water cooled), SAC = solar air collector, 0 m^3 = no buffer storage, 5 m^3 = buffer storage with 5 m^3, 10 m^3 = buffer storage with 10 m^3

Similar results are shown in Figure 12, which gives a measure for the 'price' of primary energy saving. With the technology investigated here saving of primary energy 'costs' ≥0.5 US$/kWh in Copenhagen and ≥0.25 US$/kWh in Freiburg. In Trapani cost benefits can be achieved at low values of the solar fraction (30 %). Furthermore solar assisted cooling with higher values of the solar fraction enables primary energy savings at very low extra costs. Negative values in the case of Trapani mean that primary energy savings are accomponied by decreased cost values. However, it has to be mentioned that economic results are valid for german energy price structures. Therefore, results concerning costs may be different in other countries.

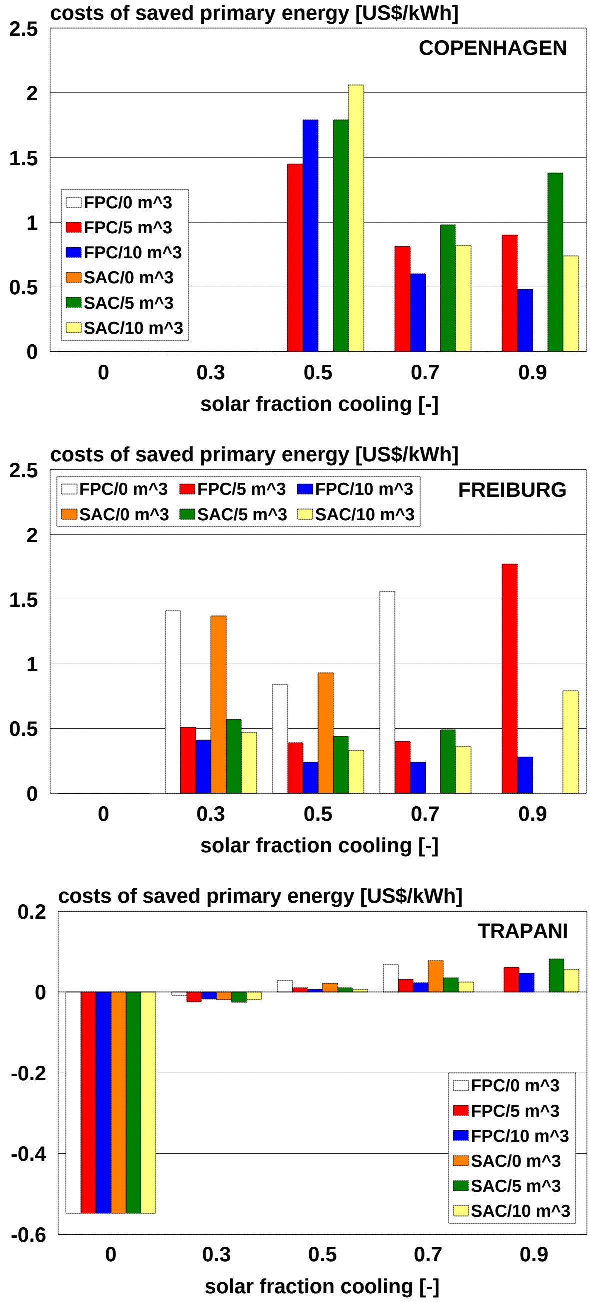

Figure 12: Overall costs for saved primary energy as function of the solar fraction for cooling for the locations Copenhagen (top), Freiburg (center) and Trapani (bottom). FPC = flat plate collector (water cooled), SAC = solar air collector, 0 m^3 = no buffer storage, 5 m^3 = buffer storage with 5 m^3, 10 m^3 = buffer storage with 10 m^3

Again overall results look most promising for the case of Trapani. Cost values increase very low up to solar fractions of about 70 % and more. Therefore, conditions for the integration of solar energy into combined desiccant/conventional cooling cycles look promising in this case from an economical point of view, too.

SUMMARY

In the paper several aspects of solar assisted desiccant cooling have been discussed. Experiences with a pilot plant indicate that the technology is market available and works well; expected solar fractions for cooling could be nearly realized. The performance of the desiccant cooling system does not completely fulfill expectations, since the measured overall COP is a bit lower than expected.

Simple system configurations without buffer storage and auxiliary heater and with a solar air collector as only heat source for the cooling cycle are a promising concept in cases, where the users accept that the indoor climate does not fulfill standard comfort criteria for few hours during the cooling season. However, this type of system should be operated under temperate climatic conditions; in particular, high values of ambient air humidity require backup systems.

Finally a parametric study showed that combined desiccant/chiller solar assisted cooling systems are feasible from an energetic as well as from an economic point of view, especially in warm-humid climtes. This is even valid, if the indoor standard comfort criteria have to be kept at every hour of the year. Combinations of sorptive dehumidification with a conventional, electricially driven backup system allow for primary energy savings up to 50 % at low increased overall costs. This track seems to be worthwile for future research and demonstration projects.

NOMENCLATURE

$P_{load,sensible}$	sensible cooling load, kW
$\dot{m}_{air}$	air mass flow, kg s^{-1}
c_{air}	specific heat capacity of air (incl. water vapour), J kg^{-1} K^{-1}
T_{out}	building outlet temperature, °C
T_{in}	building inlet temperature, °C

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REFERENCES

DIN 1946 (1994) Deutsches Institut für Normung, DIN 1946 Teil 2, Raumlufttechnik, Gesundheitstechnische Anforderungen (VDI Lüftungsregeln)

Uwe Franzke, C. Seifert (1995) Stille Kühlung. Bericht zum BMFT-Forschungsvorhaben ILK-B-4/95-2469

Hans-Martin Henning, Thomas Erpenbeck, Carsten Hindenburg, Sören Paulußen (1998) Solar Cooling of Buildings - possible techniques, potential and international development. EuroSun '98, Portoroz, Slovenia

Carsten Hindenburg (1998) Untersuchung des Einsatzes von Solarluftkollektoren in sorptionsgestützten Klimatisierungssystemen auf der Basis von Systemsimulationen in TRNSYS. Diplomarbeit Technische Universität Hamburg-Harburg

Carsten Hindenburg 1998 Einsatz von Solarluft-kollektoren in sorptionsgestützten Klimatisierungs-systemen. Achtes Symposium Thermische Solarenergie, Staffelstein 1998, Germany

George O.G. Lof (1992) Desiccant Systems. in: Solar Air Conditioning and Refrigeration (edited by A.A.M. Sayigh and J.C. McVeigh), Pergamon Press, Oxford, England

Harry I. Robison (1992) Desiccant Cooling. in: Solar Air Conditioning and Refrigeration (edited by A.A.M. Sayigh and J.C. McVeigh), Pergamon Press, Oxford, England

TRNSYS (1994) TRNSYS - A Transient System Simulation Program (Version 14.2), Madison/Wisconsin, USA