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Micro tri-generation system for indoor air conditioning in the Mediterranean climate

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Abstract

Mediterranean countries show two specific features regarding air-conditioning of buildings: a high—and growing—cooling load and high relative humidity, at least in coastal zones. In this contribution we report on the development of an innovative micro scale tri-generation system (power + heating + cooling), equipped with a rotor based desiccant system adapted to the Mediterranean conditions which receives heat for the desiccant regeneration from a combined heat and power (CHP) cycle.

The paper presents the design of the advanced desiccant air handling unit which uses a high efficient combination of a vapor compression chiller working at a high evaporator temperature and a desiccant wheel (silica gel). The electricity of the chiller is supplied by the CHP system and the heat to regenerate the desiccant is the waste heat of the CHP. System simulations have been used to optimize the hydraulic design and the operation strategy in order to minimize operation costs and maximize energy savings. Some new component models, e.g. for the advanced desiccant cycle were developed for this purpose. The final design of the entire system consisting of the CHP system, the vapor compression chiller, the advanced desiccant air handling unit and the load system is described. The load system is composed of an air duct network with induction units and a chilled water network with fan-coils in the office rooms.

Regarding energy performance results indicate an electricity saving >30% in comparison to state-of-the-art solutions based on conventional technology.

Keywords: Tri-generation; Desiccant cooling; Humid climates; Dehumidification

1. Introduction

Air-conditioning of buildings is a promising application of co-generation systems during summer. For this purpose thermally driven equipment to supply cooling driven with the co-generator waste heat has to be employed. The most widespread technology used for this purpose is based on heat driven water chillers such as absorption chillers (e.g. with the material pair lithium-bromide–water) or adsorption chillers. Water chillers are used in combination with different techniques to purge cooling loads from the rooms such as e.g. fan-coil systems, chilled ceilings or centralized air handling units (AHU). However, in order to treat latent loads, air has to be cooled below the dew-point when chilled water systems are used. Thereby the air is cooled far below the temperature level needed for comfortable indoor conditions and consequently the chiller works at a COP lower than if employed for sensible cooling, i.e. temperature control, only.

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An alternative to treat latent loads by cooling air below the dew-point is the direct treatment of ventilation air in an open sorptive cooling cycle, also referred to as desiccant cooling system. In such a cycle air dehumidification is realized using a sorptive component such as a sorptive wheel. Additionally, a temperature decrease can be achieved by combination of the sorptive dehumidification with either direct, indirect or combined (direct + indirect) evaporative cooling.

However, the standard desiccant cooling cycle, which is for instance installed in temperate climates like Central Europe, is not able to cope with the conditions of warm and humid climates such as for instance in the coastal zones of the Mediterranean countries. Therefore application of desiccant technology in such climates using sorptive rotors requires specific configurations. In the framework of the project MITES (''Micro Trigeneration System for Indoor air conditioning in the Mediterranean Climate''), a project supported by the European Union, a novel configuration of an open cooling cycle based on sorptive rotor technology has been developed. This heat driven air handling unit receives its driving heat from a motor co-generation unit and is specially designed for weather conditions with high humidity ratios of the ambient air. A pilot system has been installed in fall 2003 at the client building of the gas utility of the municipality of Palermo (AMG) in Sicily/Italy and the system is commissioned and operated with accompanying monitoring during 2004.

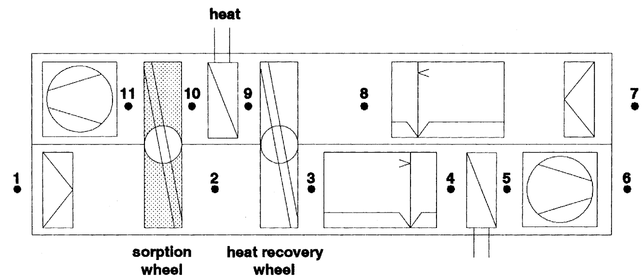

Fig. 1. Standard desiccant cooling system.

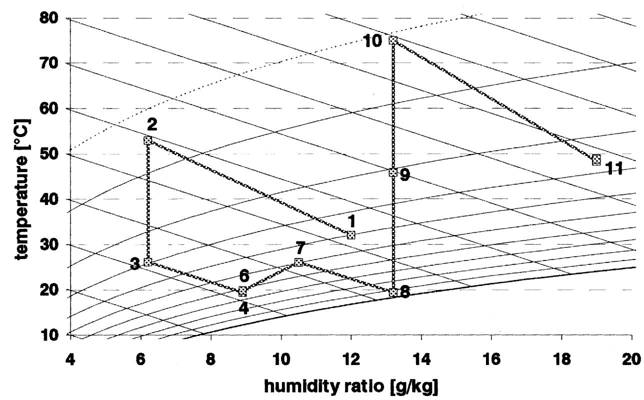

Fig. 2. Standard desiccant cooling cycle in the T–x-diagram of humid air.

2. Desiccant air handling unit configuration

As a first step of the project different configurations of desiccant air handling units were compared in order to identify the configuration which is able to provide desired supply air conditions with a minimum of energy consumption. However, before the different new designs are presented and compared, the standard desiccant cycle as used in temperate climates is described in detail in order to show the general operation principle. Based on this cycle different modifications were made in order to adjust it to the specific needs in a warm-humid climate.

2.1. Standard desiccant cooling cycle

The standard cycle which is mostly applied today uses rotating desiccant wheels, equipped either with silica gel or lithium-chloride as sorption material. All required components are standard components and have been in use for air-conditioning of buildings or factories since many years.

The standard cycle using a desiccant wheel is shown in Fig. 1 and the corresponding states of the air in the cycle are shown in Fig. 2. Systems according to this scheme are typically employed in temperate climates; the example of Fig. 1 is based on typical design conditions in Central Europe (e.g. Germany). The air follows the following processes during the system:

- $1 \rightarrow 2$ sorptive dehumidification of supply air; the process is almost adiabatic and the air is heated by the adsorption heat and the hot matrix of the wheel coming from the regeneration side;
- $2 \rightarrow 3$ pre-cooling of the supply air in counter-flow to the return air from the building;
- $3 \rightarrow 4$ evaporative cooling of the supply air to the desired supply air humidity by means of a humidifier;
- $4 \rightarrow 5$ the heating coil is used only in the heating season for pre-heating of air;
- $5 \rightarrow 6$ a small temperature increase is caused by the fan;
- $6 \rightarrow 7$ supply air temperature and humidity are increased by means of internal loads;
- $7 \rightarrow 8$ return air from the building is cooled using evaporative cooling close to the saturation line;
- $8 \rightarrow 9$ the return air is pre-heated in counter-flow to the supply air by means of a high efficient air-to-air heat exchanger, e.g. a heat recover wheel;

$9 \rightarrow 10$ regeneration heat is provided for instance by means of a co-generation system;
$10 \rightarrow 11$ the water bound in the pores of the desiccant material of the dehumidifer wheel is desorbed by the hot air;
$11 \rightarrow 12$ exhaust air is blown to the environment by means of the return air fan.

Application of the cycle described above is limited to temperate climates. Reason is, that the achievable supply air dehumidification is not high enough to enable direct evaporative cooling at conditions with far higher values of the humidity of ambient air.

3. Cycles adjusted to humid climates

For all studied cycles the same boundary conditions, i.e., temperature and humidity values of ambient air, supply air to the building, return air from the building and regeneration air to regenerate the sorption material were assumed. These values are shown in Table 1. The following modified cycles which all use cooling coils in addition to the sorptive wheel were studied regarding their energy performance:

- Standard cycle with a cooling coil added behind the heat recovery wheel on the supply air side; a scheme is shown in Fig. 3 and the corresponding air states in Fig. 4. The sorptive wheel realizes a pre-dehumidification (air states $1 \rightarrow 2$) and the cooling coil controls the air to achieve the final desired humidity (air states $3 \rightarrow 4$). A re-heater (air states $4 \rightarrow 5$) is needed, if the supply temperature shall enter the room with a comfortable temperature, i.e., a temperature not below 18 °C.
- Cycle using two sorptive wheels which are operated in series with an intermediate cooling coil (air states $2 \rightarrow 3$); a scheme is shown in Fig. 5 and the corresponding air states in Fig. 6. Using this system the complete dehumidification of ambient air is realized by sorption. A second cooling coil (air states $5 \rightarrow 6$) is necessary in order to achieve the desired supply

Table 1
Boundary conditions for cycle design

Parameter	Unit	Value
Ambient air temperature	°C	35.0
Ambient air humidity ratio	g/kg	25.0
Supply air temperature (to room)	°C	18.0
Supply air humidity ratio	g/kg	9.0
Return air temperature (from room)	°C	26.0
Return air humidity ratio	g/kg	11.5
Hot water temperature (from COG)	°C	85.0

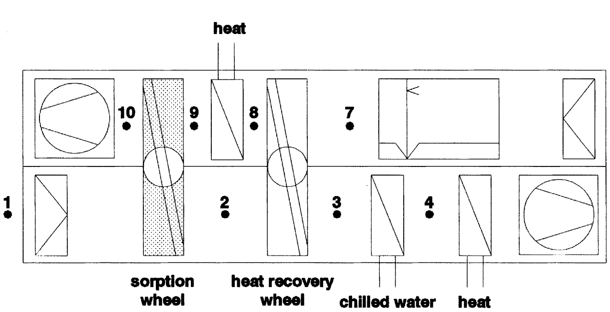

Fig. 3. Standard cycle with additional cooling-coil behind heat recovery wheel.

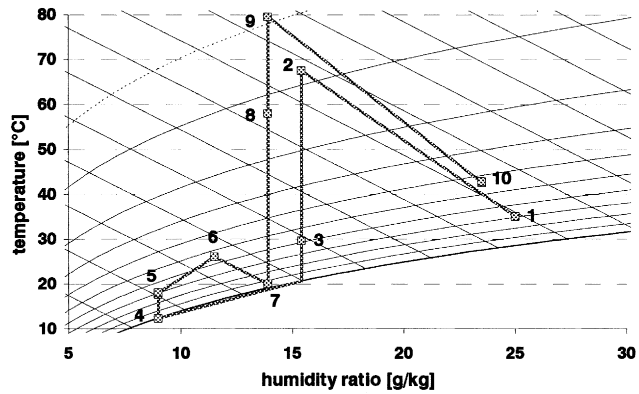

Fig. 4. Cycle of Fig. 3 in the T–x-diagram of humid air.

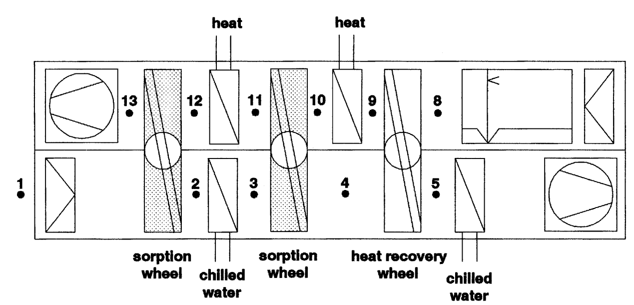

Fig. 5. Cycle with two sorption wheels.

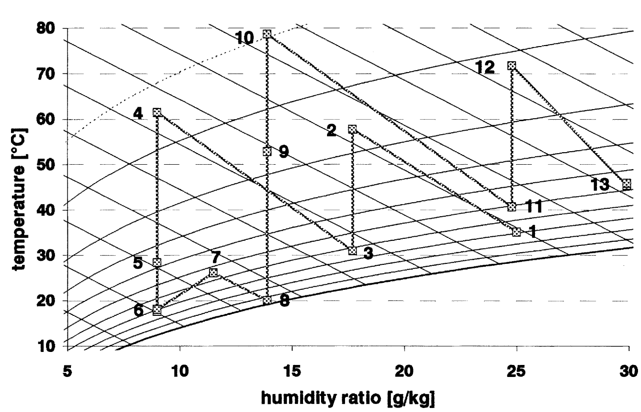

Fig. 6. Cycle of Fig. 5 in the T–x-diagram of humid air.

air temperature. Two heating coils are necessary in order to provide regeneration heat for the first sorptive wheel (air states 9 → 10) and the second sorptive wheel (air states 11 → 12). Since no dehumidification is realized by cooling air below the dew-point, the required cold water temperature is relatively high.

- Cycle employing two cooling coils, a first one in front of the sorptive wheel for pre-dehumidification (air states 1 → 2) and a second one for control of supply air temperature (air states 4 → 5); a scheme is shown in Fig. 7 and the corresponding air states in Fig. 8. Although pre-dehumidification is realized by cooling the air below the dew-point, a high value of chilled water temperature is sufficient since the dehumidification takes place at a high value of the humidity ratio and thus at a high saturation temperature of water vapor.
- At last a conventional system has been modeled in order to compare the sorptive cycles with a reference; a scheme is shown in Fig. 9 and the corresponding air states in Fig. 10. This system consists of a conventional air handling unit in which evaporative cooling of the return air is used to pre-cool the ambient air with a heat recovery system (air states 1 → 2). A cooling coil guarantees that the desired level of dehumidification is achieved (air states 2 → 3). A re-heater (air states 3 → 4) is needed, if the supply temperature shall enter the room with a comfortable temperature, i.e., a temperature not below 18 °C.

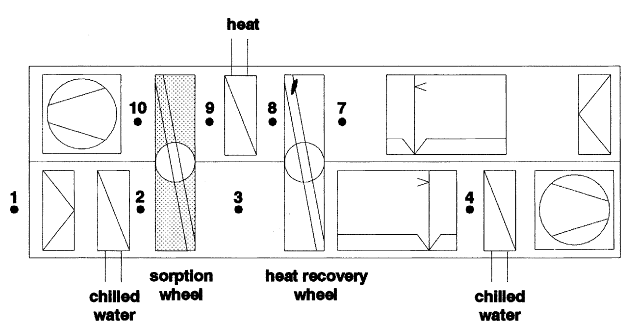

Fig. 7. Cycle with a cooling coil before the sorption wheel.

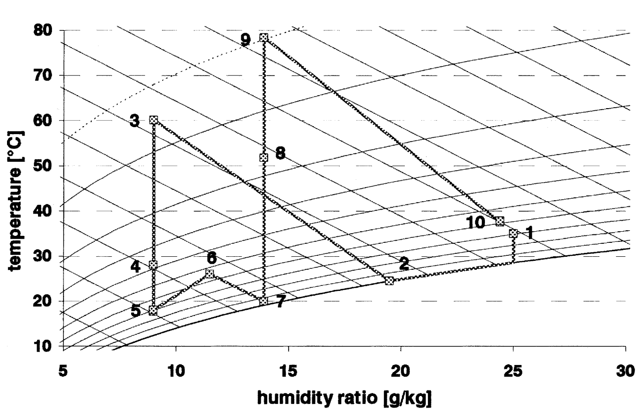

Fig. 8. Cycle of Fig. 7 in the T–x-diagram of humid air.

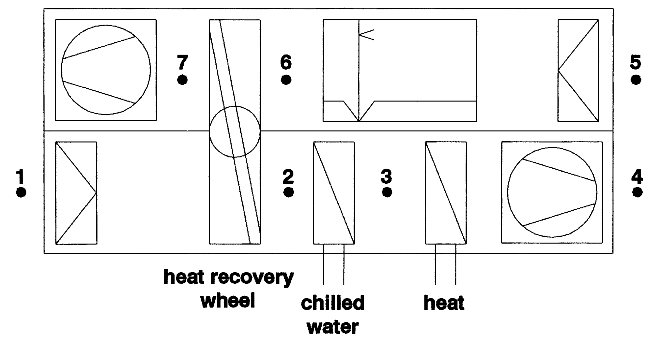

Fig. 9. Conventional reference cycle without sorption wheel.

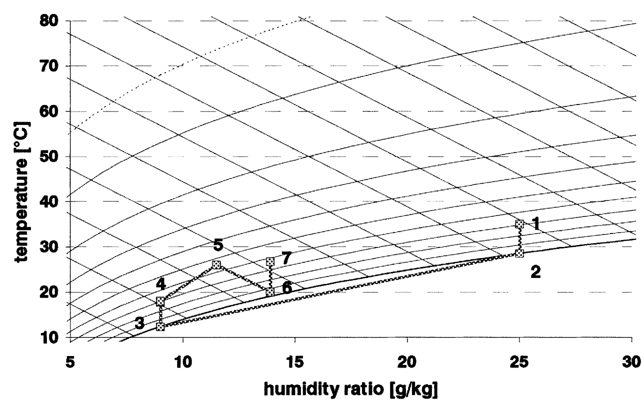

Fig. 10. Cycle of Fig. 9 in the T–x-diagram of humid air.

The calculation of the air states has been carried out using a design tool developed at the Fraunhofer Institute for Solar Energy Systems in which many different system configurations can be studied. The computer tool contains an overall of 21 components which can be switched on or off in order to derive a new configuration out of the complete system. Standard performance figures for all components were used; the used desiccant wheel model has been developed by Motta [1] and is described in Motta et al. [2]. A by-pass fraction of 20% was used for sorptive wheel regeneration, i.e., only 80% of the return air have to be heated up to the regeneration temperature and pass the sorptive wheel.

In order to compare the performance of the different cycles the following performance figures have been defined:

- The total cooling, $P_{\text{cooling,tot}}$ is defined as the enthalpy difference between ambient air and supply air multiplied with the air mass flow:

$$P_{\text{cool,tot}} = \dot{m}_{\text{air}} \cdot (h_{\text{ambient}} - h_{\text{supply}})$$

- The conventional cooling, P_{conv} denotes the the cooling supplied by the cooling coils, for instance using chilled water from a compression chiller.
- The chiller COP, $COP_{chiller}$, denotes the COP of a conventional vapor compression chiller and depends on the difference between the temperature of chilled water, $T_{chilled\,water}$ and the temperature of ambient air, which defines the condensation condition of the chiller; to calculate the $COP_{chiller}$ the performance of a typical market available compression chiller employing FKW 134a as refrigerant has been used.
- The sorptive cooling, $P_{cool,sorpt,}$ defines the amount of the total cooling which is not covered by the cooling coils:

$$P_{cool,sorpt} = P_{cool,tot} - P_{conv}$$

- The sorptive COP, COP_{sorpt}, is defined as the fraction between the sorptive cooling and the required heat for regeneration of the desiccant, P_{reg}:

$$COP_{sorpt} = \frac{P_{cool,sorpt}}{P_{reg}}$$

- $P_{el,vent}$ defines the electricity demand of the ventilators. This electricity demand depends on the overall pressure drop of the air handling units which is quite different for the different designs due to the implemented components.
- $P_{el,chiller}$ is the electricity demand of the chiller which is defined by the fraction between the conventional cooling and the chiller COP:

$$P_{el,chiller} = \frac{P_{conv}}{COP_{chiller}}$$

- The total electricity demand, $P_{el,tot}$, is the sum of the electric consumption of the chiller and the ventilators:

$$P_{el,tot} = P_{el,chiller} + P_{el,vent}$$

Results of the comparison of all the performance figures defined above for the studied systems are summarized in Table 2. All given values refer to a nominal air flow of 1000 m^3/h at return air conditions (26 °C). The following conclusions can be drawn from this comparison:

- The standard cycle with additional cooling coil behind the heat recovery wheel (scheme of Fig. 3) requires the lowest amount of conventional cooling. However, this cooling is needed at a low temperature level, since the final humidity control of the supply air is realized with this cooling coil.
- Both, the 2-wheels cycle (Fig. 5) as well as the 2-cooling-coils cycle (Fig. 7) need chilled water at a far higher temperature. This means that eventually other environmental heat sinks such as well water might be used if available.
- The 2-wheels cycle requires a far higher amount of regeneration heat than both other sorptive cycles and the highest electricity demand for the ventilators.
- The lowest overall electricity demand is shown by the 2-cooling coils cycle and a reduction of electric consumption of about 34% in comparison to a conventional system (Fig. 9) is achieved.

Finally, for the installation at the client building of the gas utility of the municipality of Palermo (AMG) the 2-cooling-coils configuration was selected, although this configuration shows a higher regeneration heat demand compared to the standard configuration. However, since the amount of waste heat from the co-generation system is far higher than the heat needed by the desiccant system, this is no limiting condition.

4. Design of the complete system

After the design of the desiccant air handling unit the design of the complete system consisting of the co-generation unit, the air handling unit, the compression chiller and the building related components has been made, Based on a detailed cooling load calculation it became obvious that not all cooling loads can be covered by

Table 2
Comparative results of the studied system configurations

Parameter	Unit	Standard	2 Wheels	2 Cooling coils	Reference
Scheme shown in	–	Fig. 3	Fig. 5	Fig. 7	Fig. 9
Total cooling, $P_{cool,tot}$	kW	19.1	19.1	19.1	19.1
Conventional cooling, P_{conv}	kW	11.2	12.7	11.7	18.8
$T_{chilled\,water}$	°C	8	14.9	15.1	8.3
Chiller COP, $COP_{chiller}$	–	3.99	4.35	4.36	4.00
Sorptive cooling, $P_{cool,sorpt}$	kW	7.9	6.4	7.4	–
Regeneration heat, P_{reg}	kW	5.9	15.5	7.2	–
Sorptive COP, COP_{sorpt}	–	1.34	0.41	1.03	–
Electricity demand ventilators, $P_{el,vent}$	kW	0.6	0.8	0.6	0.3
Electricity demand chiller, $P_{el,chiller}$	kW	2.81	2.92	2.68	4.70
Total electricity demand, $P_{el,tot}$	kW	3.41	3.72	3.28	5.00
Electricity saving	%	31.8	25.6	34.3	–

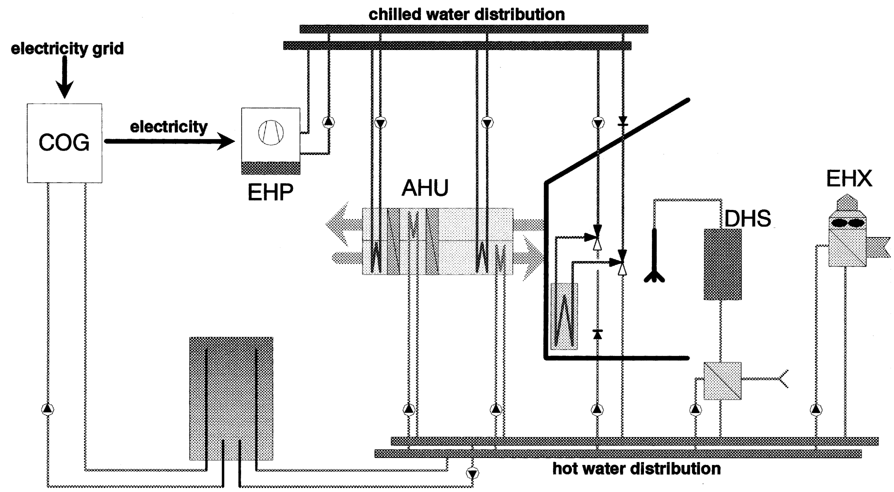

Fig. 11. Scheme of the complete system at the client building of the gas utility of the municipality of Palermo (AMG). The system is used for field testing and supplies the second floor of the building. Meaning of abbreviations: COG = co-generation plant; EHP = electrical heat pump (water chiller, air cooled); AHU = air handling unit with sorptive wheel according to the scheme in Fig. 7; DHS = domestic hot water storage; EHX = heat exchanger for rejection of excess heat.

the air handing unit. Therefore a fan coil system is operated in addition which is supplied with chilled water from the compression chiller as well. A scheme of the overall system is show in Fig. 11. Fig. 12 presents a photograph of the system showing the air handling unit mounted outdoors and the hydraulic pipe network.

During summer the co-generation system provides electricity for the compression heat pump (indicated EHP in Fig. 11) and for the other appliances (e.g. computers, artificial lighting, printers etc.) of the office rooms. The waste heat of the co-generation unit is used to heat the regeneration air of the desiccant system. Excess heat can be used for domestic hot water preparation (DHS) or is rejected to the environment by the excess water-to-air heat exchanger (EHX). An energy balance of the whole system for the design case is shown in Table 3.

It becomes clear that only a small part of the waste heat of the co-generation unit can be used for air-conditioning purposes. However, it has to be noted that this balance is valid for the design case with a very high cooling demand. At most time the cooling demand is far lower and consequently the fraction of useful exploitation of waste heat is higher.

During winter waste heat is directly used for heating using the fan-coils as well as the air handling unit.

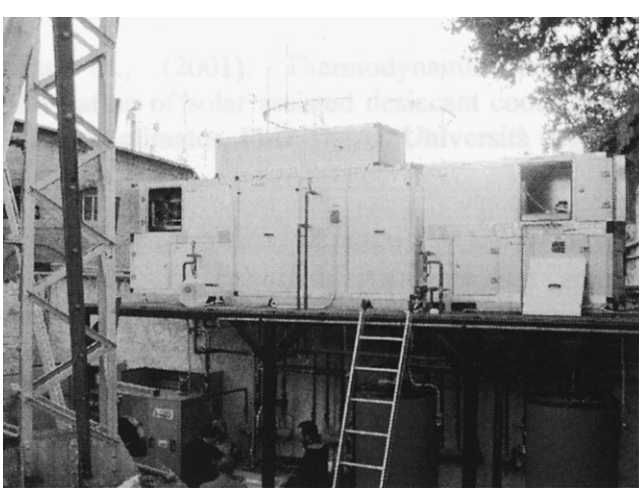

Fig. 12. Photo of the installed system in Palermo (Air handling unit).

Table 3
Energy balance of the complete system—design case

Parameter	Unit	Value
Volume flow fresh air	m^3/h	1100
Cooling power for AHU	kW	12.8
Cooling for fan-coils	kW	12.0
EHP electric demand	kW	5.70
Ventilators electric demand	kW	0.66
Appliances electric demand	kW	3.00
Total electric demand	kW	9.36
COG waste heat	kW	21.85
Regeneration heat	kW	7.92
Excess heat (+DHS)	kW	13.93

5. System simulation

In order to be able to calculate an energy balance of the entire system for a whole year a simulation model of the complete system has been set up. Goal of the simulation tool is also to investigate different control strategies and to develop a tool for design of similar systems in the future. For this purpose the simulation platform TRNSYS has been used (see TRNSYS [3]). Specific sub-routines for modeling of the co-generation unit and the desiccant air handling unit with two cooling coils have been developed. A 2-step approach has been

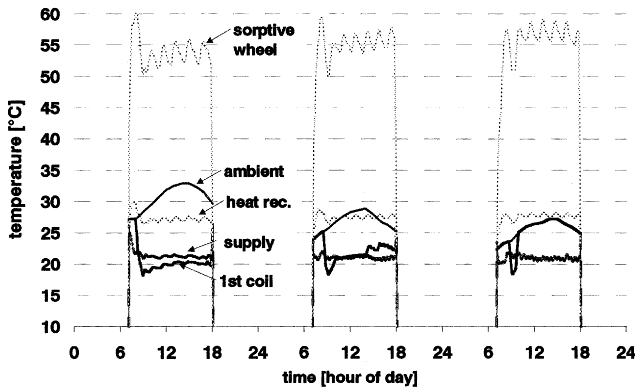

Fig. 13. Temperature profiles at different positions of the AHU for three selected days.

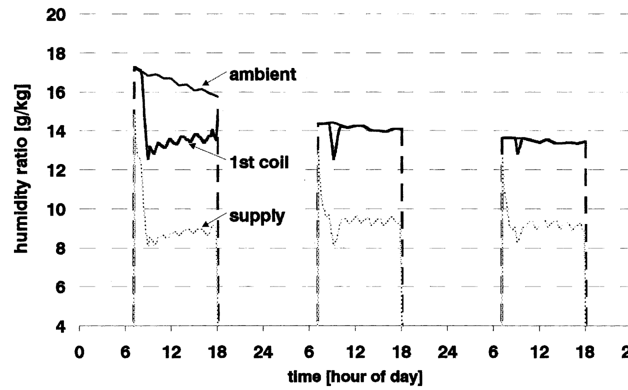

Fig. 14. Profiles of the humidity ratio at different positions of the AHU.

used: in a first step a building simulation of the second floor of the AMG- building in Palermo has been performed. Thereby an annual load file containing hourly, values of cooling/heating loads and ambient air temperature and humidity values was produced. Based on this load file a simulation of the technical system as shown in Fig. 11 has been carried out.

For this purpose the following control scheme has been used (scheme according to Fig. 7):

- During use of the building (workdays from 8 a.m. to 6 p.m.) the air handling unit is operated with a constant volume flow (fresh air) of 1100 m^3/h. This fresh air demand results from the occupation in the building.
- With increasing cooling loads priority is given to the thermally driven cooling, i.e., as long as the humidity of ambient air allows use of evaporative cooling in the supply air, the air handling unit is operated according to the cycle shown in Fig. 2.
- As soon as the humidity ratio of the supply air exceeds the desired level, the evaporative cooler in the supply air is switched off and the cooling-coil before the sorptive wheel is switched on. The mass flow through this coil is adjusted in order to control the supply air humidity ratio. The second cooling coil is also switched on and used to control the supply air temperature.
- The fan-coil system is operated in order to purge excess sensible loads which are not covered by the air handling unit.
- During winter the heat recovery unit is employed to pre-heat the fresh air. The air is further heated with waste heat from the co-generation system using the second coil.

In Fig. 13 the temperatures at different positions of the desiccant air handling unit are shown for a sequence of three subsequent days in September. Fig. 14 shows the corresponding time profiles of the humidity ratio. It can be clearly seen that on the first day the cooling coil for pre-dehumidification is operated all the time while on the other days the humidity of ambient air is low enough at most times that the sorptive wheel achieves a sufficiently low value of supply air humidity.

6. Conclusions

Different design options of air handling units using sorptive wheels for climates with a high level of environmental air humidity are presented and compared. Based on the comparison of the configurations at design conditions, a system using a combination of sorptive wheels and cooling coils was selected. An electricity saving of more than 30% compared to a conventional air handling unit is expected by using waste heat of a co-generation unit. A pilot system has been installed at the building of the gas utility (AMG) in Palermo/Sicily. The system will be monitored during the first half year of 2004.

Acknowledgements

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