Final report

Collection of Good Practices for DEC design and installation

Date: November, 2015

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IEA Solar Heating and Cooling Program

The Solar Heating and Cooling Programme was founded in 1977 as one of the first multilateral technology initiatives ("Implementing Agreements") of the International Energy Agency. Its mission is *"to enhance collective knowledge and application of solar heating and cooling through international collaboration to reach the goal set in the vision of solar thermal energy meeting 50% of low temperature heating and cooling demand by 2050.*

The member countries of the Programme collaborate on projects (referred to as "Tasks") in the field of research, development, demonstration (RD&D), and test methods for solar thermal energy and solar buildings.

A total of 53 such projects have been initiated to-date, 39 of which have been completed. Research topics include:

- Solar Space Heating and Water Heating (Tasks 14, 19, 26, 44)
- Solar Cooling (Tasks 25, 38, 48, 53)
- Solar Heat for Industrial or Agricultural Processes (Tasks 29, 33, 49)
- Solar District Heating (Tasks 7, 45)
- Solar Buildings/Architecture/Urban Planning (Tasks 8, 11, 12, 13, 20, 22, 23, 28, 37, 40, 41, 47, 51, 52)
- Solar Thermal & PV (Tasks 16, 35)
- Daylighting/Lighting (Tasks 21, 31, 50)
- Materials/Components for Solar Heating and Cooling (Tasks 2, 3, 6, 10, 18, 27, 39)
- Standards, Certification, and Test Methods (Tasks 14, 24, 34, 43)
- Resource Assessment (Tasks 1, 4, 5, 9, 17, 36, 46)
- Storage of Solar Heat (Tasks 7, 32, 42)

In addition to the project work, there are special activities:

- SHC International Conference on Solar Heating and Cooling for Buildings and Industry
- Solar Heat Worldwide – annual statistics publication
- Memorandum of Understanding with solar thermal trade organizations
- Workshops and conferences

Country Members

Australia	Germany	Singapore
Austria	Finland	South Africa
Belgium	France	Spain
China	Italy	Sweden
Canada	Mexico	Switzerland
Denmark	Netherlands	Turkey
European Commission	Norway	United Kingdom
	Portugal	United States

Sponsor Members

European Copper Institute	Gulf Organization for Research and Development
ECREEE	RCREEE

Current Tasks & Working Group:

Task 42	*Compact Thermal Energy Storage*
Task 43	*Solar Rating and Certification Procedures*
Task 45	*Large Systems: Solar Heating/Cooling Systems, Seasonal Storages, Heat Pumps*
Task 46	*Solar Resource Assessment and Forecasting*
Task 47	*Renovation of Non-Residential Buildings Towards Sustainable Standards*
Task 48	*Quality Assurance and Support Measures for Solar Cooling*
Task 49	*Solar Process Heat for Production and Advanced Applications*
Task 50	*Advanced Lighting Solutions for Retrofitting Buildings*
Task 51	*Solar Energy in Urban Planning*
Task 52	*Solar Energy and Energy Economics in Urban Environments*
Task 53	*New Generation Solar Cooling and Heating (PV or Solar Thermally Driven Systems)*
Task 54	*Price Reduction of Solar Thermal Systems*

Completed Tasks:

Task 1	*Investigation of the Performance of Solar Heating and Cooling Systems*
Task 2	*Coordination of Solar Heating and Cooling R&D*
Task 3	*Performance Testing of Solar Collectors*
Task 4	*Development of an Insolation Handbook and Instrument Package*
Task 5	*Use of Existing Meteorological Information for Solar Energy Application*
Task 6	*Performance of Solar Systems Using Evacuated Collectors*
Task 7	*Central Solar Heating Plants with Seasonal Storage*
Task 8	*Passive and Hybrid Solar Low Energy Buildings*
Task 9	*Solar Radiation and Pyranometry Studies*
Task 10	*Solar Materials R&D*
Task 11	*Passive and Hybrid Solar Commercial Buildings*
Task 12	*Building Energy Analysis and Design Tools for Solar Applications*
Task 13	*Advanced Solar Low Energy Buildings*
Task 14	*Advanced Active Solar Energy Systems*
Task 16	*Photovoltaics in Buildings*
Task 17	*Measuring and Modeling Spectral Radiation*
Task 18	*Advanced Glazing and Associated Materials for Solar and Building Applications*
Task 19	*Solar Air Systems*
Task 20	*Solar Energy in Building Renovation*
Task 21	*Daylight in Buildings*
Task 22	*Building Energy Analysis Tools*
Task 23	*Optimization of Solar Energy Use in Large Buildings*
Task 24	*Solar Procurement*
Task 25	*Solar Assisted Air Conditioning of Buildings*
Task 26	*Solar Combisystems*
Task 27	*Performance of Solar Facade Components*
Task 28	*Solar Sustainable Housing*
Task 29	*Solar Crop Drying*
Task 31	*Daylighting Buildings in the 21st Century*
Task 32	*Advanced Storage Concepts for Solar and Low Energy Buildings*
Task 33	*Solar Heat for Industrial Processes*
Task 34	*Testing and Validation of Building Energy Simulation Tools*
Task 35	*PV/Thermal Solar Systems*
Task 36	*Solar Resource Knowledge Management*
Task 37	*Advanced Housing Renovation with Solar & Conservation*
Task 38	*Solar Thermal Cooling and Air Conditioning*
Task 39	*Polymeric Materials for Solar Thermal Applications*
Task 40	*Towards Net Zero Energy Solar Buildings*
Task 41	*Solar Energy and Architecture*
Task 44	*Solar and Heat Pump Systems*

Completed Working Groups:

CSHPSS; ISOLDE; Materials in Solar Thermal Collectors; Evaluation of Task 13 Houses; Daylight Research

Table of Content

1. EXECUTIVE SUMMARY

The work plan of IEA SHC Task48 addresses quality assurance and support measures for ‘Solar Cooling Technology’ with a strong focus on solar heat driven chillers like ab- and adsorption cooling machines. Nevertheless, activity B2 of SHC Task 48 lead by Austrian partners is dedicated to keep an eye on the open cycle principle with respect to new technical research and developments and as well to produce an extensive report on Good Practice examples of existing solar heat driven desiccant evaporative cooling (SDEC) systems. A desiccant evaporative cooling (DEC) system fulfils all tasks of an air-conditioning system: a) temperature and humidity control and b) control of hygienic air quality by supplying fresh air. Generally speaking the DEC technique applies three thermodynamic principles to treat air without using conventional compression chiller technology: a) dehumidification of supply air with the help of sorption material, b) efficient sensible heat recovery and c) cooling of supply and return air by using evaporative cooling effect. The solar heat is introduced in order to discharge the sorption material loaded by water vapor of ambient air. A profound DEC technology introduction is written in the 3rd edition of the ‘Solar Cooling Handbook’[1].

General targets of the B2 activity are:

- To support quality assurance and support measures for SDEC technology
- To give an overview about worldwide installed SDEC systems
- To express newest R&D activities
- To highlight on existing quality labels of SDEC subsystems
- To produce helpful guidance in order to stimulate stakeholder to realize SDEC

This activity aims at producing a comprehensive report on quality assurance and support measures for ‘Solar heat driven desiccant evaporative cooling systems’. This SDEC technology is not the major focus in the SHC Task 48, but a limited number of activities contributors tried to observe and highlight on one hand new R&D results and on the other hand to document on GOOD PRACTICE CASES of already operated and monitored SDEC systems in three different climatic regions. Finally this cooperation in the framework of the International Energy Agency generated a report with the following chapters:

CHAPTER 2 - WORLDWIDE INSTALLED SDEC SYSTEMS,

This chapter document on the latest updated worldwide survey on existing solar air-conditioning DEC systems. According to the market survey conducted already in the

[1] Henning H.-M., Motta M., Mugnier D., Solar Cooling Handbook - A Guide to Solar Assisted Cooling and Dehumidification Processes, AMBRA, 2013, ISBN: 978-3-99043-438-3

previous SHC Task 38[2], solar heat driven DEC systems have a low market share with regard to the other closed cycle systems (absorption and adsorption chillers) driven by solar heat. The survey identified 30 different SDEC systems, where a different concepts and technologies are applied. This chapter allows getting an overview into the variety of worldwide operated SDEC systems.

CHAPTER 3 - NEW TECHNICAL DEVELOPMENTS

Chapter 4 provides documentation on some R&D activities on four solar heat driven air-conditioning concepts and systems. The technical scope of this data collection is not limited to any specific sorption technology, but there is a dominance of open cycle application. There was not a fix structure given thus the authors decided what and how to present their R&D work.

CHAPTER 4 - EXISTING QUALITY LABELS OF DIFFERENT SUBSYSTEMS OF SDEC SYSTEMS

The purpose of this chapter is to describe the existing quality labels, standards and certifications to define the performance specifications of desiccant wheel and DEC system. This chapter is divided in 4 parts: a) Existing certification for regenerative heat ex-changers; b) Standards for active dehumidification wheels; c) Manufacturers' technical data for desiccant wheel and d) Desiccant-based dehumidification equipment;

CHAPTER 5 – GOOD PRACTICE

With this chapter three selected 'Good Practice SDEC systems' from Austria, Australia and Italy are presented along the entire project phase, i.e. design and operational phase. The SDEC projects were scientifically accompanied by SHC Task48 participants, therefore first analysis of simulation results of the SDEC technology are documented. The SDEC systems are equipped with measurement devices which fulfil the requirements of the 3rd level evaluation according to the IEA SHC Task 38 monitoring procedure[3]. The energy performance of the SDEC systems operation is displayed by monthly values of both energy fluxes and key performance indicators. The 'Good Practice SDEC system' report on each system closes with findings and lessons learned in order to guide next projects; What quality and support measures lead to a successful SDEC system implementation with high energy performance, high quality of indoor comfort and high user friendliness for facility manager?

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2 http://iea-shc-task38.org/reports/Report_B1_final-2.pdf/view

3 http://task38.iea-shc.org/data/sites/1/publications/IEA-Task38-Report_A3a-B3b-final.pdf

AUTHORS AND CONTRIBUTION

Table 1: SHC Tak48 B2 report contributing authors and entities

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ACKNOWLEDGEMENTS

The Austrian contribution to IEA SHC TASK48 is funded by the Austrian Ministry for Transport, Innovation and Technology within the Call 'IEA Ausschreibung 2012'.

The Australian authors acknowledge the financial support of NSW Government's Climate Change Fund for carrying out this work.

2. WORLDWIDE INSTALLED SDEC SYSTEMS

Authors: T. Selke, M. Perisic, McNevin Ch.

According to the market survey conducted in SHC Task 38[4], solar-heat driven desiccant evaporative cooling (SDEC) systems have a low market share in the Solar Air-Conditioning (SAC) market compared to absorption and adsorption chillers. Out of 113 SAC systems, only 18 open cycle systems identified were equipped with DEC technology. Solid sorption materials are clearly dominating the applied DEC technology; only two DEC systems have been identified that operate with a liquid sorption material.

Screening process and data base

Starting during the second quarter of 2012, experts from the Austrian Institute of Technology (AIT) conducted a successive market screening as part of activity B2 of the IEA SHC's Task 48. Thirty SDEC plants were identified world-wide during two screening phases. Some SDEC data had been previously collected during IEA Task 38. This data was updated through intensive communications between the contact persons of each identified SDEC and AIT. This led to the update and confirmation of 15 SDEC data sets. For nine SDEC systems, there was no reply and, in case of six plants, no information and/or contact person could be found. Figure 21 illustrates the response rate of the SDEC survey. For the 15 systems with no information or contact response, the data cannot be guaranteed as up-to-date, correct or complete.

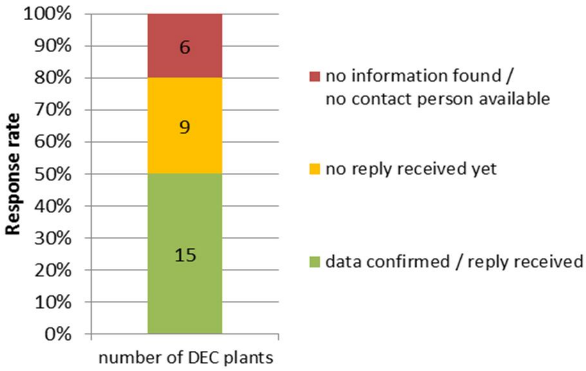

Figure 1: Response rate of screening

Analysis and findings

Within this chapter, the SDEC systems will be described by a) the location of the installation (country), b) the building category (office, school, etc.), c) the sorption material, d) the solar collector technology and e) the heating and cooling back-up systems.

The majority of the operational SDEC systems are located in Germany, followed by Italy and Austria (Figure 2). Approximately two-thirds of the installed SDEC plants are located in these three countries. Overall, the market screening indicates that the 30 SDEC plants are located in 13 different countries.

DEC technology, without suitable technical adaption, is not applicable for all regions, especially in regions with extremely hot and humid climates. The thermodynamic limits and

[4] http://iea-shc-task38.org/reports/Report_B1_final-2.pdf/view

the limited dehumidification ratio of a sorption wheel indicate that the standard configuration of the DEC technology is appropriate for moderate climates (i.e., central and southern Europe).

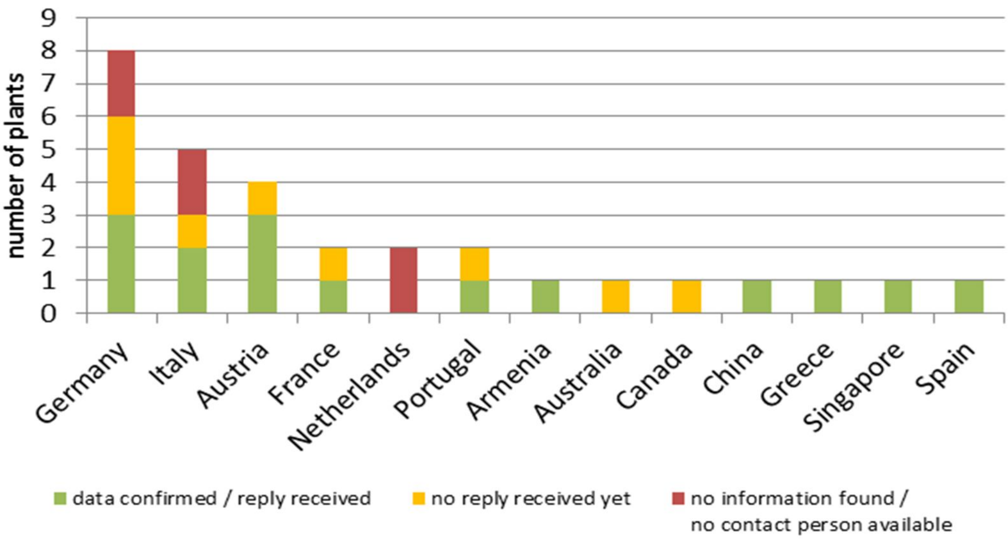

Figure 2: Solar-driven DEC plants – Location and response rate

Figure 3 shows the different building categories which are air-conditioned by SDEC systems. The SDEC plants are used in seven different building categories (offices, test facilities, multi-purpose spaces, public buildings, educational buildings, industrial buildings and kitchen spaces). Ten SDEC systems supply conditioned air to offices, four are applied in test facilities, three SDEC systems air-condition multi-purpose spaces and a further three are used for public buildings. Four SDEC systems' building type is unknown. The installation of SDEC systems in laboratories indicates ongoing R&D and technology improvements in the field of SAC in buildings.

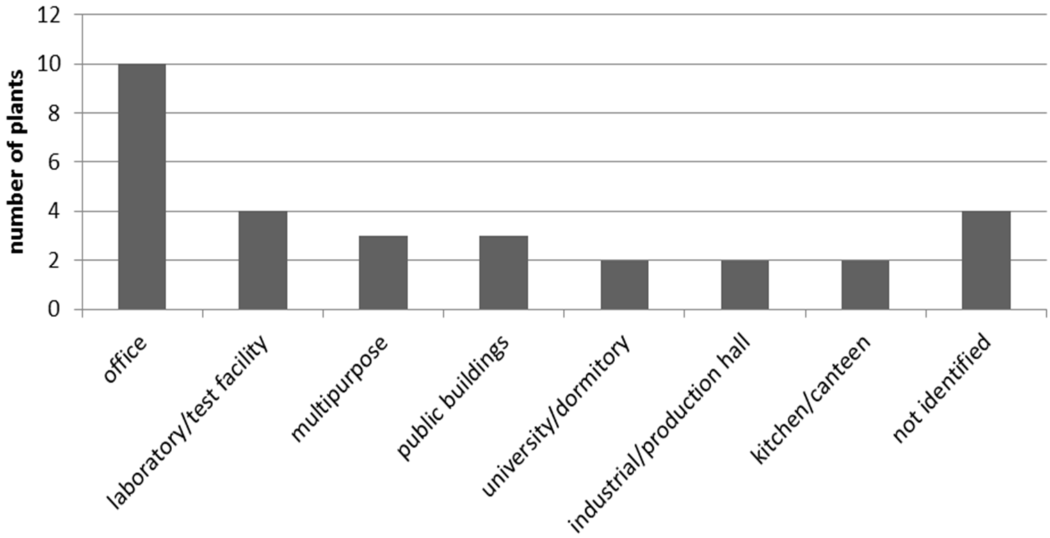

Figure 3: Building categories – Overview

DEC technology utilizes three physical operations to condition air: a) air dehumidification via sorption materials, b) direct evaporative cooling and c) heat recovery. Different sorption materials are readily available to fulfil the dehumidification process in both liquid and solid forms. Solid sorption materials in use by the systems under study include silica gel, zeolite or a cellulose matrix impregnated with lithium-chloride. Solid sorption materials are used by 21 out of 30 identified DEC plants. One system was not identified and the rest dehumidify air using liquid sorption materials such as aqueous solutions of lithium-chloride. Other liquid desiccant solutions exist but are not in use by the systems under study. The distribution of the different sorption materials is visualized in Figure 4.

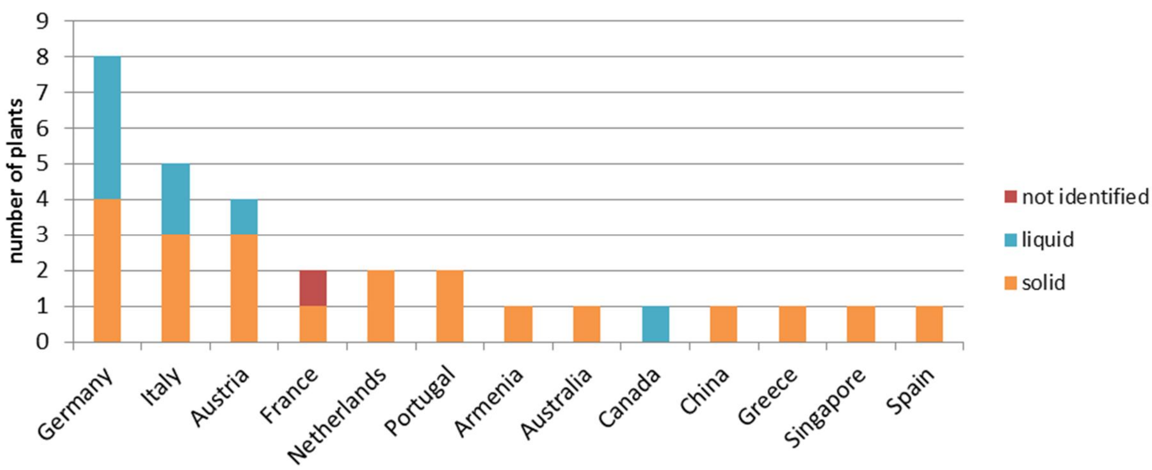

Figure 4: DEC sorption material used in the identified countries

The capacity of air-conditioning systems is typically represented by the nominal cooling power in kilowatts and/or the nominal air volumetric flow rate in m³ per hour. The authors categorized the studied DEC systems into three categories based on flow rate in Figure 5. The first category contains all DEC systems below 5,000 m³ per hour, the second category counts all systems between 5,000 and 15,000 m³ per hour and the final category represents the plants above 15,000 m³ per hour. Seven SDEC units fit within the low air flow category, 19 SDEC units can be linked to the middle category and only three SDEC units fit into to the above 15,000 m³ per hour category. The nominal air volumetric flow rates indicate the air flow design parameter for the SDEC unit but, that does not guarantee that the SDEC system is always operated at this flow rate.

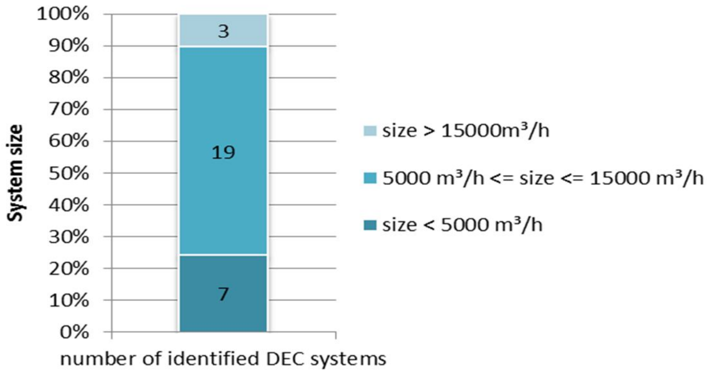

Figure 5: SDEC system size

Heat-driven DEC systems require specific temperature levels in order to regenerate the sorption material after it becomes saturated with moisture from the process air. In the SDEC systems under study, the heat is provided by solar collector systems of different types and sizes. The type of collector used mainly depends on the temperature needed for regeneration. Systems with regeneration temperatures from 60°C to 90°C typically use flat-plate collectors (FPC), while those with regeneration temperatures of over 90°C use evacuated tube collectors (ETC) (due to the lower heat losses at higher temperatures and therefore better efficiency). Compound parabolic concentrator collectors (CPC) can only be used effectively if a high percentage of the solar irradiation consists of direct radiation and thus have limited appeal in many locations. The findings of the investigation on solar collector technology are illustrated in Figure 6, Figure 7 and Figure 8. FPC technology is the most common type of collector used, with 13 systems utilizing this technology. Six SDEC are coupled with air collectors and five SDEC systems use ETCs. Only one system was reported to use CPCs (by a SDEC system located in Portugal). The solar collector technology is unknown in four systems. The solar collector arrays' gross areas range from 10 m² to approximately 400 m² in the systems under study.

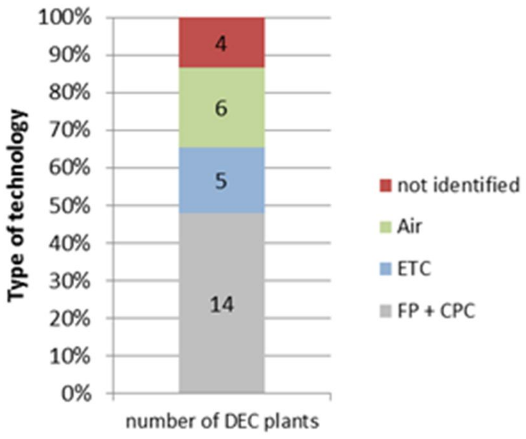

Figure 6: Solar collector technology (Air: air collector ETC: evacuated tube collector FPC: flat plate collector CPC: Compound parabolic concentrator collector)

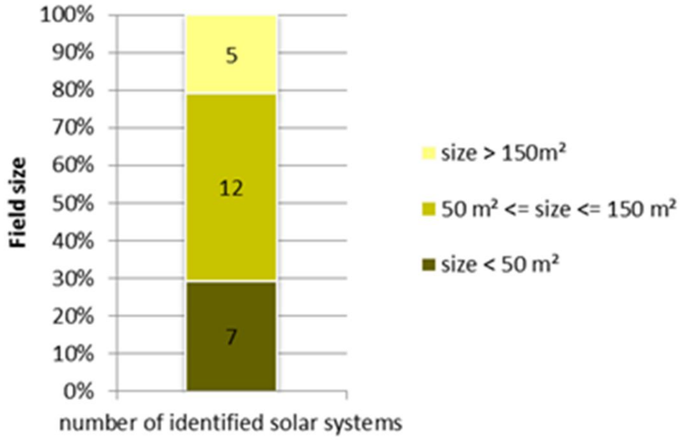

Figure 7: Collector field size

Comparing the solar technology to the countries of their deployment, a mix of collector technologies was observed with no direct link between countries and collector technology. In one special case three different collector types (FPC, ETC and CPC) have been installed at one SDEC plant in Germany.

Unfortunately, there was no reliable data describing the back-up thermal systems used for driving the DEC process during periods of insufficient solar irradiance.

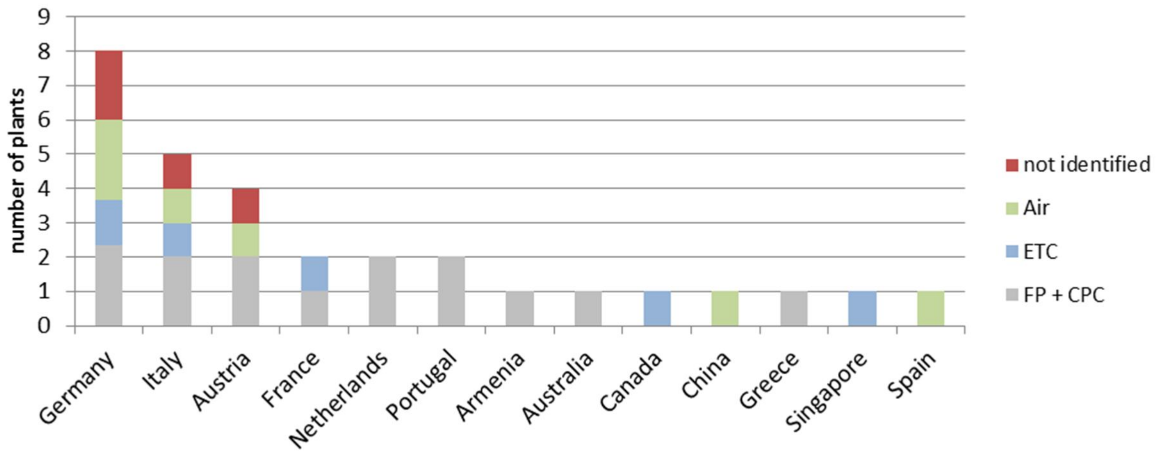

Figure 8: Solar collector technologies referring to countries

Remarks of the author: This article is partly published in SHC Newsletter (contributed by SHC Task 48 expert Tim Selke of the Austrian Institute of Technology, tim.selke@ait.ac.at; and SHC Task 48 Operating Agent Daniel Mugnier of TECSOL, France, daniel.mugnier@tecsol.fr. For more information go to the SHC Task 48 website http://task48.iea-shc.org/)

3. NEW TECHNICAL DEVELOPMENTS

3.1. FREESCOO

Authors: Finocchiaro P., Beccali M.

Compact DEC system operated with a fixed and cooled adsorption bed

An innovative compact solar air conditioner designed for ventilation, dehumidification, heating and cooling was developed. The system was designed for the air-conditioning of under-roof spaces and can be configured to be installed both on flat or sloped roofs. A casing contains the solar air collector, two adsorption beds, an integrated cooling tower, two wet-plate heat exchangers, fans and all other auxiliaries needed to accomplish the air handling process.

The system used two fixed desiccant packed-beds of silica gel, which are operated in a batch process, and two wet evaporative heat exchangers connected in series. The adsorption bed uses a fin and tube heat exchanger with the spaces between the fins filled with silica gel grains. The adsorption material is cooled by water flowing through the tubes. A system of air dumpers provides the commutation between the two adsorption beds in order to guarantee a continuous dehumidification process. For a detailed description of the concept and performance results refer to the bibliography of this section.

A cooling tower, which is integrated in the system, is used to reject the adsorption heat generated by the desiccant bed operating when in dehumidification mode. Regeneration is carried out using a solar air collector. The air flow rate passing the adsorption bed is 200 m^3/hr, which is 40% of the air delivered to the conditioned space (500 m^3/hr). Electricity is only consumed by the operation of three fans and two pumps. Cooling power is controlled by variable speed fans. Figure 9 shows the concept and the scheme of the system.

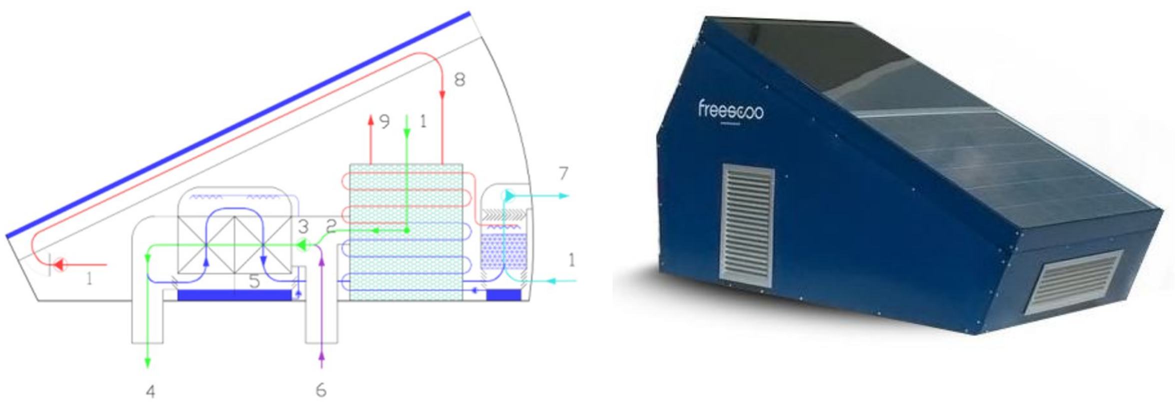

Figure 9: Solar air conditioner: scheme of the system (left) and picture of the freescoo unit (right)

The thermodynamic cycle of the process-air is described in Figure 10. A flow rate of outside air (1) is drawn through one of the adsorption beds for its dehumidification and partial cooling. Due to the simultaneous moisture and heat exchange, dehumidification process is carried out at almost constant temperature (2). Afterwards, dehumidified air is mixed with the return air from the building, which is at condition (4), reaching the conditions of point (3). The mixed air, which has a flow rate equal to 140% of the air flow rate supplied to the building, enters the wet heat exchangers reaching the supply conditions at point (5). In order to produce the cooling effect, at the outlet of the second wet heat exchanger, a portion of the air flow rate equal to 40% is drawn to the secondary side. The heat released in the adsorption bed is rejected to a water loop which is connected to the cooling tower integrated into the system. The air flowing through the cooling tower comes from the secondary side of the wet heat exchangers.

		Description	x	T	h
		-	g/kg	°C	kJ/kg
Process air	1	Outside air	16.0	36.0	77.2
	2	Adsorption bed	8.0	34.0	54.6
	3	Mixing	9.4	28.6	52.6
	5	Outlet Wet HX - prim	9.4	19.0	42.8
Building	4	Return air	10.0	26.0	51.6
Secondary air in Wet HX	5	Inlet Wet HX - sec	9.4	19.0	42.8
	6	Outlet Wet HX - sec	10.7	17.0	44.2
Cooling tower	6	Inlet CT	18.0	24.0	69.9
	7	Outlet CT	25.5	29.5	94.8
Regeneration air	1	Outside air	16.0	36.0	77.2
	8	Solar collector	16.0	58.0	100.0
	9	Outlet Desorption	24.0	39.0	100.9

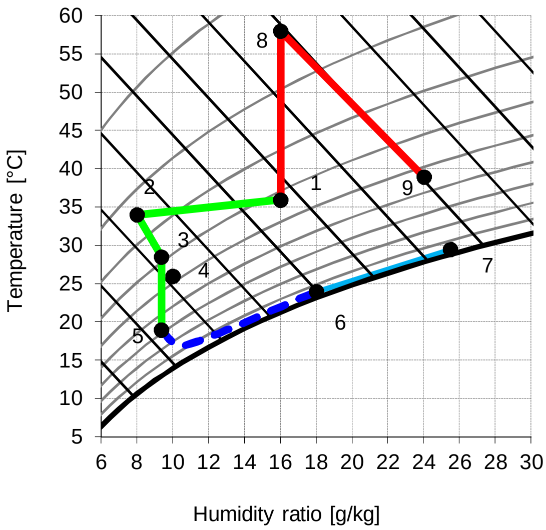

Figure 10: Thermodynamic cycle on a psychrometric chart

Control strategy

The control strategy of the system is the following. If there is no need for cooling in the building, solar energy is used to regenerate the adsorption material of the desiccant beds. In particular, one bed is regenerated until the temperature difference between the air at the outlet of the solar collector and the air coming out from the bed is higher than a fixed threshold. If the difference is lower and the solar fan is at the minimum speed, the control system commutes to the other bed for its regeneration. If the system has to provide cooling, the main fan is used to provide fresh and dehumidified air to the building. Building temperature and humidity can be controlled independently. Temperature can be adjusted controlling the speed of the main fan and by the status of the recirculation pump of the wet heat exchangers. Humidity can be adjusted by controlling the status of the cooling tower pump and partially controlling the speed of the main fan. A variation in the temperature of adsorption material will result in a different dehumidification capacity and consequently, this property can be used to adjust the humidity in the conditioned space. When cooling is required, the operation of the two adsorption beds is based on the humidity of the return air. If the humidity set-point is exceeded, then the control system activates the commutation procedure from one bed to the other. Before the end of this phase a pre-cooling of the bed which was operated in regeneration mode is carried out, preparing it for the next operation in adsorption mode.

Energy performances

Results come from monitoring data acquired during 15 days of full operation of a freescoo unit installed at ENEA in Italy in summer 2014.

First of all, climatic conditions are shown in Figure 12. Temperatures and humidity ratio occurred during the selected weeks very well represent the typical summer operation conditions of the specific site.

In the following diagrams, data are related to a system operation time from 9:00 in the morning to 18:00 in the afternoon.

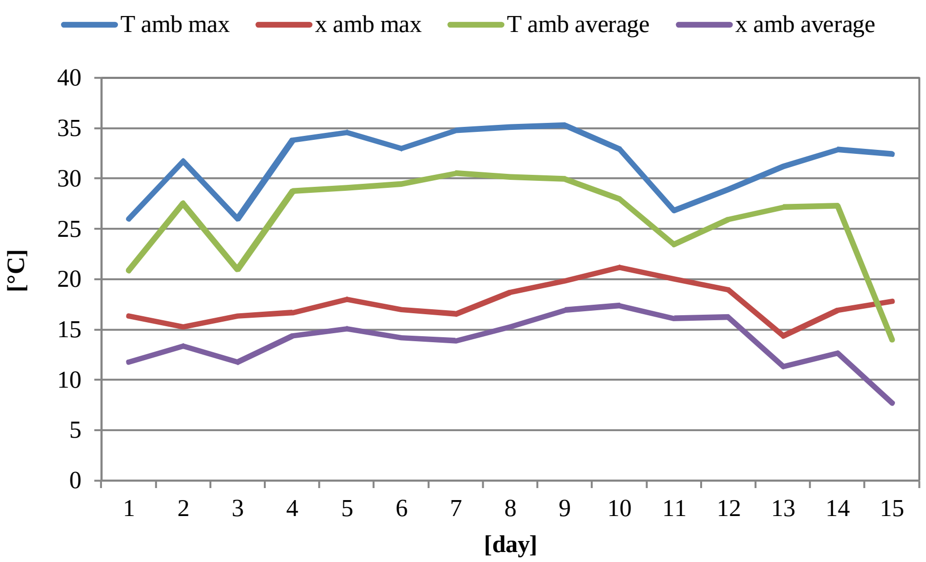

Figure 11: Average temperatures and humidity for the selected days of operation/ freescoo prototype at ENEA

Figure 12 summarizes the performances of the system in terms of energy production and electricity consumption. It has to be pointed out that the values shown for electricity consumption do not take into account the electricity production from the PV modules. This permits to better investigate the intrinsic electric efficiency of the machine. The average energy efficiency ratio calculated as the whole cooling energy delivered to the total electricity needed is 8.2 – see Figure 13.

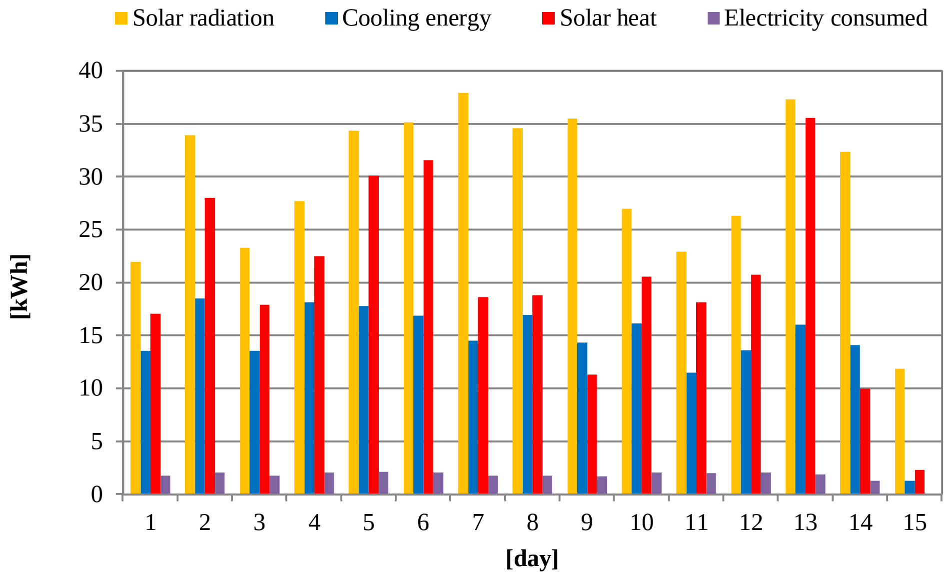

Figure 12: Energy produced and electricity consumed for the selected days of operation

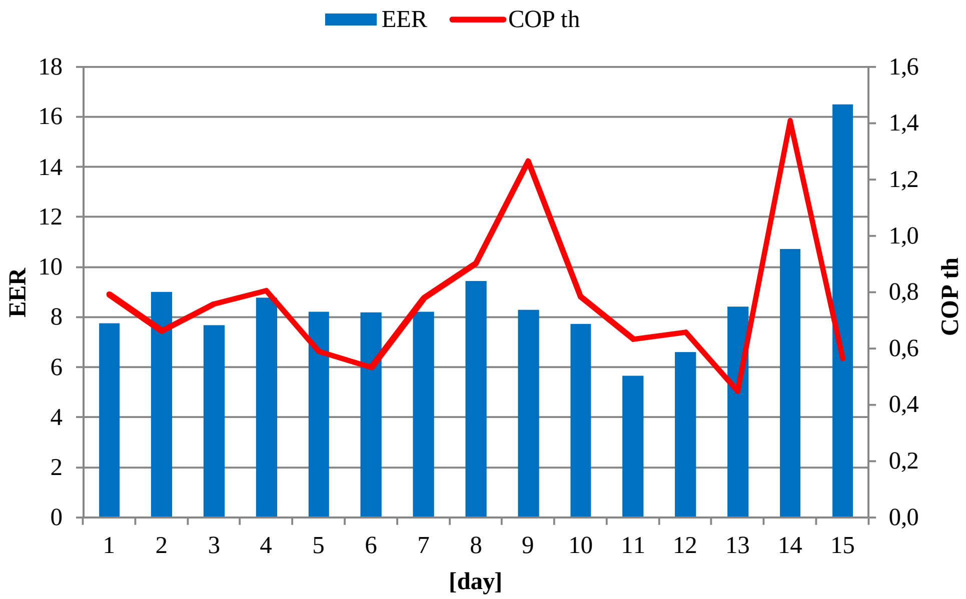

Figure 13: EER and thermal COP for the selected days of operation

Finally some data related to the stand alone operation of the system are shown. In particular, Figure 14 reports the electricity coming from the grid and the PV production during the selected days. As one can notice, system was able to run in stand-alone mode for seven days without any use of energy form the grid. This is a remarkable result which has to be considered as an important added value to the performance figures of the system.

If the calculation of the EER is made considering the real electricity derived from the grid, the average value for the considered operation time is 30.7.

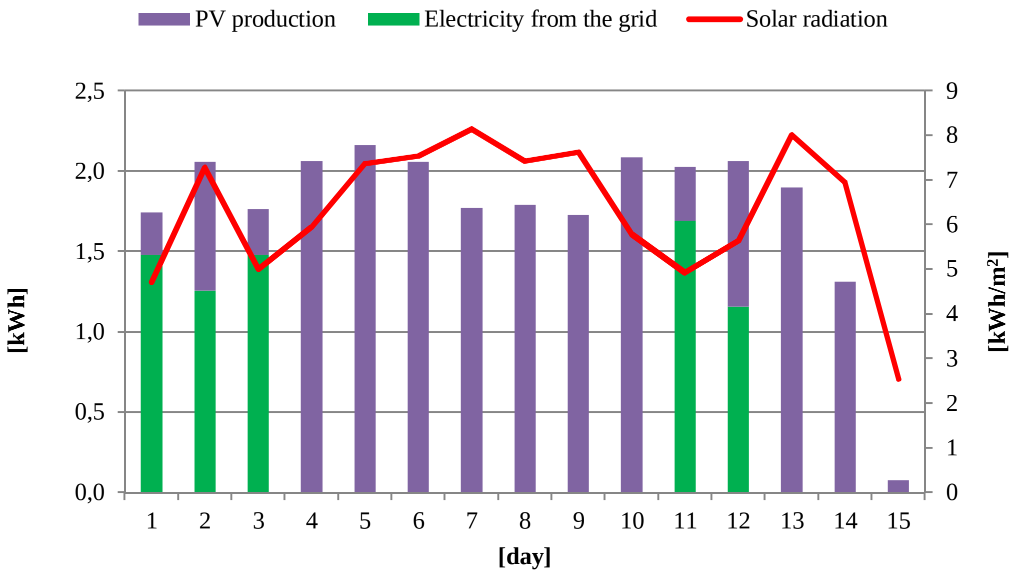

Figure 14: Electricity consumption from the grid and PV production during two weeks of operation

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Finocchiaro P, Beccali M, Gentile V. Experimental investigation of adsorption performances of an heat exchanger packed with silica gel for application in solar desiccant cooling systems. Proceedings of OTTI 5th Solar Air-Conditioning Conference, Black Forest: 25 – 27 Sept 2013, p. 210-215

Finocchiaro P., Beccali M., Calabrese A., Moreci E. Second generation of freescoo Solar DEC prototypes for residential applications SHC 2014 Energy Procedia 2015

3.2. ECOS

Author: Bongs C., Morgenstern A.

Prototype operation of an open sorption based air-conditioning system under tropical climatic conditions in Singapore

System description

The first prototype of an open sorption based ventilation device has been operated and investigated in the tropical climatic conditions of Singapore. The technology was based on the principle of the ECOS-system (Evaporatively COoled Sorptive Heat Exchanger) using two sorptive coated cross-flow air-to-air heat exchangers operated in an alternating mode. The supply air passing the channels with the sorptive coating is dehumidified while the building return air is passed through the cooling channels. Water is sprayed into these channels and evaporates which leads to a significant cooling effect on the supply air side. Thus, an enhanced dehumidification effect is achieved and at the same time the supply air is cooled to a temperature below the ambient air temperature [Morgenstern 2009, Bongs 2011 and Bongs 2012]. With the operation of the two heat exchangers in parallel, a quasi-continuous supply of fresh air is provided to the room.

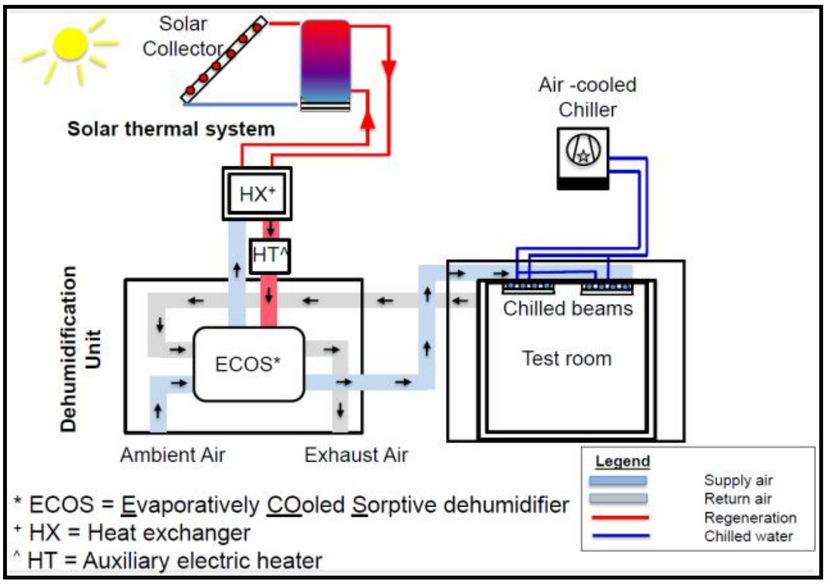

Figure 15: System scheme of the prototype installation at SERIS

With cooperation between the Solar Energy Research Institute of Singapore (SERIS) and the Fraunhofer-Institute for Solar Energy Systems (ISE), a prototype system of the ventilation device was installed at SERIS to provide supply air to a test room (Figure 15). Figure 16 shows the prototype installed at the SERIS laboratory. As the required cooling capacity could not be provided with the ECOS-system as a standalone appliance, an additional air-cooled chiller was installed for further cooling of the supply air via chilled beams. The aim of the system configuration was to demonstrate the significant advantage of the separation of handling the latent and sensible loads compared to a conventional system. Since the dehumidification was achieved by the sorption process, no cooling below the dew point was needed and the chilled beams were operated at a relatively high set point temperature of 18°C.

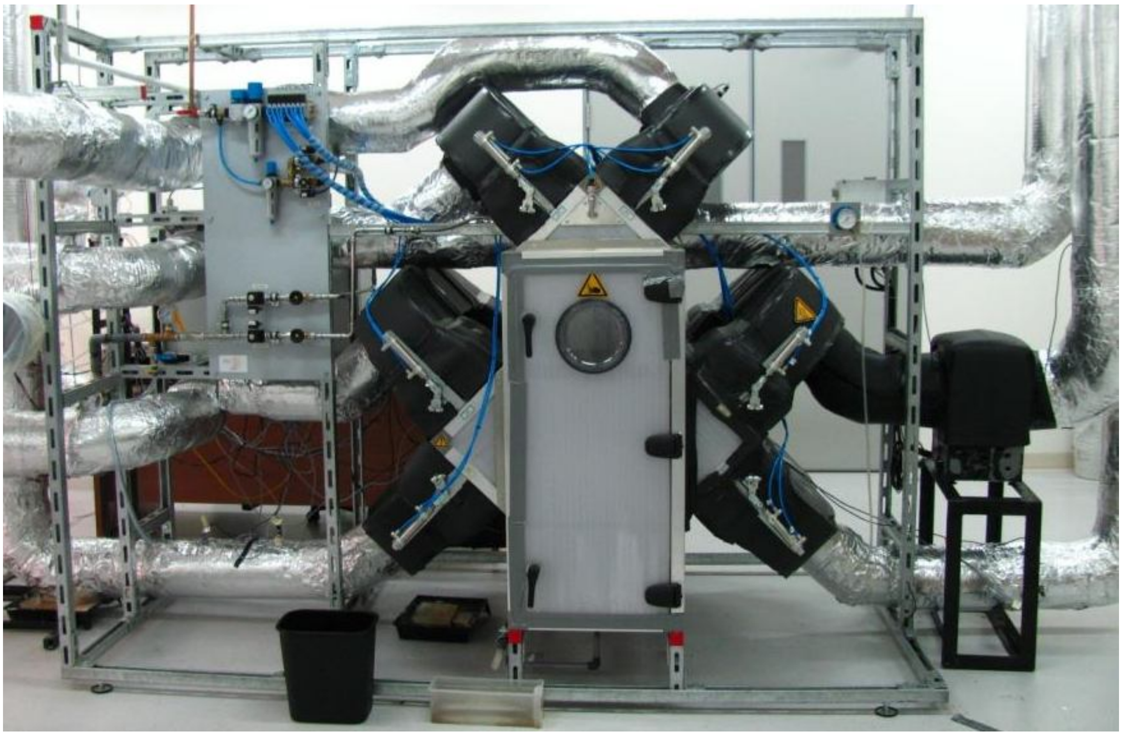

Figure 16: Prototype system installation at SERIS

System operation

The ventilation device was operated with equal air flow rates of 200 m³/hr for the supply-air, the return-air and the regeneration-air. The regeneration temperature was adjusted between 75 and 80°C to achieve the best dehumidification. Figure 17 and Figure 18 show an example of the measured values of the supply and return air. To ease the evaluation of the system behavior, averaged values of the supply air were calculated and shown in the figures.

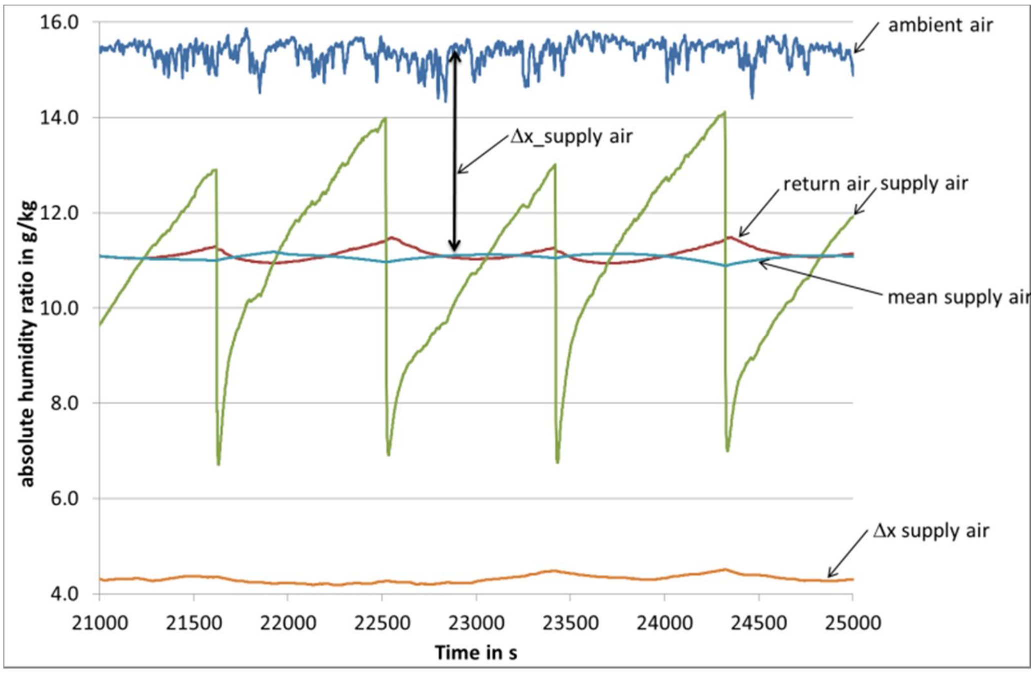

Figure 17: Sample curves of absolute humidity ratio results at equal air volume flow rates

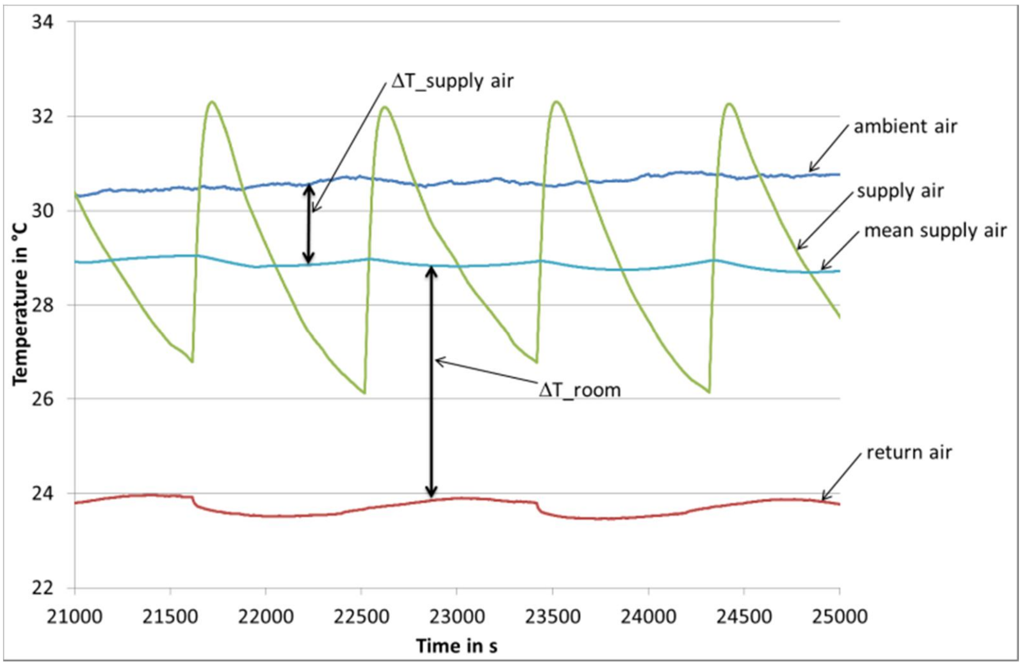

Figure 18: Sample curves of temperature behavior at equal air volume flow rates

The measurement campaigns were conducted using ambient air, thus, fluctuations occurred in the inlet air humidity and temperature. The inlet air humidity ratio ranged from 14 to 18 g/kg and the temperature ranged between 28 to 32°C. After evaluating the data, the optimum cycle time and the optimum pre-cooling time were determined to be 15 minutes and two minutes, respectively, for the given prototype.

Operation results and options for optimization

From the examples in Figure 17 and Figure 18, it is obvious that both heat exchangers have different dehumidification and cooling behaviors. The chilled beams were active which resulted in further temperature control of the room's supply air. No additional dehumidification was caused by the chilled beams due to the high set point temperature of 18 °C. From the return air values, it can be seen that the room smoothed the humidity fluctuations imposed by the ECOS operation. The room acted as a buffer volume despite the fact that there were no occupant loads in the test room during these experiments.

Differences in the ambient air humidity ratio resulted in a variable driving force for the adsorption process at equivalent regeneration temperatures. At higher humidity values (e.g., $x_{ambient}$=17.1 g/kg), this led to a dehumidification of up to Δx=5.3 g/kg while lower values (e.g., $x_{ambient}$=15 g/kg) led to a dehumidification of up to Δx=4.6 g/kg.

Higher values of dehumidification were achieved with a modified system operation when the regeneration air flow rate was double the supply air flow rate. This led to a mean dehumidification up to Δx=6.27 g/kg which lay in the upper range of the expected dehumidification performance. Lower desorption temperatures led to a significantly lower dehumidification rate and are thus not applicable in the high-humidity Singapore climate.

In terms of temperature reduction, the ECOS device did not reach the expected performance. The achieved values of temperature depression were only up to 2.25°C. There were several reasons for this. Higher spray water flow rates might be required which could be attained by using a different type of spray nozzle. Furthermore, it was discovered that a different air flow management design could lead to an optimized thermal management and therefore higher sensible cooling effectiveness.

In spite of the very good dehumidification performance the values of the COP are still relatively low (about 0.31) compared with classical, rotary wheel based DEC-systems. From previous experiments analysing the behaviour of a single component of sorptive coated heat exchangers, higher COP's were expected. The main reason for the low COP was the relatively low temperature depression achieved by the evaporation process in the return air. Improvement of sensible cooling performance would be within the scope of future work.

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3.3. INTER-COOLING TWO-ROTOR, TWO-STAGE ROTARY DESICCANT COOLING SYSTEM

Author: DAI Yanjun

Theoretical analysis

Figure 19 shows an example of process air in desiccant wheel presented on a psychrometric chart. The states p1 and r1 are the inlet condition of process air and regeneration air respectively. p2* has the same enthalpy as state p1 and the same relative humidity ratio as r1. r2* has the same enthalpy as state r1 and the same relative humidity ratio as p1. p2* and r2* are the ideal outlet state of process air as well as regeneration air. However, due to the irreversibility in actual process, the actual outlet states of process air and regeneration air are represented by p2 and r2, which should fall into the zone of “p1→p2*→r1→r2*→p1”.

According to the second thermodynamic law, the isothermal dehumidification as shown by process p1→p4 in Figure 19 is thought as one of ideal air conditioning process with the smallest irreversibility. On the other hand, the regeneration temperature of the ideal dehumidification system is minimal and outlet process air could achieve the lowest humidity ratio. Since lower regeneration temperature can be used and thermal energy can be recovered in the intercooler, the thermal energy used to power the system is reduced significantly. If the process air flows alternately over a number of desiccant wheels and intercoolers (heat exchangers), the system is named as multistage rotary desiccant cooling system whose thermodynamic process is represented by p1→p3 in Figure 19. With other conditions being equal, for the same outlet humidity ratio of process air (p3* and p2), regeneration temperature of multistage system (r3*) is lower than that of one-stage system (r2). Under the limiting condition, when process air flows alternately over infinite desiccant wheels and intercoolers, an ideal infinite multistage rotary desiccant cooling system can be produced and its dehumidification process (p1→p4) is an isothermal one.

In summary, by adopting inter cooling multistage rotary desiccant cooling system has great potential in reducing the energy need and enhancing the energy performance. Furthermore, since low humidity ratio can be achieved using such system, it can be used in high humid environment, as well as to achieve lower supply air temperature by using the evaporative cooler.

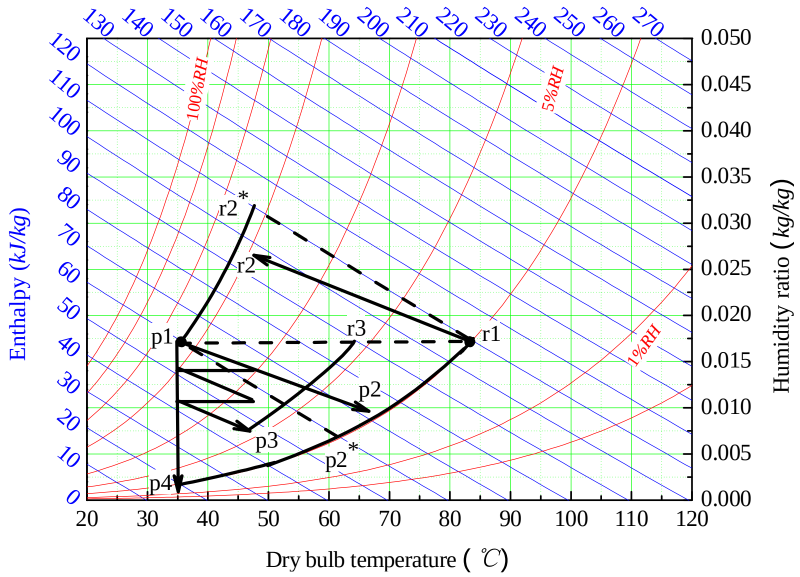
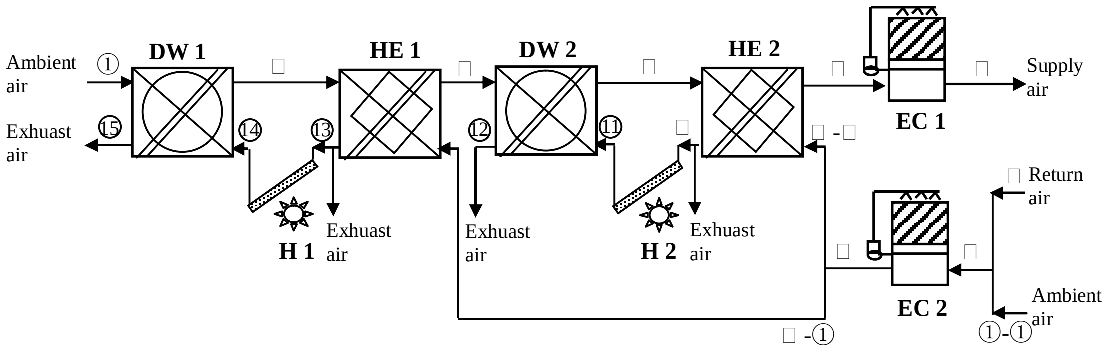


Figure 19: Comparison between one stage and multistage system in psychrometric chart

Realization of ideal infinite multistage desiccant cooling system is not practical. Therefore, inter-cooling two-rotor two-stage rotary wheel desiccant cooling (TSDC) system is proposed to make a tradeoff. Based on the above discussion, by the use of a novel configuration, dehumidification process of this original two-rotor two-stage rotary wheel desiccant cooling system is much more near to the isothermal one, and it has lower regeneration temperature and less irreversibility loss compared with the one-stage system.

The basic components of TSDC include desiccant wheel, heat exchanger, evaporative cooler and air heater. Figure 20 and Figure 21 represent the schematic form and psychrometric chart of TSDC. It can be seen that in the process air side, the air is dehumidified in series by two desiccant wheels. In this case, the air can be processed to a much lower humidity ratio in comparison with one-stage system. However, in the regeneration air side, if the two stages are connected in series, the air temperature at state 12 will be too high to cool the process air in heat exchanger 1. Hence, a parallel connection mode is adopted for regeneration air flow to achieve an effective cooling effort. The unique feature of this system is the novel configuration of the dehumidification and cooling components and the utilization of composite desiccant materials that both effectively improve the system performance.



(DW: Desiccant Wheel, HE: Heat Exchanger, DEC: Direct Evaporative Cooler, H: Heater)

Figure 20: Schematic figure of two-stage rotary wheel desiccant cooling system

The thermodynamic process of conventional one-stage rotary desiccant cooling system is also plotted in Figure 21 (for the process air: 1*→2*→3*→4*, for regeneration air: 5*→6*→7*→8*→9*). The data listed in Figure 21 are the simulated results obtained from Reference. It can be seen from Figure 21 that under the condition of the same outdoor (states 1 and1*) and indoor air (states 5* and 7) state, for the same humidity ratio of supply air, regeneration temperature of one-stage system is as high as 100°C, however that of TSDC is about 75°C. Meanwhile, the temperature of supply air in TSDC is also lower than that of one-stage system. Therefore, results of intuitive method show that TSDC has the merits of low regeneration temperature and good cooling capacity compared with one-stage system.

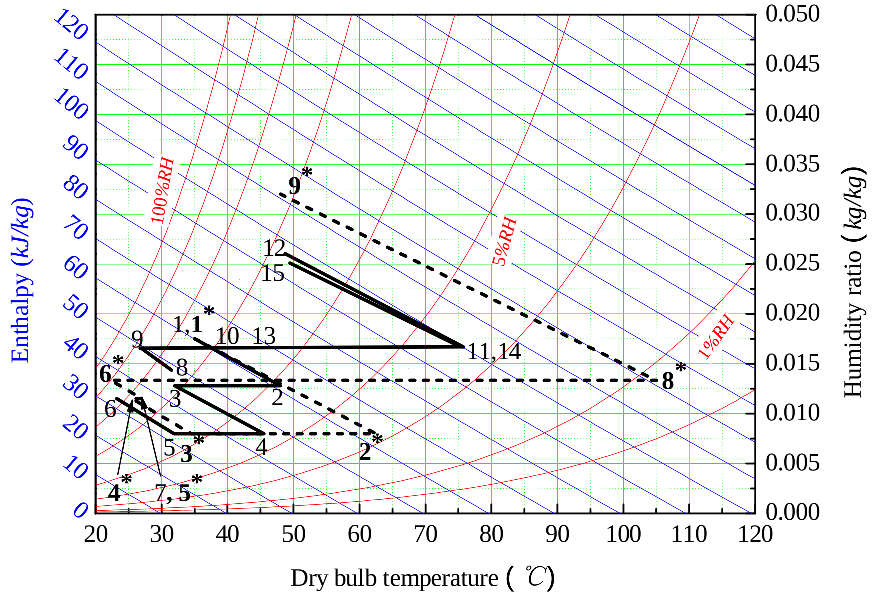


Figure 21: Two-stage desiccant cooling system and one-stage one plotted on phychrometric chart

A comprehensive energy evaluation of thermodynamic system should integrate the analysis based not only on first law but also on second law. And with the development and establishment of the second law of thermodynamics, more and more interests have been concentrated on the investigation of energy quality. Under the condition as plotted in Figure 21, the exergy loss and exergy efficiency of both TSDC and one-stage system are calculated. The exergy loss rate and exergy efficiency of both TSDC and one-stage system can be calculated and listed in Table 2. It can be seen from the table that as predicted by psychrometric chart, the exergy loss rate of TSDC decreases about 12% compared with one-stage system; meanwhile the exergy efficiency of TSDC increases from 71.5% to 83.1%, increases by 17%. These results qualitatively validated TSDC has the advantage of less irreversibility loss.

Table 2: Comparison results of the exergy loss rate and exergy efficiency

TSDC	Value	One-stage system	Value
Exergy loss rate i (W)	770.6	Exergy loss rate i (W)	859.0
Exergy efficiency (%)	83.1	Exergy efficiency (%)	71.5

Experimental investigation on two-rotor two-stage rotary desiccant cooling system

An experimental set-up with volume of 2m×0.95m×1.5m was built in our lab to test the performance of this novel two-stage rotary desiccant cooling system. The experimental set-up mainly consists of four parts: air-precondition unit which is used to provide air in specified conditions, desiccant wheel which is the core of the system, heat exchanger that is adopted to recover thermal energy, and air heater which makes regenerates the desiccant material. Figure 22 gives the photographic view of the air precondition unit and experimental set-up respectively.


(a)

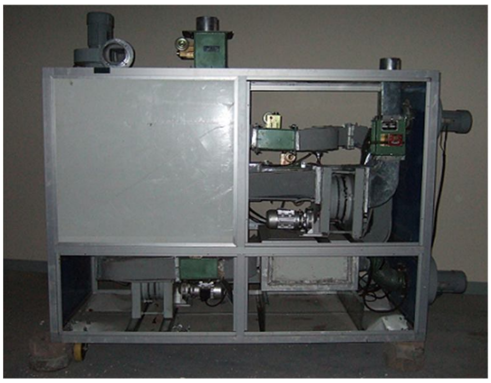

(b)

Figure 22: Photographic review of (a) air precondition unit (b) the experimental set up

There are two categories of performance indices adopted in the simulation analysis. The first group considers the effectiveness of the system in meeting the need of the indoor conditions, such as outlet temperature, humidity ratio of process air and cooling capacity.

The second group deals with the performance of whole system. Moisture removal is an important index to indicate the dehumidification capacity of desiccant wheel. It can be calculated by:

$$D = Y_{p,in} - Y_{p,out} = Y_1 - Y_5 \quad \text{Eq. [1]}$$

where Yp,in and Yp,out mean the inlet and outlet humidity ratio of process air respectively.

Cooling capacity is calculated by:

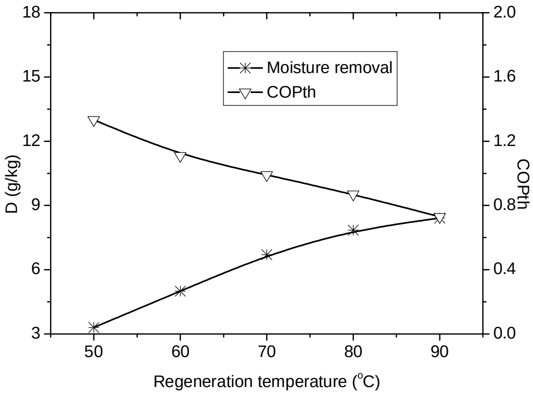
$$Q_c = M_p(h_6 - h_1) \quad \text{Eq. [2]}$$

Thermal coefficient of performance COP_{th} is used as another indicator to evaluate the performance of this TSDC. It is defined as the ratio of cooling power (Qc) to the regeneration heat used to regenerate the two desiccant wheels (Qr1, Qr2). The electrical power input of the fans is neglected for "thermal energy" is the primary source of energy.

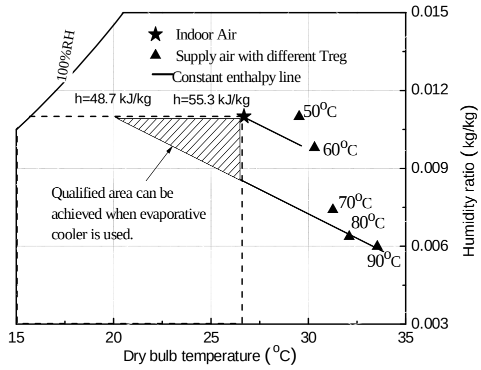
$$COP_{th} = \frac{Q_c}{Q_{r1} + Q_{r2}} = \frac{M_p(h_1 - h_6)}{M_{r1}(h_{14} - h_{13}) + M_{r2}(h_{11} - h_{10})} \quad \text{Eq. [3]}$$

First, system performances were tested under ARI summer condition. The average values of moisture removal capacity D and thermal coefficient of performance COP_{th} at different regeneration temperature Tr (T11 and T14) are shown in Figure 23 (a). Moisture removal capacity D shows the expected characteristic behavior: it increases smoothly with increasing Tr due to the desiccant being well regenerated. Whereas, COP_{th} decreases with increasing in Tr. Compared with the tested results of one-stage system, effect of regeneration temperature on the performance of these two systems is similar. Since two-stage system needs more energy to drive at higher Tr, its COP_{th} decreases more rapidly compared to the one-stage system in higher regeneration temperature. However, due to low regeneration temperature can be adopted, COP_{th} close to 1 can be achieved when Tr is around 75°C.

The tested outlet states at different regeneration temperature from 50°C to 90°C (triangle in the figure) and indoor air state (pentalpha in the figure) are plotted in Figure 23 (b). When the temperature and humidity ratio of supply air are lower than the indoor conditions, they are "qualified supply air state", which is schematically shown by the dashed line in Figure 23 (b). The qualified supply air area obtained by applying a direct evaporative cooler is plotted as the shadow area in Figure 23 (b). It is shown that this system could provide satisfied supply air when regeneration temperature is higher than 60°C. To achieve a tradeoff between the system energy performance and the supply air condition, regeneration temperature between 65°C and 80°C is more reasonable for the system operation.



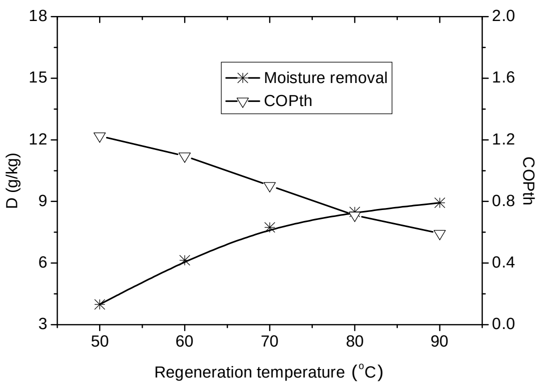
(a)



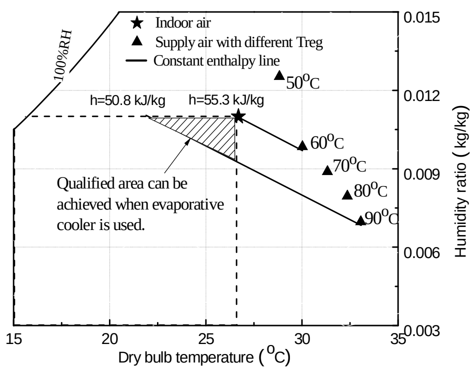
(b)

Figure 23: (*a) System performance with different regeneration temperature under ARI summer condition (b) Supply air state under ARI summer condition when an evaporative cooler is used*

The performance of TSDC system under ARI humid condition, as well as Shanghai summer condition is illustrated in) Figure *24* and Figure 25. These results are similar to that obtained under ARI summer condition: the system could provide appropriate supply air when regeneration temperature is higher than 60°C under ARI humid condition, in order to avoid the system operating with relatively low COPth, regeneration temperature from 65°C to 75°C is recommended in this condition. Under Shanghai summer condition, since more latent heat load is removed, COPth is higher compared to the previous two conditions at the same regeneration temperature. The COPth can be beyond 1.0 even if Tr reaches 90°C. It is observed in Figure 25 that when regeneration temperature is higher than 75°C, the system could provide satisfied supply air. Similarly, to make a tradeoff between supply air and COPth, Tr between 80°C and 90°C is recommended.

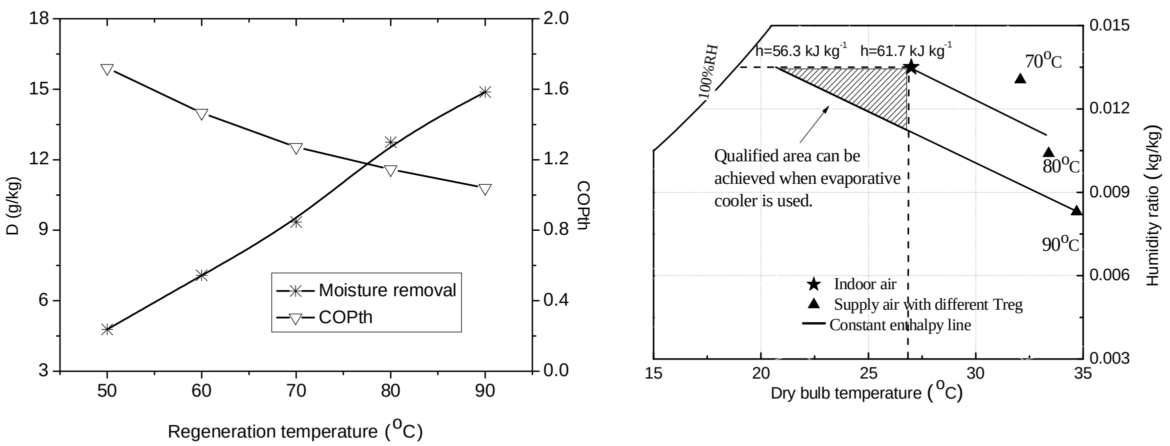


(a)



(b)

Figure 24: (a) System performance with different regeneration temperature under ARI humid condition (b) Supply air state under ARI humid condition when an evaporative cooler is used



a) (b)

Figure 25: (a) System performance with different regeneration temperature under Shanghai summer condition (b) Supply air state under Shanghai summer condition when an evaporative cooler is used

In conclusion, when regeneration temperatures are higher than 60°C, 60°C and 75°C respectively, this TSDC system could provide supply air meeting the requirements of temperature and humidity ratio under ARI summer, humid and Shanghai summer conditions. The required regeneration temperature is lower; therefore, low-grade thermal energy such as solar and geothermal energy can be efficiently utilized and the operating costs can be significantly reduced.

Performance comparison between TSDC and one-stage system

Performance comparison between TSDC and one-stage system is summarized in Figure 26. It can be found that TSDC is superior compare to conventional one-stage system. For the same moisture removal capacity, the required regeneration temperature of TSDC is 66°C, which is about 34% lower than that of one-stage system. Meanwhile, the cooling power and thermal COP of TSDC increase by 15% and 70% compared with one-stage system respectively. On the other hand, under the same regeneration temperature (100°C), moisture removal of TSDC is 76% higher than one-stage system. At the same time, cooling power and thermal COP are 32% and 28% higher.

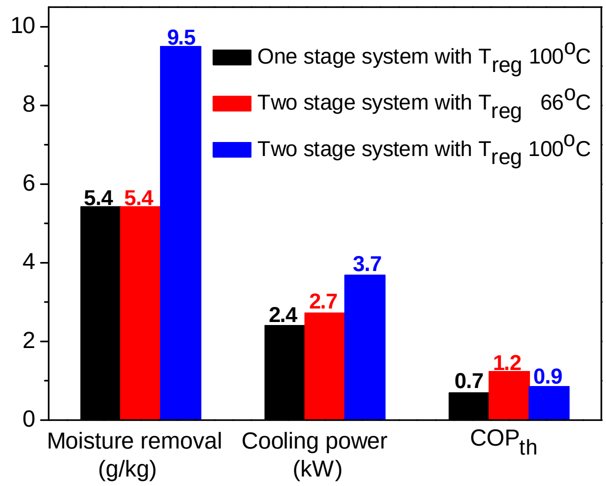


Figure 26: Performance comparison of two-stage and one-stage rotary desiccant cooling system

3.4. SOLAR DRIVEN INTERNALLY-COOLED, LOW-FLOW, FALLING-FILM LIQUID DESICCANT AIR-CONDITIONER

Authors: McNevin C., Harrison S.

System Description

A liquid desiccant air-conditioning system (LDAC) was designed around a prototype internally cooled/heated, low-flow, falling-film liquid desiccant unit produced by AIL Research [1]. The low flow characteristic ensured zero desiccant carry over into the air streams while the internal cooling and heating allowed for increased performance at the low desiccant flow rate. The LDAC system was supplied thermal energy, used for regeneration, by a 95 m^2 evacuated tube solar thermal array and a back-up gas boiler. The LDAC system is shown in Figure 27.

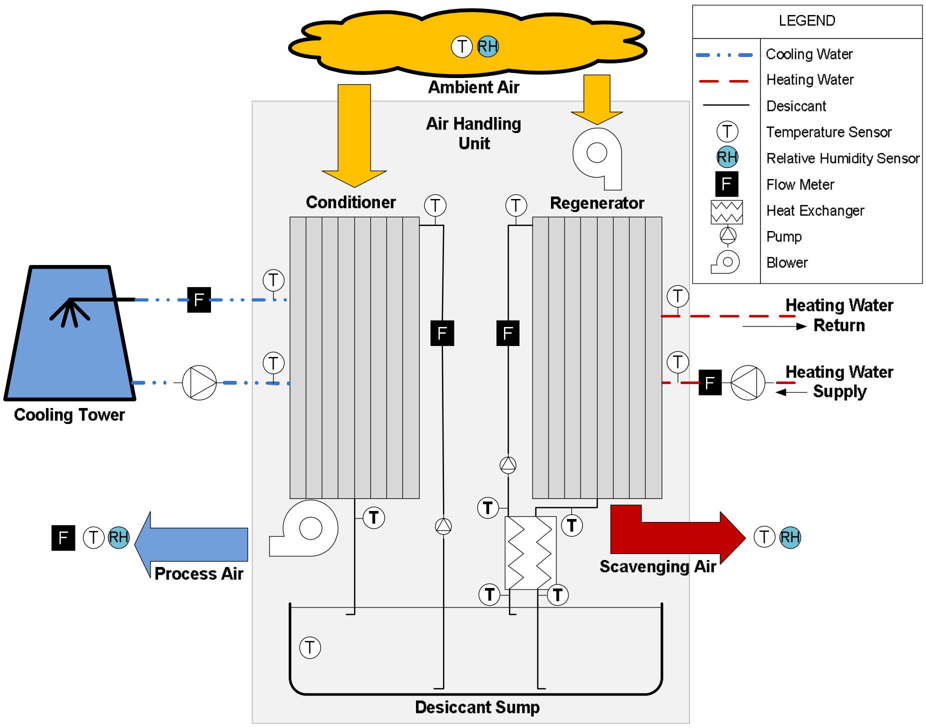


Figure 27 Simplified schematic of the liquid desiccant unit [2]

The conditioner was made up of 96 curved parallel plates. Each plate had internal passages for cooling water to pass through. The cooling water was used to cool the desiccant solution in order to improve its ability to absorb moisture and to provide a degree of sensible air cooling. The heat was rejected from the cooling water by an evaporative cooling tower. The process air stream was blown between the plates, at approximately 70 m^3/min, where it came into direct contact with the falling-film of desiccant solution in a cross flow configuration.

The regenerator required a slightly different design than the conditioner to account for the higher temperatures and greater thermal stresses. Operation was identical to that of the conditioner. The plates were internally heated to evaporate the moisture from the desiccant and into a scavenging air stream. To reduce the thermal energy demand of the regenerator, a heat exchanger recovered heat from the hot outlet desiccant solution into the desiccant stream entering the regenerator.

The solar thermal system provided a low environmental impact heat-source for the regenerator. The system was composed of five parallel banks of evacuated tube collectors. A pair of 435 liter insulated hot water tanks was used as a buffer between the LDAC and solar systems. This allowed each system to operate independently. During times of low thermal loads on the solar system (i.e. when the regenerator was not running), a dry cooler automatically activated when the fluid temperature rose above a set point of 95°C to prevent collector stagnation. A two-stage natural gas boiler was used to supplement the solar system during times of lower solar availability. f shows how the solar system and the LDAC unit were connected.

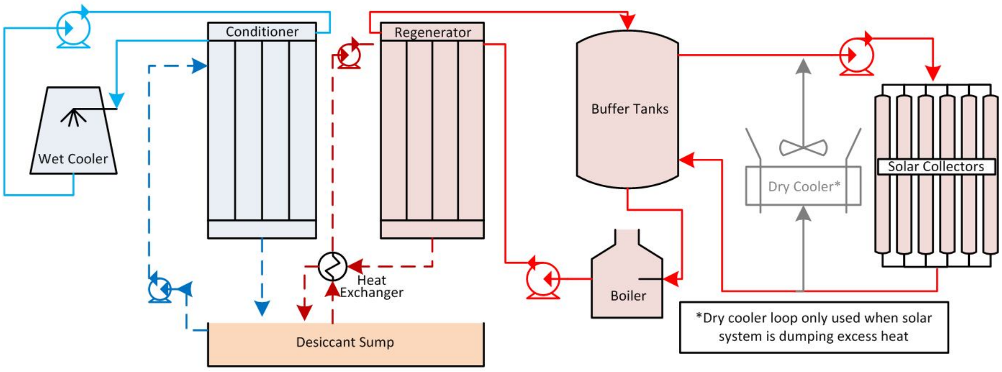


Figure 28. Simplified system schematic [3]

The LDAC system was controlled by a PLC system. The regenerator was activated by the PLC unit when the outlet process air's relative humidity exceeded 30%. This ensured that when the air leaving the unit became overly humid, the desiccant was strengthened to improve the water absorption ability of the desiccant. A separate controller ran the solar thermal system. It activated the water circulating pump once a temperature rise was seen in the collectors. A system of solenoid valves was used to send the water either to the dry cooler or the buffer tanks dependent on temperature. When the solar availability was low and the temperature increase across the collectors was small, the controller would deactivate the system.

Figure 29. (a) Liquid desiccant air handling unit and cooling tower as installed at Queen's University (b) Photo of the solar thermal collector array [4]

System Performance

The system was run using ambient air for both the regenerator and conditioner inlets for a series of tests. There was no load or building space connected to the system. The total cooling rate ranged from 9.2 kW to 17.2 kW. The average thermal COP was 0.40 and the electrical COP was 2.43. This included all power draws of the system (i.e., all pumps, fans, and the cooling tower). The solar array provided 40% of the required thermal energy with an average collector efficiency of 53%. These averages occurred over a wide range of weather conditions recorded over twenty, ten-hour long tests. A sample of the data recorded is shown in Figure 30.

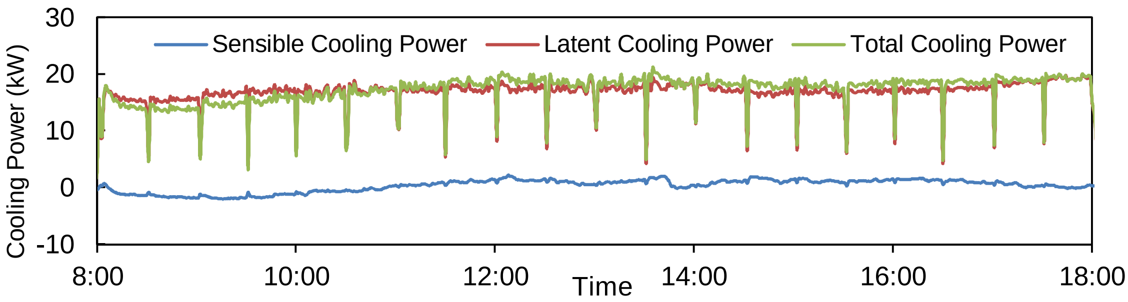


Figure 30. Experimental test results recorded on a sunny and humid day. Sharp spikes in thirty minute intervals are due to the short shut down of the system while collecting desiccant samples [5]

A TRNSYS simulation for the entire system has been developed. It uses an experimentally-determined effectiveness model to accurately predict the systems operation and performance. The latest iteration of the model has being used to predict performance improvements possible by use of energy recovery from waste streams [3], and load shifting using cold water storage [6]. Both of these methods indicate considerable improvements in COP, cooling rate, and solar fraction are possible.

Operation of the system has shed light on some interesting factors, these include,

- The system should be used as a dedicated outdoor air system in conjunction with systems that can provide sensible cooling as system
- Climates with high humidity and long cooling seasons will see the most benefit of this type of system
- Cooling water temperature plays a large role in the cooling performance of the system so the heat rejection system needs to be carefully selected to optimize performance
- Pumps and fans can consume large amounts of electrical power, correctly sized, high-efficiency units should be used to maximize the system's competitiveness
- In climates where temperatures can go below the freezing, all water must be drained from the system to prevent damage with extra care given to units which are prone to water retention, such as the conditioner and regenerator
- Limit use of exposed metals in components in contact with, or in close proximity to the

desiccant solution, as the solution is highly corrosive and will cause materials to oxidize

- Solar thermal systems are appropriate for this type of system as they can contribute greatly towards reducing non-renewable energy consumption and can provide heat during winter
- Use heat recovery where possible to minimize wastage and reduce primary energy use
- Energy storage systems (either thermal storage of hot and cold water or chemical storage of concentrated desiccant) can greatly improve system performance and reduce costs by distributing peak energy loads

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4. EXISTING QUALITY LABELS OF DIFFERENT SUBSYSTEMS OF SDEC SYSTEMS

Authors: Freine A., Aprile M., Muscherà M., Nagidi K.

QUALITY LABELS FOR SOLAR DEC SYSTEMS AND DESICCANT WHEELS

The purpose of this chapter is to describe several quality standards and certifications which define the performance specifications of desiccant wheels and solar DEC systems. This chapter is divided into four sections:

- Existing certification for regenerative heat exchangers;
- Standards for active dehumidification wheels;
- Manufacturers' technical and performance data for desiccant wheels;
- Desiccant-based dehumidification equipment.

4.1. EXISTING CERTIFICATIONS FOR REGENERATIVE/ROTARY HEAT EXCHANGERS

In this first section, existing certification methods for regenerative or rotary heat exchangers have been presented. Desiccant wheels are a sub-type of this device category. The definition of these devices[5], given by Eurovent and the AHRI certifications, consider the wheel's latent energy transfer. However, these certification procedures are not fully suitable for rating desiccant wheel regeneration when used in solar DEC systems. The following section presents an analysis of the EUROVENT RS 8/C/002-2014 (used mainly in Europe) and AHRI 1060 (used mainly in USA) certification programs and their limitations.

EUROVENT RS 8/C/002-2014: RATING STANDARD FOR THE CERTIFICATION OF REGENERATIVE HEAT EXCHANGERS

The Eurovent RS 8/C/002-2014 certification is a voluntary rating standard written by the European Committee of Air Handling and Refrigeration (Eurovent). They are a representative of the European refrigeration, air-conditioning, air-handling, heating and ventilation industry. Eurovent also represents trade associations from European and non-European countries.

The certification mark given by Eurovent RS 8/C/002-2014 guarantees that the products have been submitted to the Eurovent Certification Program for rating. This mark informs designers, installers and end users that the products have been accurately rated under standard conditions.

This rating standard establishes definitions and specifications for testing and rating Regenerative Heat Exchangers (RHEs). It defines a common method to rate the performance

[5] is a device incorporating an alternating storage system or a rotating cylinder or wheel for the purpose of transferring energy (sensible or total) from one air stream to the other. It incorporates heat transfer material, a drive mechanism, a casing or frame, and includes any seals which are provided to retard the bypassing and leakage of air from one air stream to the other.

of RHEs and allow for comparisons to be made between different models.

The following performance indices form the certified characteristics given by the standard (determined at air speeds of 2.0 m/s, 3.5 m/s, 5.0 m/s):

a) Sensible efficiency;

b) Latent efficiency;

c) Pressure drop.

All the tests presented in this standard are required to follow:

- EN 308 (June 1997): Heat exchangers – Test procedures for establishing performance of air to air and flue gases heat recovery devices;
- ARI Standard 1060-2001: Rating Air-to-Air Heat Exchangers for Energy Recovery Ventilation Equipment.

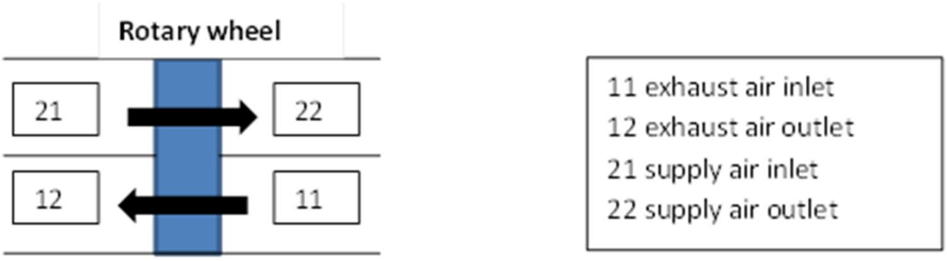
Definitions of terms used in the standard (see 10.1.1 for variables definition)

- Face air velocity: $v = \frac{\dot{V}_s}{A_{tot}/2}$;

  Standard conditions:

Table 3. Standard conditions for the Eurovent RS 8/c/002-2014 rating standard

Density	*1.2 kg/m^3 (0.075 lb/ft^3)*	*Relative humidity*	*50 %*
Temperature	*20 °C (68 °F)*	*atmospheric pressure*	*101.3 kPa (29.92 inHg)*



- Temperature efficiency (sensible): $\eta_T = \frac{T_{22}-T_{21}}{T_{11}-T_{21}}$
- Humidity efficiency (latent): $\eta_w = \frac{w_{22}-w_{21}}{w_{11}-w_{21}}$
- Pressure drop: the loss in total pressure between the inlet and outlet of a fluid stream
- External leakage: air loss between the casing and the environment: $\lambda_{ext} = \frac{\dot{m}_{L,ext}}{\dot{m}_{nom}}$
- Internal exhaust air leakage: air seepage between two air streams: $\lambda_{co} = \frac{\dot{m}_{carry\ over}}{\dot{m}_{nom}}$

The following test specifications are laid out by the standard:

- Tests must be performed:
  - at the rotor speed specified by the manufacturer;
  - at the manufacturer's specified purge angle or setting;
  - all ratings must be performed at the same rotor speed and purge setting, except for the external leakage test.
- External Leakage and Internal exhaust air tests will follow the guidelines presented in Table 4.

Table 4. Test specifications for external and internal exhaust air leakage

	Test conditions	Specifications
External Leakage	v=3.5 m/s (7.83 mph) According to EN 308	λ_{ext}< 3%
Internal exhaust a leakage (dragging according to EN 308)	$\Delta p\Big\|_{11}^{22}$ = 0÷20 Pa (0.0803 inH_2O) Nominal air flow v=3.5 m/s (7.83 mph) ρ=1.16÷1.24 kg/m^3 (0.0724÷0.0774 lb/ft^3) According to EN 308	λ_{co}< 3%

- Sensible Efficiency, Latent Efficiency and Pressure Drop tests will follow the guidelines presented in Table 5 (refer to Table 3 for indices definition).

Table 5: Sensible efficiency, latent efficiency and pressure drop test conditions

		Regular Test conditions		Additional Test conditions	
		Heating	Cooling	Heating	Cooling
Inlet supply airflow 21	T	2 °C (35.6 °F)	35 °C (95 °F)	-3 °C (26.6 °F)	35 °C (95 °F)
	RH	80%	50%	90%	14g/kg 22g/kg
Inlet exhaust airflow 11	T	22 °C (71.6 °F)	25 °C (77 °F)	22 °C (71.6 °F)	25 °C (77 °F)
	RH	45%	50%	11g/kg	50%
Outlet supply airflow 22	v	2 m/s - 3,5 m/s - 5 m/s (4.47 – 7.83 – 11.19 mph)		3,5 m/s (7.83 mph)	
Outlet exhaust airflow 12	v	Same as “outlet supply airflow 22” (mass flow ratio = 1)			
Pressure difference	$\Delta p\Big\|_{11}^{22}$	0 ÷ 20 Pa (0,0803 inH2C (in both air flows and for each test, the average of the two measure values has to be compared with average of the two rated values)			
Rotor speed		Specified by manufacturer			

Equipment classified as a “sorption rotor” has to fulfill the following additional requirement for latent efficiency:

- Sorption rotor : $\eta_w \geq 60\% \times \eta_T$

The performance specifications of products certified by the Eurovent standard can be found on the Eurovent certification website. Follow the link below:

- http://www.eurovent-certification.com (under Certified Products -> Search engine)

The latent and sensible efficiencies, plus the pressure drop of the product, are given at each of the three required air speeds (2 m/s, 3.5 m/s, 5 m/s).

ANSI/AHRI CERTIFICATION 1060-2011: PERFORMANCE RATING OF AIR-TO-AIR HEAT EXCHANGERS FOR ENERGY RECOVERY VENTILATION EQUIPMENT

The Air-conditioning, Heating, and Refrigeration Institute (AHRI) is a global trade association representing manufacturers of HVACR and water heating equipment with a strong focus on North American companies.

The AHRI certification program guarantees that the manufacturers’ claims have been validated by stringent tests.

This section will discuss the standard “ANSI/AHRI standard 1060: Performance rating of Air-to-

Air Heat Exchangers for Energy Recovery Ventilation Equipment 2011", as approved by American National Standards Institute (ANSI).

This certification concerns all types of Air-to-Air heat exchangers including heat pipes, fixed plates and rotary heat exchangers.

The following performance indicators are used as the **certified characteristics** and they are determined at both 75% and 100% of the rated airflow rate:

a) Pressure drop;

b) NET Energy Transfer Effectiveness (Sensible, Latent and Total);

c) EATR (Exhaust Air Transfer Ratio);

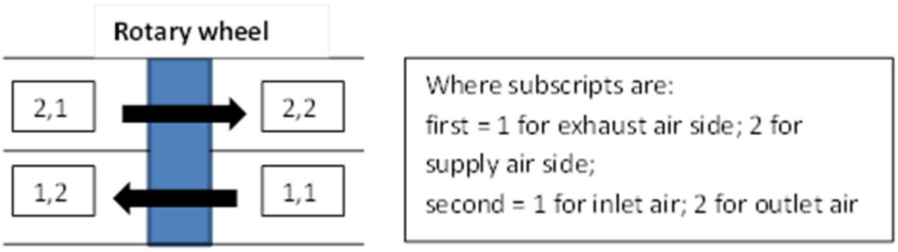
d) OACF (Outdoor Air Correction Factor).

Definitions of terms used in the standard (see 10.1.1 for variables definition)

- Standard air:

Table 6. Standard conditions for the ANSI/AHRI standard 1060 rating standard

Density	0.075 lb/ft^3 (1.2 kg/m^3)	Relative humidity	0%
Temperature	70 °F (21.1 °C)	atmospheric pressure	29.92 inHg (101.3 kPa)



- Effectiveness[*1]:

Sensible Effectiveness

$$\eta_T = \frac{(T_{21} - T_{22}) \cdot \dot{m}_2}{(T_{21} - T_{11}) \cdot min(\dot{m}_1, \dot{m}_2)}$$

Latent Effectiveness

$$\eta_w = \frac{(w_{21} - w_{22}) \cdot \dot{m}_2}{(w_{21} - w_{11}) \cdot min(\dot{m}_1, \dot{m}_2)}$$

Total Effectiveness[*2]

$$\eta_h = \frac{(h_{21} - h_{22}) \cdot \dot{m}_2}{(h_{21} - h_{11}) \cdot min(\dot{m}_1, \dot{m}_2)}$$

Net Effectiveness[*2]: The effectiveness is adjusted to account for the leakage (airflow.

*1: The Eurovent definition assumes the two flow rates are equal

*2: Additional parameters with respect to the Eurovent definitions

- Pressure drop: difference in the static pressure between the entering and the leaving supply airflow

- Exhaust Air Transfer Ratio (EATR)*3:

$$EATR = \frac{\dot{m}_{22} - \dot{m}_{21}}{\dot{m}_{11} - \dot{m}_{21}}$$

- Outdoor Air Correction Factor (OACF) *3:

$$OACF = \frac{\dot{m}_{2,1}}{\dot{m}_{2,2}}$$

*3: these two parameters quantify the internal and external leakages

The following test specifications are laid out by the standard:

- Tests performed:

  - rotor placed in the airstream as specified by the manufacturer;
  - at rotor speed specified by manufacturer.

In order for the tests to be valid, they must meet all the requirements outlined by “Testing tolerance of AHRI standard 1060”.

- External Leakage, Internal exhaust air:

The tests for air leakage are evaluated with the calculation of EART and OACF in three different conditions.

Pressure differential : Leaving supply airflow stati pressure minus entering exhau: (return) airflow static pressure	$\Delta p \Big|_{11}^{22}$	Test 1: 0 mm$H_2$0 Test 2 and 3: two or more of the following: -5.0, -3.0, -1.0, -0.50, 0.50, 1.0, 2.0, 3.0, 5. mmH20

- Sensible Efficiency, Latent Efficiency and Pressure Drop tests will follow the guidelines presented in Table 7 (refer to Table 6 for indices definition):

Table 7. Sensible efficiency, latent efficiency, total efficiency and pressure drop test conditions

		Regular Test conditions	
		Heating	Cooling
Inlet suppl airflow 21	T dry-bulb	35 °F (~1.7 °C)	95 °F (~35 °C)
	T wet bulb	33 °F (~0.56 °C)	78 °F (~25.6 °C)
Inlet exhau airflow 11	T dry-bulb	70 °F (~21.1 °C)	75 °F (~23.9 °C)
	T wet bulb	58 °F (~14.4 °C)	63 °F (~17.2 °C)
Outlet suppl airflow 22	v	100% and 75% of the rated Airflow(s)	
Inlet exhau airflow 12	v	Same as “outlet supply airflow 22”	
Pressure differential	$\Delta p\Big\|_{11}^{22}$	0 Pa (for effectiveness tests)	
Rotor speed		Specified by manufacturer	

The performance specifications for products certified by the AHRI can be found on the ARI Certified Component website at the link below:

- http://www.ahridirectory.org

The results can be divided into two parts:

- Internal and external leakage (EART and OACF): The three tests are done at different pressure differentials between the leaving supply airflow static pressure and the entering exhaust airflow static pressure.

- The effectiveness and the net effectiveness of the sensible, latent and total heat transfer in both heating and cooling modes for each of the two required air flow rates.

COMMENTS ON THE AFOREMENTIONED STANDARDS

The main drawback of the aforementioned standards is that they do not take a heater device before the desiccant wheel on the return air side into account. This means the influence of regeneration temperature is not considered.

Further comments specific to each standard are presented below.

EUROVENT RS 8/C/002-2014

The definition of “latent efficiency” given does not lead to the calculation of “efficiency” since the denominator doesn’t represent the ideal maximum dehumidification effect. Therefore, this figure could be higher than one, violating the thermodynamic definition of "efficiency".

Another issue is that the given definition of "sorption rotor" is questionable (see the sample calculation presented in Appendix 5.2. Moreover, if $x_{11} = x_{21}$ (as occurred in the case of Milan's demonstration plant, see chapter 6.3), the formula is not valid since the denominator would be zero.

Another restriction is that the supply airflow must equal the exhaust airflow. This is not a typical operating condition in many systems

ANSI/AHRI standard 1060

The given definition for latent and total effectiveness is not useful for a solar DEC design since the denominator of the two equations does not represent the driving force of the heat and mass transfer operations.

4.2. STANDARDS FOR ACTIVE DEHUMIDIFICATION WHEELS

The previous section reviewed regenerative/rotary heat exchanger certification programs as laid out in two different standards. In this section, more specific standards (ARI 940/98 and ANSI/ASHRAE 139-2007) for latent energy exchangers are discussed. The return air in these systems is pre-heated before going into the wheel, which represents an “active dehumidification wheel”.

ARI 940/98 DESICCANT DEHUMIDIFICATION COMPONENTS

The purpose of this standard is to establish definitions, classification, testing requirements, performance rating, minimum data requirements, nameplate data and conformance conditions for thermally regenerated dynamic desiccant dehumidifiers.

The main outcomes of the standard were (see 10.1.1 for variables definition):

- The definition of the Moisture Removal Capacity (MRC) in kg/hr (lb/hr):

$$\mathrm{MRC} = \mathrm{V_s} \cdot \frac{1}{\mathrm{v_s}} \cdot \frac{\mathrm{w_{inlet}} - \mathrm{w_{outlet}}}{1000\ {}^{\mathrm{g}}/_{\mathrm{kg}}} \qquad \text{Eq. [4]}$$

$$\mathrm{MRC} = \mathrm{V_s} \cdot \frac{1}{\mathrm{v_s}} \cdot \frac{\mathrm{w_{inlet}} - \mathrm{w_{outlet}}}{7000\ {}^{\mathrm{grains}}/_{\mathrm{lb}}}$$

- The guideline specifying that standard ratings shall be determined at the standard rating conditions specified in Table 8, using a minimum of three of the four listed conditions.

Table 8: Standard rating conditions for ARI 940/98

Condition number	Process air inlet conditions		Regeneration air inlet conditions	
	Dry bulb**	Wet bulb**	Dry bulb**	Wet bulb**
1	95 °F (35 °C)	75 °F (23.9 °C)	95 °F (35 °C)	75 °F (23.9 °C)
2	80 °F (26.7 °C)	75 °F (23.9 °C)	80 °F (26.7 °C)	75 °F (23.9 °C)
3	80 °F (26.7 °C)	67 °F (19.4 °C)	95 °F (35 °C)	75 °F (23.9 °C)
4	45 °F (7.2 °C)	45 °F (7.2 °C)	80 °F (26.7 °C)	75 °F (23.9 °C)

*All tests are run at atmospheric pressure.

**The tolerance for all temperatures during the test is ±0.5 °F (±0.3 °C).

ANSI/ASHRAE 139-2007

The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) has published the standard "ANSI/ASHRAE 139-2007: Method of Testing for Rating Desiccant Dehumidifiers Utilizing Heat for the Regeneration Process", which has been approved as an American National Standard[6] (ANS) by ANSI.

The purpose of this standard is to provide test methods for determining the moisture removal capacity of heat-regenerated desiccant dehumidifiers as well as their thermal energy performance. This was done with the intention of enabling comparisons of capacity and performance between models.

This standard can be applied to dehumidification devices using either liquid or solid desiccants that are operated at atmospheric pressure and are regenerated by thermal energy.

The following the main three performance metrics are defined by this rating standard:

- Moisture Removal Capacity (MRC): the definition is equal to Eq. [4] (kg/hr or lb/hr);
- Regeneration Energy (RE): the amount of heat used to regenerate the desiccant (measured at the power terminals or by the amount of fuel supplied to the heating device) (W or Btu/hr);

[6] Is a national voluntary consensus standard developed under the auspices of (ASHRAE). Consensus is defined by the American National Standards Institute (ANSI) as "substantial agreement reached by directly and materially affected interest categories. This signifies the concurrence of more than a simple majority, but not necessarily unanimity. Consensus requires that all views and objections be considered, and that an effort be made toward their resolution." Compliance with this standard is voluntary until and unless a legal jurisdiction makes compliance mandatory through legislation.

- Regeneration Specific Heat Input (RSHI): defined as the ratio of the regeneration energy to the moisture removal capacity [kJ/kg or Btu/lb].

$$RSHI = \frac{RE}{MRC} \cdot 3.6 \quad [\frac{kJ}{kg}]$$

$$RSHI = \frac{RE}{MRC} \quad [\frac{Btu}{lb}]$$

COMMENTS ON THE AFOREMENTIONED STANDARDS

For both standards (ARI 940/98 and ANSI/ASHRAE 139-2007) there are few proposed rating conditions and a lack of references for the given conditions.

In general, the operating variables that affect the dehumidifiers' performance are the following:

- Process air moisture;
- Process air temperature;
- Process air velocity through the desiccant;
- Reactivation air temperature;
- Reactivation air moisture;
- Reactivation air velocity through the desiccant;
- Amount of desiccant presented to the reactivation and process airstreams;
- Desiccant sorption-desorption characteristics.

The relationships between the operational variables and the system's performance are investigated in [6].

The test guide NREL TP-550-26131 [7], suggests additional performance metrics divided into two categories; one for wheel designers and one for application engineers. Short descriptions of these metrics are listed below. To see the detailed descriptions and formulas, please refer to [7].

- NTUmass: NTU for mass transfer, defined by the heat transfer analogy;
- Grain depression per unit of residence time: this figure is defined with the aim of normalizing tests for differing face velocities, open areas, and wheel depths;
- RSHIHX: Regeneration specific heat input, including the effects of a heat exchanger on the process air outlet;
- RSHD: Regeneration specific heat drop, is an indicator of the wheel's energy consumption. This value is less sensitive to mass-flow ratio than RSHI. High RSHD may indicate poor grain depression or it may show that the wheel is able to utilize lower temperature air for regeneration, or that the matrix is retaining heat;
- AHR: Adsorption heat ratio is a method to quantify the heat dump-back (transfer of heat from regeneration air to process air). If AHR is equal to 1 the process is adiabatic.

Fractional AHR indicates the degree of heat dump-back;

- MRCBtuh: MRC expressed as a cooling rate (normalized by volumetric flow rate);
- COPlatent: latent coefficient of performance;
- MRC/fan power: a measure of the ratio between mass transfer and pressure drop.

The uncertainty calculation is another source of concern. ANSI/ASHRAE 139-2007 sets limits on uncertainty for instrumentation, but it does not discuss the total combined uncertainty of MRC and RSHI. The described requirement that the moisture mass balance fall within 5% of 1.0 must not be taken as the accuracy of these calculated results. A calculation procedure is proposed in [7] ("Total Combined Uncertainty" chapter), it consists of an evaluation of the total combined uncertainty for a given figure of merit, considering both random and bias uncertainties.

The desiccant wheel performance evaluation should also be completed with some consideration for the outlet air quality as many desiccant materials can collect common indoor pollutants during the sorption phase. In [8] definitions exist for a classification of the pollutant sorption reactions. This includes humidity-neutral sorption, humidity-reduced sorption, humidity-enhanced sorption, humidity-pollutant displacement, and desiccant-catalyzed pollutant conversion reactions.

4.3. MANUFACTURERS' TECHNICAL DATA FOR DESICCANT WHEELS

Desiccant wheel manufacturers typically do not provide a latent efficiency or effectiveness. Instead, they provide one of the following:

- Moisture Removal Capacity - see Eq. [4] (also referred to as the Nominal Capacity);
- Process air outlet conditions presented in performance maps or tailored software.

No manufacturer refers to a certification label or specific standard for the testing of their active desiccant wheel's performance. Thus, the description of the test conditions used for the assessment is crucial. The following factors should be reported:

1. for both the process air and the regeneration air:
    a. the temperature;
    b. the relative humidity;
    c. the air velocity;
2. the use of or lack of use of a purge sector;
3. the ratio between the process area and regeneration area;
4. The rotation velocity.

Refer to [6] for a detailed discussion about the influence of all variables on wheel performance.

A final concern over the manufacturer provided information is that most of the time, the performance ratings are presented for standard conditions, which are not suitable for solar powered regeneration. It would be interesting if the manufactures could present the performance under varying values of the key test parameters (i.e., the regeneration temperature, air velocity, rotation rotor speed, etc.). Some manufacturers provide software which characterizes the performance of the wheel at specific conditions but they still recommend that the designer contact them to select the most suitable wheel.

4.4. DESICCANT-BASED DEHUMIDIFICATION EQUIPMENT

This section presents a test methodology for desiccant-based dehumidification equipment. In order to make DEC systems more common in typical HVAC systems and to help the development of this technology, it is important to provide performance information in a form that would be familiar to a HVAC design engineer. Moreover, the testing procedures for desiccant dehumidifiers should be similar to those which have been used for decades in the testing of conventional unitary air-conditioning products.

ANSI/ASHRAE 174-2009: METHOD OF TEST FOR RATING DESICCANT-BASED DEHUMIDIFICATION EQUIPMENT

ASHRAE has published a standard for desiccant-based dehumidification equipment, called “ANSI/ASHRAE 174-2009”. The purpose of this document is to provide test methods for rating the performance of desiccant-based dehumidification equipment operating at atmospheric pressure. The standard takes all the energy transfer devices used by the desiccant wheel system into account.

An ideal companion document to this ASHRAE method-of-test standard would be a rating standard prepared by an appropriate trade association, which specifies standard rating conditions for desiccant-based dehumidifier products. This rating standard should provide definitions and regulations for the following metrics:

1. Total cooling capacity;
2. Sensible cooling capacity;
3. Latent cooling capacity;
4. Equipment energy consumption:
    a. Chilled water energy input flow;
    b. Hot air energy input flow;
    c. Hot water energy input flow;
    d. Dry steam energy input flow;
    e. Evaporation or spray cooling water energy;
    f. Gas energy input flow;
    g. Electrical energy input flow;
    h. Separated air flow energy (situated to remove heat from the unit);
    i. Condensate water flow rate and energy.
5. Total coefficient of performance;

6. Sensible coefficient of performance;
7. Latent coefficient of performance.

LIMITATIONS OF THE TEST CONDITIONS PROPOSED

The test conditions proposed in ANSI/ASHRAE 174-2009 do not consider the humidification process and how the water and energy consumption used for indirect humidification could differ from one system to another. Thus it is important to specify the required water quality for the test to make it independent of the water hardness.

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[3] ARI 940/98 Desiccant Dehumidification Components

[4] ANSI/ASHRAE 139-2007: Method of Testing for Rating Desiccant Dehumidifiers Utilizing Heat for the Regeneration Process

[5] ANSI/ASHRAE 174-2009: Method of Test for Rating Desiccant-Based Dehumidification Equipment

[6] The Dehumidification Handbook, Munters Corporation, 2002. Chapter 6, desiccant dehumidifier performance.

[7] S.J. Slayzak, J.P. Ryan, NREL/TP-550-26131: Desiccant Dehumidification Wheel Test Guide, National Renewable Energy Laboratory, 2000.

[8] ASHRAE Handbook, Sorbents and Desiccants (chapter 32), 2013.

5. GOOD PRACTICE SDEC SYSTEMS

5.1. ENERGYBASE SDEC IN VIENNA, AUSTRIA

Author: Selke T.

5.1.1. BUILDING AND SDEC SYSTEM FACTS

Figure 31: The ENERGYbase office building in Vienna (©Hertha Hurnaus)

The ENERGYbase office building, (owned by the Vienna Business Agency[7] and located in the 21st district of Vienna) has been in operation since 2008. This "Future Building" was built to demonstrate cutting-edge office designs, energy efficient technology and the use of renewable energy gained from on-site resources with the goal of stimulating further use and development of these techniques.

In conventional office buildings in Vienna, heat is generated in central-heating facilities by burning fossil fuels or the building is connected to the public district heating network. Heat is typically distributed via hot water and radiators or by ducted central air systems. Vapor-compression chillers are normally used for cooling and typically circulated air is the cooling medium. Providing fresh air during the summer and winter seasons is difficult for these typical systems given that opening windows increases energy consumption at these times of year.

[7] https://viennabusinessagency.at/

ENERGYbase was designed using the 'Passivhaus[8]' architectural concept. The three energy system targets were: a) highly energy efficient applied systems; b) maximize use of renewable and environmental friendly energy sources and; c) provide comfortable conditions for all occupants. The designers achieved these goals through the use of environmentally friendly materials, high quality insulation, a well-sealed building envelope and innovative energy technologies (e.g., photovoltaics, solar-assisted air-conditioning, and advanced heat pump technology) as well as innovative control systems to provide a high quality of indoor comfort. For example, the photovoltaic system design aimed supply 20% of the total electricity required for lighting, heating, cooling and air treatment. These technologies and techniques reduced the energy consumed for heating, cooling and artificial lighting of ENERGYbase by 80% compared to a standard office building at that time. The annual electricity demand of the building was calculated in the design phase with 25 kWh_{el} per year and useful office area (7,500 m²)

Building Facts

Type of building	Office
Location	A- 1210 Vienna
In operation since	2008
System operated by	Siemens Facility Management
Useful area	7,500 m²
Air-conditioned area	5,000 m²
Solar system used for space heating	Yes
Solar system used for DHW	No
Additional Information	
Renewables	
Use of Solar energy	Photovoltaics 45 kW_{peak} solar power Flat-plate thermal collector system with approx. 285 m²
Use of shallow geothermal energy	Ground water coupled heat pump
Other innovation	Green ventilation (i.e., biological supply air treatment in wintertime for pre-humidification and filtering)
	Thermal mass activation for sensible heating & cooling

SDEC System Facts

Desiccant cooling units

Technology	Desiccant and evaporative cooling
N° of DEC systems	2 DEC system (twins)
Nominal volumetric flow rate	2 x 8,240 m³/hr
Nominal capacity	~ 40 kW per DEC system
Brand of cooling units	robatherm
Cooling load subsystem	Central AHU

[8] http://passivehouse.com/02_informations/01_whatisapassivehouse/01_whatisapassivehouse.html

Dehumidification	Sorption wheel (Klingenburg SECO 1770)
Regeneration power	80 kW per DEC system

Solar thermal collector fields

Collector type	Flat-plate collectors
Brand of collector	Sonnenkraft /MEA DESIGN
Collector area	285 m²
Tilt angle, orientation	32° South
Collector fluid	Water-Propylene glycol 70/30
Typical operation temperature	80°C
Heating load subsystem	Concrete core activation, central AHU

Heat back-up system

Unit #1

Technology	Electrical heating coil
Nominal heating capacity	80 kW
Heating load subsystem	Central air handling unit

Unit #2

Technology	Heat pumps
Nominal heating capacity	2x160 kW_{th}
Heating load subsystem	Concrete core activation

Heat storage system

Number of units	1
Technology	Solar hot water tank
Storage capacity	15,000 l
Shared with heat back-up systems	No
Type of connection	-

Climate

Located	48°12' N/ 16°22' E
T_{mean} (T_{max} / T_{min} hourly)	9.5 °C (28.9 / -14.6°C)
Global radiation on horizontal	1 122 kWh/m² · year
Global diffuse on horizontal	627 kWh/m² · year
Global direct on horizontal	495 h/m² · year

5.1.2. DESIGN PHASE

SDEC Design Principles

The Austrian Institute of Technology (AIT), Austria's largest non-university research institute, proposed the implementation of a solar-assisted air-conditioning system which used desiccant evaporative cooling technology to the meeting the low energy consumption targets of ENERGYbase. The SDEC system was designed to use solar thermal power as the only input heat during the summer season. The HVAC systems of the ENERGYbase building was designed to use both water and air based energy distribution systems. The office area temperature was controlled by the water based heating and cooling system (i.e., concrete core activation). The air treatment system only met the latent cooling load and was used control the humidity and to supply fresh air to the offices. This reduces the energy consumption required to maintain comfortable conditions. Design principles for the SDEC System

Solar heat regeneration	100% of the regeneration heat is supplied via solar thermal energy during the summer
Sorption wheel	Used for dehumidification (summer operation) and humidity recovery (winter operation)
Purpose of air handling unit	Control of the latent loads of the office building
Design outdoor air conditions(Summer)	35°C and 15 $g/kg_{dry\ air}$
Design set point of supply air	Moderate set values (dry bulb temperature / humidity ratio): 23°C / 8 g per $kg_{dry\ air}$
Design volumetric air flow rate	Operate with a low constant flow rate (due to hygienic requirements) and to maintain an air change rate below 1 in the office areas
DEC configuration	Standard desiccant cooling system (slightly adapted to make use of biological humidification during winter)
DEC system provider	One provider delivers the complete package (including control strategy)

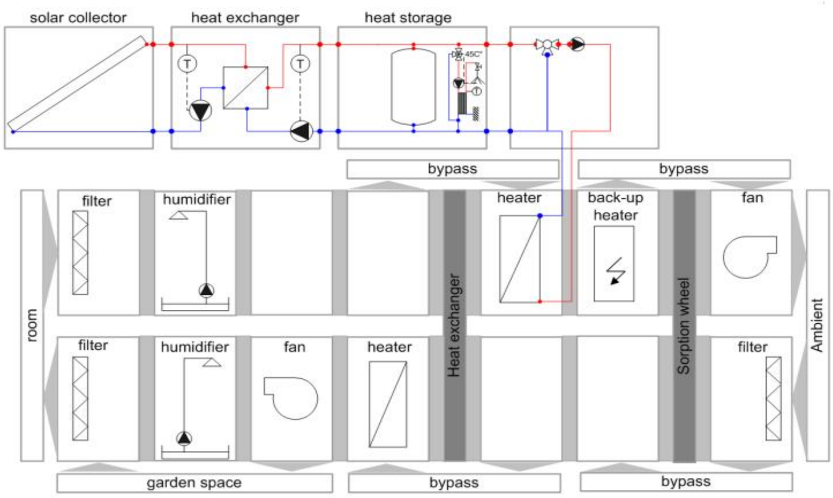

Figure 32: Simplified hydraulic scheme of the ENERGYbase SDEC System (the electrical back-up heater was used for emergency cases only)

Integral Planning Support

The complexity of planning and integrating non-standard technologies and systems into a building project led to significant challenges. Therefore, an integrated planning approach, involving scientific energy experts, has been used to ensure the ambitious ENERGYbase project targets were met. AIT, as a center of excellence for sustainable buildings and energy systems, has scientifically supported the planning team. AIT did the essential design and energy performance investigation for the SDEC system. The DEC technology's dynamic performance required simulations of the energy performance of the building and the SDEC system to optimize the system and ensure that the required performance targets could be met.

First Calculation

A high solar fraction was required to maximize the energy savings of the SDEC system. This minimized the amount of thermal energy from non-renewable sources needed to regenerate the sorption wheel and to heat the supply air during the winter. AIT used the transient system simulation software TRNSYS[9] (version 16.1) to predict the system's performance including the solar fraction. Standard TRNSYS types were used. A TRNSYS type is a sub-model that represents a system component. Many different types can be linked together to model an entire system. Slight modifications to some built-in TRNSYS types were required to model the SDEC system's control strategy.

The simulations were conducted using weather data for Vienna (as determined by the METEONORM[10] computer software package). The solar system model included 285 m^2 of gross flat-plate collector area coupled to two hot water storage tanks (7,400 liters each). The flat-plate collector was modeled using product performance specifications and was oriented to the south with a slope of 32 degrees. The solar system operated using a low specific flow rate of 20 $kg/hr \cdot m^2$. A five minute time step was used for the simulation. Figure 33 displays the simulation results of one summer week.

[9] http://www.trnsys.com/

[10] http://meteonorm.com/en/

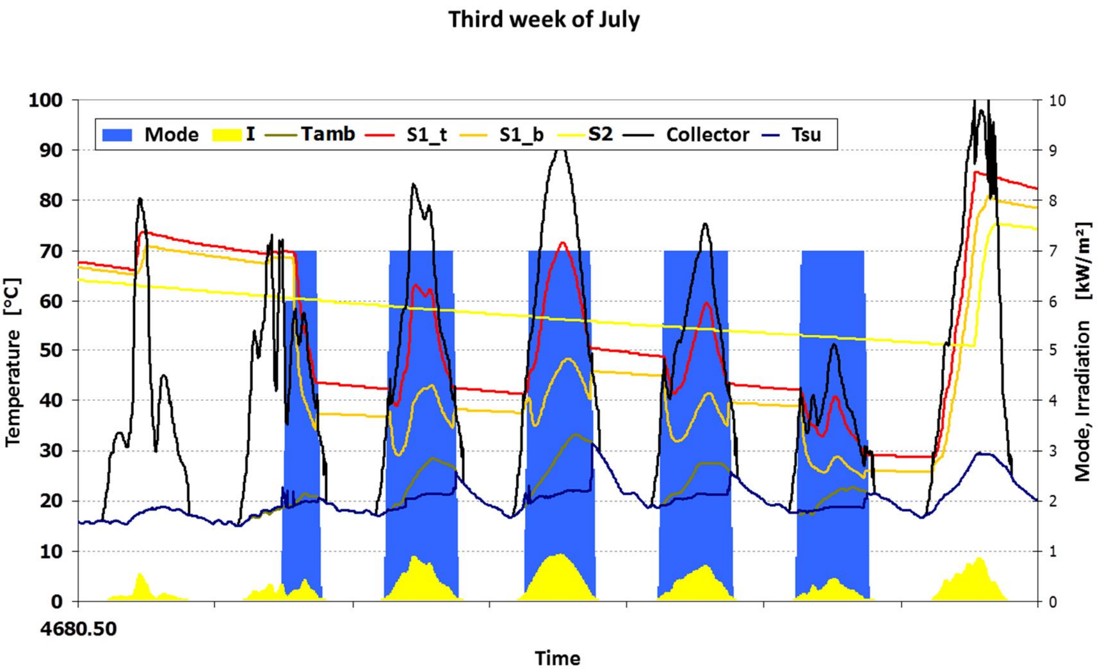

Figure 33: TRNSYS simulation results – DEC system performance over one summer week showing mode, solar radiation on collector surface (I), ambient temperature (T_{amb}), top of storage tank temperature (S1_t), bottom of storage tank temperature (S1_b), collector outlet temperature (Collector) and the supply air temperature (T_{su}), S2 not specified.

Key Findings

The following key findings were identified during the analysis of the simulation's results:

- During the selected summer week (Figure 33), the SDEC system operated mainly in desiccant evaporative cooling mode (i.e., dehumidification by means of the sorption wheel, evaporative cooling by means of the return and supply air humidifier, and energy recovery by means of the heat recovery wheel);
- On sunny days, the flat-plate collector system generated sufficient temperatures to load the hot water storage tank (T_{coll} < 90°C) ;
- On sunny days, the hot water storage system provided sufficient temperatures of approximately 70°C; temperature stratification in the storage tank was also achieved;
- The DEC system provided supply air temperature in the range of 23°C, even when the ambient temperature exceeded 30°C;
- It was possible to operate the ENERGYbase DEC system completely via solar energy while still providing comfortable conditions when the temperature control for the building was achieved using a separate water based system).

5.1.3. OPERATIONAL PHASE

MONITORING DATA AND SDEC SYSTEM PERFORMANCE EVALUATION

AIT designed an energy monitoring system for the assessment of the SDEC unit's energy performance. The SDEC system was equipped with measurement devices following the 3rd level evaluation specifications according to the IEA SHC Task 38 Monitoring Procedure for Solar Cooling Systems[11]. Figure 34 displays the ENERGYbase SDEC monitoring system. The energy performance evaluation was based on data from 2010. Several selected key performance indicators were calculated including the collector field efficiency, the thermal seasonal energy efficiency ratio for both cooling and heating mode ($SEER_{thermal,\ heating\ and\ cooling}$) and the electric seasonal energy efficiency ratio ($SEER_{electric}$). Table 9 and Table 10 list the relevant monthly energy data and key performance indicator over different time frames.

During 2010, the operation of the SDEC systems consumed around 49.9 MWh of electricity. This included power supplied to the DEC system, the solar system and the ground water coupled heat pump used to post heat supply air in winter. In total, approximately 350 MWh of solar radiation incident was provided by the sun to the 285 m² of the flat-plate collector system. The solar system transferred around 70.9 MWh_{heat} to the hot water tank. This corresponds to an annual collector field efficiency of 20%. 507 kWh of electricity was consumed to operate the solar thermal system.

From June to August 2010 the ENERGYbase SDEC system achieved a thermal energy efficiency ratio (EER_{th}) in the range of 0.50 to 0.54. The total electric seasonal energy efficiency ratio $SEER_{electric}$ was determined to be 5.05.

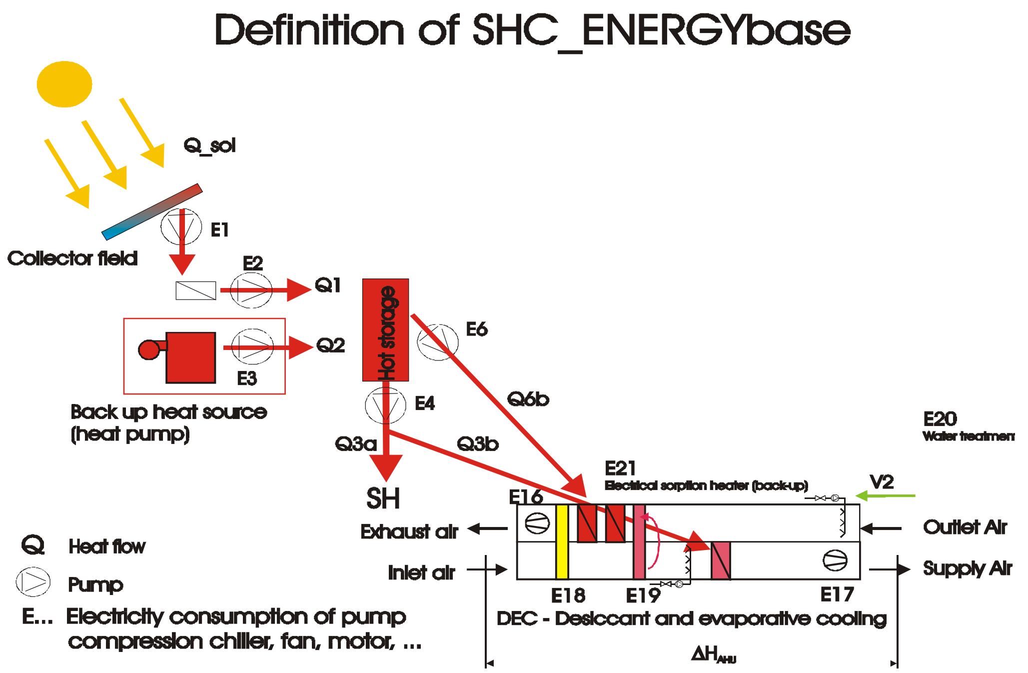

Figure 34: The SDEC performance monitoring system created according to the SHC TASK38 Monitoring Procedure

11 http://task38.iea-shc.org/data/sites/1/publications/IEA-Task38-Report_A3a-B3b-final.pdf

Table 9: ENERGYbase energy performance during the summer months of 2010

			total	Apr	May	Jun	Jul	Aug	Sep
A	Solar irradiation on total collector apertur area	kWh	243.512	41.193	33.431	44.684	48.802	41.069	34.333
B	Solar heat delivered Q1	kWh	46.143	6.864	3.981	8.654	14.348	8.516	3.780
C	Heat delivered to regeneration coil Q6b	kWh	31.803	316	1.220	7.990	12.994	7.924	1.359
D	Heat delivered to heating coil Q3b	kWh	1.096	1.075	-	-	-	-	21
E	Heat delivered from heat pump Q2,	kWh	-	n.m.	n.m.	n.m.	n.m.	n.m.	n.m.
F	Delta Enthalpie heating ΔHAHU	kWh	26.510	18.168	3.340	438	331	406	3.828
G	Delta Enthalpie cooling ΔHAHU		15.890	185	339	3.971	6.979	4.279	136
H	Electricity DEC, E16 – E19	kWh	15.074	3.066	2.275	2.456	2.736	2.491	2.050
I	Electricity solar, E1 + E2	kWh	252	63	28	39	60	38	24
J	Electricity heat pump E3 (calculated)	kWh	343	336	-	-	-	-	7
K	Electricity water preparation (calculated)	kWh	510	85	85	85	85	85	85
L	Total electricity	kWh	16.179	3.550	2.388	2.580	2.881	2.614	2.166
	Collector field efficiency A/B	[-]	0,202	0,17	0,12	0,19	0,29	0,21	0,11
	SEER DEC thermal (Cooling) G/C	[-]	0,50	0,59	0,28	0,50	0,54	0,54	0,10
	SEER DEC thermal (Heating) F/D	[-]	4,82	16,90					182,27
	SEER DEC electrical (F+G)/L	[-]	5,05	5,17	1,54	1,71	2,54	1,79	1,83

Table 10: ENERGYbase energy performance during the winter months of 2010

			total	Jan	Feb	Mrz	Oct	Nov	Dec
A	Solar irradiation on total collector apertur area	kWh	106.788	9.384	14.736	33.126	25.767	12.984	10.791
B	Solar heat delivered Q1	kWh	24.772	1.261	3.862	8.752	6.617	2.936	1.344
C	Heat delivered to regeneration coil Q6b	kWh	65	-	-	37	-	26	2
D	Heat delivered to heating coil Q3b	kWh	47.766	12.006	12.298	6.399	1.031	3.717	12.315
E	Heat delivered from heat pump Q2,	kWh	-	n.m.	n.m.	n.m.	n.m.	n.m.	n.m.
F	Delta Enthalpie heating ΔHAHU	kWh	209.153	47.583	55.512	43.211	17.657	18.626	26.564
G	Delta Enthalpie cooling ΔHAHU		196	-	-	196	-	-	-
H	Electricity DEC, E16 – E19	kWh	18.029	2.684	3.444	3.572	2.094	2.717	3.518
I	Electricity solar, E1 + E2	kWh	255	21	41	81	61	35	16
J	Electricity heat pump E3 (calculated)	kWh	14.927	3.752	3.843	2.000	322	1.162	3.848
K	Electricity water preparation (calculated)	kWh	510	85	85	85	85	85	85
L	Total electricity	kWh	33.721	6.542	7.413	5.738	2.562	3.999	7.467
	Collector field efficiency A/B	[-]	0,202	0,13	0,26	0,26	0,26	0,23	0,12
	SEER DEC thermal (Cooling) G/C	[-]	0,50			5,30			
	SEER DEC thermal (Heating) F/D	[-]	4,82	3,96	4,51	6,75	17,13	5,01	2,16
	SEER DEC electrical (F+G)/L	[-]	5,05	7,27	7,49	7,57	6,89	4,66	3,56

Figure 35 illustrates the monthly primary energy demand of both the SDEC system and a reference system. In total the SDEC system operated with a primary energy demand of 124.75 MWh_{PE} (using a primary conversation factor for the European electricity grid of 2.5). The reference air treatment system was designed according to the definitions of IEA SHC Task 38 monitoring procedure and the energy performance data was calculated via a TRNSYS simulation.

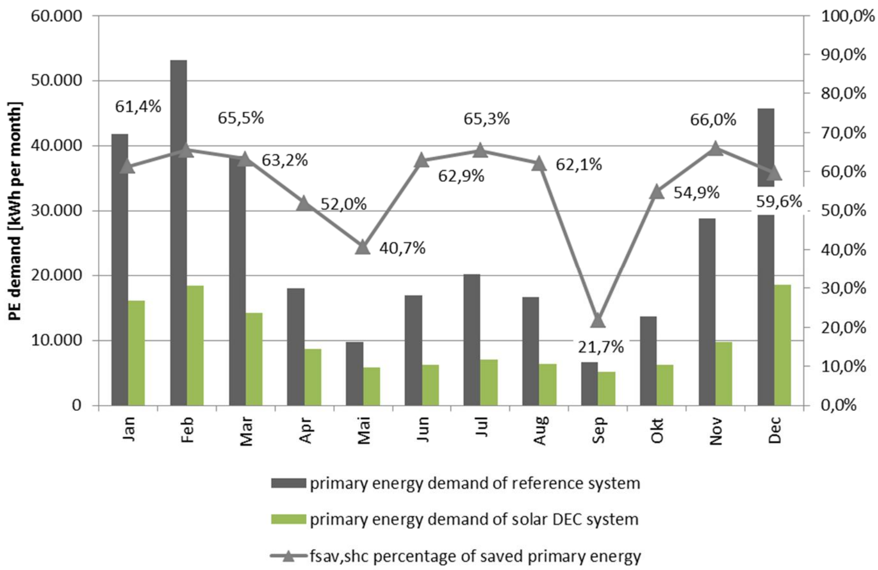

Figure 35: Primary energy demand of the solar DEC system in 2010 versus a reference system

Figure 35 shows the potential for primary energy savings was the highest in winter, where primary energy savings of 138.5 MWh_{PE} in the months of January to March and October to December were seen. In the summer months (June, July, August) primary energy savings of 34.1 MWh_{PE} were achieved. The transition months (April, May and September) had the lowest fraction of energy improvements with savings of only 14.8 MWh_{PE}. This clearly shows that the DEC system has a high potential for primary energy savings when there is a high demand for heating and humidification (winter) and cooling demand (summer). The potential for primary energy savings was low when the outdoor air conditions were similar to the desired indoor conditions.

In addition to the IEA SHC Task 38 monitoring procedure, the individual components of the SDEC system were studied for a more in-depth analysis. Figure 36 illustrates the total operating hours of the SDEC system and the operational hours of the following components and run modes:

- sorption wheel in enthalpy recovery mode (winter);
- sorption wheel in dehumidification mode (summer);
- heat recovery wheel in heating mode;
- heat recovery wheel in cooling mode;
- return air humidification (adiabatic cooling);
- supply air humidification in summer (adiabatic cooling).

The sorption wheel was mostly operated as an ’enthalpy rotor’ for temperature and humidity recovery (87.5% of the operating hours). For 335 of the operating hours, the sorption wheel functioned as a dehumidification device (12.0% of the operating hours). In heat recovery mode, the SDEC system controller activated the sorption wheel (to act as an enthalpy exchanging device in order to recover temperature and humidity and to protect against icing in winter time). The heat recovery rotor would then be activated if the supply air set-points were not met by the sorption wheel. The sorption wheel operated for 2,433 hours to recover sensible and latent energy while the sensible heat recovery rotor operated for only 1,297 hours of this time.

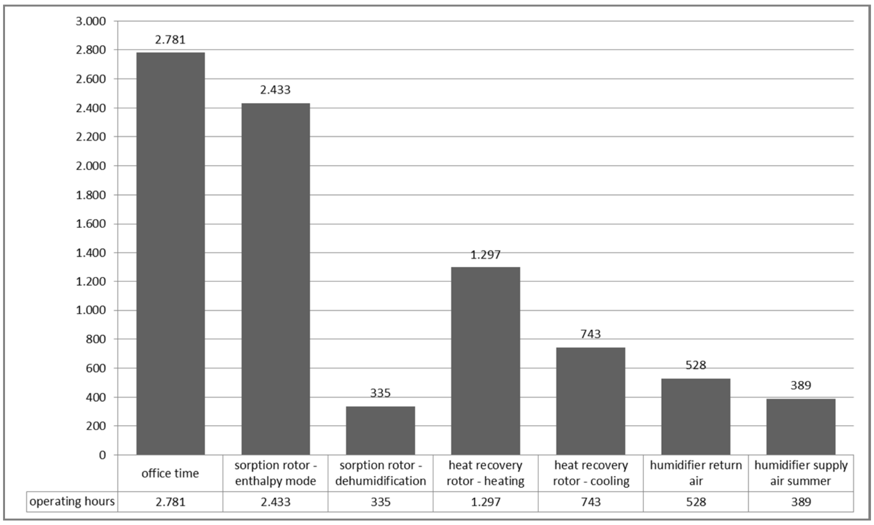

Figure 36: Operational run times of various SDEC components in 2010

FACILITY MANAGEMENT STATEMENTS

The facility manager of the ENERGYbase office building cooperates with AIT on several projects. This active relationship between the researcher and the operator benefits both parties. During the first years of operation the facility manager tested the capacity of the SDEC system, its control strategies and its general operation. During an interview with the facility management, the following statements related to SDEC technology were noted:

General:

- The SDEC system operates like any other ventilation system with regards to the basic functions of heating, cooling, dehumidification, humidification and supply of fresh air.
- The SDEC system provides more options for different operation modes due to the large number independently operated components. This could be exploited to maximize comfort and energy savings, but to achieve this, the general functioning of the SDEC system should be well understood by the facility manager.

Benefits:

- A more frost secure performance in the switch-on operation phase is recognized because the DEC system is installed on the roof top within a thermally high quality building shell. For instance no flushing of heating coils in the starting mode is necessary.

- The energy efficient performance observed during the winter time allowed the SDEC system to operate without back-up heat when the ambient temperatures were above 5ºC (i.e., the heating system can be switched off during the shoulder season).
- The DEC system is capable of supplying massive amounts of humidification using only a minimal amount of thermal energy, it is also capable of dehumidifying the ambient air up to 6 g/kg$_{dry\ air}$, which is sufficient for the Vienna climate.
- Regeneration temperatures in the range of 50ºC to 60ºC are sufficient to achieve the comfortable conditions

Disadvantages:

- Slightly higher maintenance effort is required, when compared to a conventional system, because of the additional number of components including the second wheel and the two nozzle-sprayer devices.
- The control of such a SDEC system is complex due to the numerous operational modes. The facility manager must be well trained to ensure the best operation of the system.

CONTROL STRATEGY

The main task of the air treatment system was to provide fresh air and to meet the latent cooling load of the building. The controlled target parameter of the SDEC system was the supply air temperature, which was set at 21°C in heating mode and 22°C in cooling mode. Different operational modes were then sequentially activated according to the building's load. Priority was assigned from the lowest to the highest energy consuming mode. During cooling mode, the control sequence was set as follows:

- ventilation: variable speed fans were controlled according to the static pressure difference between the supply and return air streams, all other equipment was turned off and the bypasses around the rotary heat exchanger and the sorption wheel were open;
- heat recovery: when the return temperature was lower than the outdoor temperature, the bypass around the rotary heat exchanger was closed and the rotary heat exchanger was activated (the speed was PI controlled up to 10 rounds per minute);
- return air evaporative cooling: if the return temperature was higher than the outdoor temperature, the return humidifier was turned on (the frequency of the pump was PI controlled between 20 and 50 Hz);
- solar heat regeneration: the bypass around the sorption rotor was closed, the wheel was activated (the speed was set at 20 rounds per hour) and the two-way valve on the hot water return, from the regeneration coil, was opened PI controlled, (the regeneration air temperature was limited to 70°C);
- supply air evaporative cooling: the supply humidifier was turned on, the frequency of the pump inverter was PI controlled between 20 and 50 Hz and the relative humidity, at the

humidifier outlet, was limited to 80%;

- auxiliary heater regeneration (emergency case only): in an emergency case, the electrical heater was manually turned on if the solar collectors are not providing sufficient heat and no heat was available from the hot water buffer storage (the electrical heater was PI controlled and the regeneration air temperature was limited to 70°C).

Improvements / Optimization Measures

In cooperation with the facility manager, AIT developed an improved overall control strategy for the SDEC system. A new control parameter, the absolute humidity of the supply air, was used instead of the temperature. The goal of this was to increase indoor comfort and reduce energy consumption.

In 2013, the controller hardware was replaced with a new system which enabled the facility manager to program the new control strategy into the system. Keeping in mind that the room temperature was controlled all year round by the active concrete ceiling system, the SDEC system was able to be responsible for only the indoor air humidity. This meant that the supply air temperature of the SDEC system varied within a certain range. The implementation of the new control strategy was expected to provide the following benefits:

- Winter operation: Higher indoor air humidity values will be achieved combined with the drawback of lower supply air temperatures;
- Summer operation: Lower room air humidity values will be achieved with the drawback of higher supply air temperatures.

The target supply air humidity was set to approximately 9 g/kg/hr dry air (with some flexibility due to the dead band range built into the controller).

An additional benefit of the updated control strategy is the capacity to use solar heat to provide frost protection to the lithium chloride sorption wheel during the wintertime.

The new control system has been implemented and its performance is being monitored but, unfortunately, an updated analysis of the performance has not been completed in time to publish this report.

5.1.4. KEY FINDINGS / LESSONS LEARNED / RECOMMENDATIONS

The two solar heat driven DEC units - each with a nominal air flow of 8.240 m³/hr - installed on the roof top of the ENERGYbase office building have been operated since August 2008. They have been controlled, observed and monitored by the facility manager as well as researchers from AIT. The SDEC system and the accompanying concrete core activation fulfil all of the required services of an air-conditioning system (i.e., temperature and humidity control and providing high quality fresh air). The entire project, starting from the initial decisions and design concepts to the analysis of the recorded data has been achieved with help from researchers from AIT. The constant supervision by experts at all stages of development and operation was one of the factors contributing to the success of this building and promoted the transfer of knowledge and experience gained from research activities into practical applications.

Ten years of technical and scientific research into SDEC system design and operation has led to the development of the following principles to aid in the design of Good Practice systems:

- The design of the SDEC system and solar system was based on regenerating the sorption material using just solar heat with no thermal back-up and no use compression chiller backup systems;
- The sorption wheel should be used for both dehumidification, during summer operation, and humidity recovery, in winter operation (enthalpy recovery mode);
- The DEC system should remove the latent loads from the building while the sensible cooling of the offices should be provided by other heat extraction technologies (e.g. cooled ceiling, concrete core activation, etc.);
- Use moderate set points for the supply air to minimize energy consumption (i.e., 23°C and 9 $g/kg_{dry\ air}$);
- The design volumetric air flow rate should be based on the minimal hygienic needs of outdoor air of the building (as determined by the number and activities of occupants, space function and building volume), the air flow rate of the SDEC system in the ENERGYbase building led to an air change rate of less than 1;
- DEC systems are complex air treatment units due to the large number of components and their dynamic interactions. A DEC system provider capable of delivering a complete package, including control strategy, supports a higher quality of system's operation;
- Transient system simulations should be used to support the design phase due to the limited design standards that are applicable to this type of system, (simulations are especially helpful in sizing solar systems and analyzing the a tradeoffs between cost effectiveness and technical feasibility);
- An energy monitoring system should be carefully designed to provide the data needed for a system performance assessment.

During the operational phase, the SDEC system performance was continually observed and improved by the facility management based on the acquired monitoring data. Success factors and lessons learned from the operational phase of the SDEC system are stated in the following bullets:

- Continuous cooperation and knowledge exchange between the facility management and researchers on site leads to constant improvements of the systems performance;
- In moderate climates, where the air treatment is dominated by heating of the supply air, the use of the sorption wheel in enthalpy recovery mode provides significant primary energy savings during the winter (in 2010, the SDEC system's operation required approx. 122 MWhPE whereas the primary energy consumption of a reference system, based on the primary energy calculation approach of SHC TASK 38, was calculated to

be 309 MWhPE);

- The monitoring data confirmed that the regeneration process for the SDEC system could be driven completely via solar heat, with the acceptance of variation in the temperature of the supply air.

The overall system, including two DEC systems, a 285 m² solar thermal system, a control system, a water treatment system and the air ducts, cost 400.000 EUR, This correlates to 24 € per m³/hr at a nominal air flow of 16.500 m³/hr. A breakdown of the costs is shown in Figure 37.

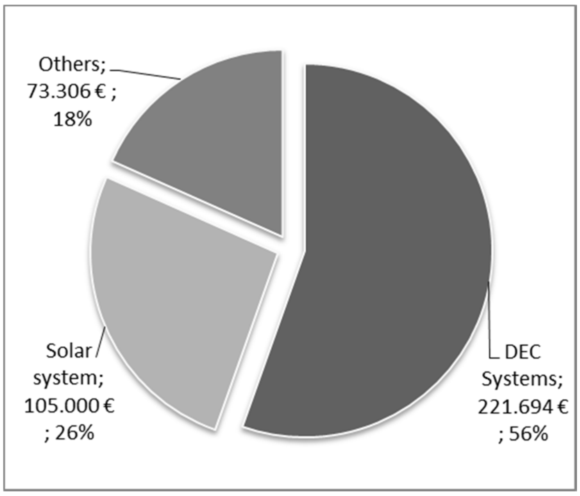

Figure 37: The cost distribution of the SDEC system in Vienna, cost includes VAT

OUTLOOK

The entire design process, operation and performance analysis of the SDEC system was well documented and published in several scientific papers. This was due to the continuous support from energy researchers and the facility management starting from the design phase and continuing all the way to the performance assessment of the SDEC system's operation. This also led to performance improvements, such as the new control strategy previously discussed. The project, on top of providing comfortable conditions with low energy consumption rates, has been a useful tool for spreading the new ideas and technologies featured by the building. For instance AIT offers technical tours through the ENERGYbase office building that highlight the successful operation of the SDEC system.

The large investment in the solar collector system and the short length of time the SDEC system operated in desiccant cooling mode shows further improvement could be possible by focusing on optimizing the solar system sizing and costs to better match the climate experienced in Vienna.

A study of the SDEC system over the course of its entire life cycle (production phase, use phase and disposal phase) is crucial to fully understand the benefits of the system. Therefore, a Life Cycle Analysis (LCA) of the SDEC system in the ENERGYbase office building was carried out[12]. The also LCA considered the impact of photovoltaics (PV) on the SDEC and on the reference system. The key findings of the LCA are stated below:

- Despite the fact that the SDEC system is 160% heavier than the reference system, the reference air-conditioning system requires 59% more global primary energy requirement (GER) and emits an additional 386.9 $CO_{2\ eq}$ during a life time of 25 years;
- The use phase of the SDEC system requires 21 times more GER than the production and the disposal phase combined;
- The use of PV panels improved the ecological performance of both air-conditioning systems but, even though the GER and GWP values were improved, the reference system could not reach the excellent values of the SDEC system;
- The LCA on the ENERGYbase's SDEC system is a first step as more LCAs are needed in other locations and applications to fully assess SDEC technology.

Using the Good Practice SDEC system experience gained at the ENERGYbase building, DEC technology been utilized in another innovative office building project called aspern IQ[13]. In this new system, the regeneration heat will be supplied by the local district heating system at approximately 60°C.

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[13] http://www.asperniq.at/wp-content/uploads/2013/12/aspern-IQ-Folder-Engl.pdf

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5.2. TAFE SDEC BASED TRIGENERATION SYSTEM IN NEWCASTLE, AUSTRALIA

Authors: Sethuvenkatraman S., White S.

5.2.1. BUILDING AND SDEC SYSTEM FACTS

Figure 38: Arial view of Hamilton TAFE building showing solar collectors on the roof

Air Conditioning Industries and CSIRO, in cooperation with the Hunter TAFE Institute, upgraded the existing water and air handling system at the Hamilton Campus in Newcastle (Figure 38) with a solar based heating, cooling and domestic hot water system (tri generation). The installation was designed to provide a net reduction in energy consumption compared to the previous HVAC and gas powered domestic hot water systems. The new units were designed to preheat domestic water and provide air-conditioning for a learning kitchen, a dining room and several offices. The new air-conditioning system consisted of two SDEC Tempered Air Conditioner units (labelled TAC 3 and TAC 7). TAC 7 supplied the kitchen, preparation room and store while TAC 3 served either the offices or the dining room if cooling/heating was requested (a request button was installed in the dining room). TAC 3 could only serve one area at a time; therefore a conventional compression chiller air-conditioner maintained comfortable conditions in the offices if cooling was requested in the dining room. TAC 3 and TAC 7 were identical in design with the only differences being higher air and water flow rates for TAC 7.

The upgrade from the conventional system had two purposes: to demonstrate the functionality of the DEC system and to improve the carbon footprint of the building by reducing both electricity and gas consumption. Given the kitchen environment, it was not expected that the air temperature could be, or should be, tightly controlled at all points in the kitchen. Due to this level of tolerance, a less strictly regulated “tempered” air stream was considered adequate. A once-through fresh air supply was also required. This relaxed constraint suited solar desiccant cooling applications where the amount of cooling is variable as it is reliant on solar availability.

Chilled and/or hot water was supplied to air handling units to treat the fresh air ducted to the teaching kitchens. The site also has high demands for potable hot water (for cleaning and food preparation). The thermal energy required for this duty, for space heating and for the DEC system, were achieved using the same flat-plate solar collectors. This type of collector is widespread and relatively inexpensive in Australia.

Building Facts

Type of building	Teaching institute
Location	Hamilton, New South Wales, Australia
In operation since	2012
System build contractor	Air conditioning Industries (ACI), Australia
Air-conditioned area	500 m²
Solar system used for space heating	Yes
Solar system used for DHW	Yes
Additional Information	
Renewables	400 m^2 of flat-plate solar thermal collectors on the roof of the facility
Other innovations	Two stage desiccant cooling system operating only with solar heat

SDEC System Facts

Desiccant cooling units

Technology	Desiccant and evaporative cooling
Nominal air handling capacity	~ 12000 m^3 /hr of air
Brand of evaporative cooling units	Munters
Cooling load subsystem	Central chiller
Dehumidification	Desiccant rotor international (EDC 1525 x100)
Regeneration details	Heat from the storage tank is supplied through regeneration coils. Regeneration air flow is only present if the tank water temperature is above 70^{o}C.

Solar thermal collector fields

Collector type	Flat-plate collectors
Brand of collector	Rinnai
Collector area	400 m²
Tilt angle, orientation	Approximately 18°, north facing
Collector fluid	Water
Typical operation temperature	55°C to 80°C

Heat back-up system

Technology	Gas boiler backup (SAACKE make)
Nominal capacity	Name plate details not available
Heating load subsystem	Central air handling unit

Heat storage system

Unit # 1

Number of units	1
Technology	Water tank
Storage capacity	4500 l
Shared with heat back-up systems	No
Type of connection	-

Unit # 2

Number of units	1
Technology	Water tank
Storage capacity	4500 l
Shared with heat back-up systems	No
Type of connection	-

Climate

Located	Newcastle, Australia
T_{mean} (T_{max}, T_{min})	20.1°C(T_{max} = 21.9°C, T_{min} =14.7°C) [source : BoM, Australia)
Annual Global solar radiation on horizontal	1706 kWh/m^2 [Source : METEONORM[14]]
Global diffuse on horizontal	743 kWh/m^2
Global direct on horizontal	971 kWh/m^2

5.2.2. DESIGN PHASE

SDEC Design Principles

The summer and winter operational schematics for the unit are shown in Figure 39. Within each

[14] http://meteonorm.com/ (obtained from TRNSYS)

TAC unit, two desiccant wheels, in series, are used to achieve the required dehumidification. The air was cooled following each wheel by a cooling coil supplied with water from a cooling tower. This reduced the temperature increase of the process air due to the heat of adsorption. The cross-sectional area of the desiccant wheel was divided into two sections with 2/3 of the area being used for the supply air side and the remainder for the regeneration side. This ratio was selected to maximize the supply airflow for the given size of wheel without compromising dehumidification performance. The wheel rotates at a constant speed and both processes (dehumidification and regeneration) occur simultaneously. This allows for continuous operation. The desiccant wheel was regenerated by passing heated fresh air back through the wheel in a counter-flow arrangement. The air was heated by two heat exchangers located before each desiccant wheel, on the regeneration side, which were supplied by hot water from the hot water storage tanks. No backup heat source was used and thus, desiccant cooling only occurred when there was sufficient solar heat in the tank. Gas was never used for providing regeneration heat and was only used for space heating and domestic hot water.

After passing through the desiccant wheels, the resulting dehumidified air was cooled by a direct evaporative cooler. The air temperature decreased due to the latent heat of the water vaporizing into the air. If the air was above the supply temperature set point after these stages, further cooling was achieved using a chiller coil supplied by the existing mechanical chiller. This coil was located in the final section of TAC unit before the air was supplied to the ventilation duct.

Altering the flow-paths and passing the air stream through heating coils allowed for winter space heating. The air flow bypassed the desiccant wheels, the evaporative cooler and the chiller coil and instead passes through two separate heating coils. The first coil was heated by solar thermally heated hot water, and the second (for additional backup heating), was heated by the existing gas space heater.

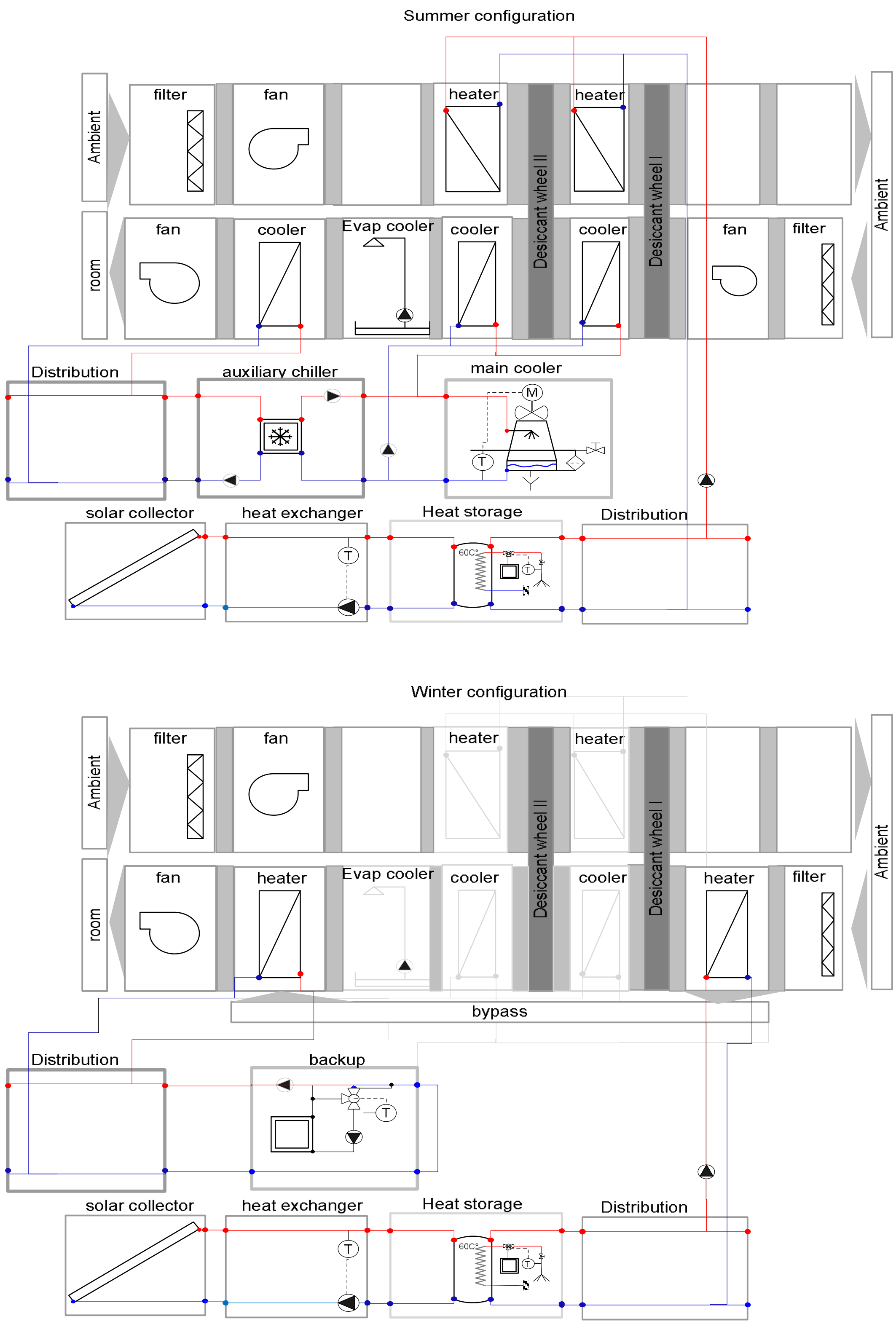


Figure 39: TAFE desiccant based tri-generation system details for winter and summer configurations

Design Principles for SDEC System:

- The SDEC system's regenerator operated fully using only solar heat, no thermal backup was used;
- Unlike most desiccant systems, this system did not exchange or recover heat with the outgoing indoor air in order to avoid any contamination by kitchen odour and grease;
- The two rotor design maximized dehumidification with low temperature heat from flat-plate collectors;
- The system was designed to treat ambient air to 16.5^{o}C with an absolute humidity of 8 g/kg with a desiccant system utilization rate of over 70% during a standard working day (9am to 5pm);
- The system was designed to providing cooling in desiccant dehumidification mode and evaporative cooling mode.

The hot water storage tanks were a core component of the system, storing the water heated by the solar collectors and distributing it for cooling, heating and domestic hot water usage. The collectors supplied heat to two hot water storage tanks (equivalent to 22.5 l/m^2 of collector area) that fed the SDEC system directly and the pre-heated the DHW via an in-tank heat-exchanger. Process water for the TAC units was drawn from the top of the tanks. This configuration was chosen to maximize the temperature supplied to the TAC units. A heat exchanger was located at the bottom of each tank for preheating the domestic hot water on the tube side. Each exchanger was divided into four parallel coils. This arrangement ensured that some solar DHW pre-heating occurred even if temperatures were insufficient to supply the SDEC system. This approach maximized the collector efficiency, prioritized the SDEC supply temperature (while the SDEC was operational) and allowed for DHW pre-heating as an ancillary benefit.

Modeling of the System

The TRNSYS simulation studio was used to calculate the performance of the desiccant system designs (a single rotor desiccant system and a two rotor system were compared) over a full year of operation. This enabled the optimization of the system's design including all key components (i.e., the collector field, the storage tanks and the heat exchangers). Results indicated that an intercooled two desiccant wheel design was preferable.

A parametric sensitivity analysis was performed on the following variables:

- The ratio of desiccant supply air to regeneration air flow rate (S in Figure 40) ;
- The solar flat plate collector area;
- The hot water storage tank volume.

The impact of these variables on the average annual "fraction of demand met" is illustrated in Figure 40. The "fraction of demand met", over an entire year, is the temperature reduction achieved by the system (from ambient), expressed as a fraction of the temperature reduction if the supply air was introduced to the kitchens at 16.5°C (with 16.5°C being a sufficiently low temperature to ensure that ideal conditions were maintained in the occupied space). The parametric sensitivity analysis showed that each of the 2100 L/s air handling units would require approximately 200 m^2 of solar collectors. Target comfort levels could not be met 100% of the time with the solar heat source alone due to hot climatic conditions, irrespective of collector area.

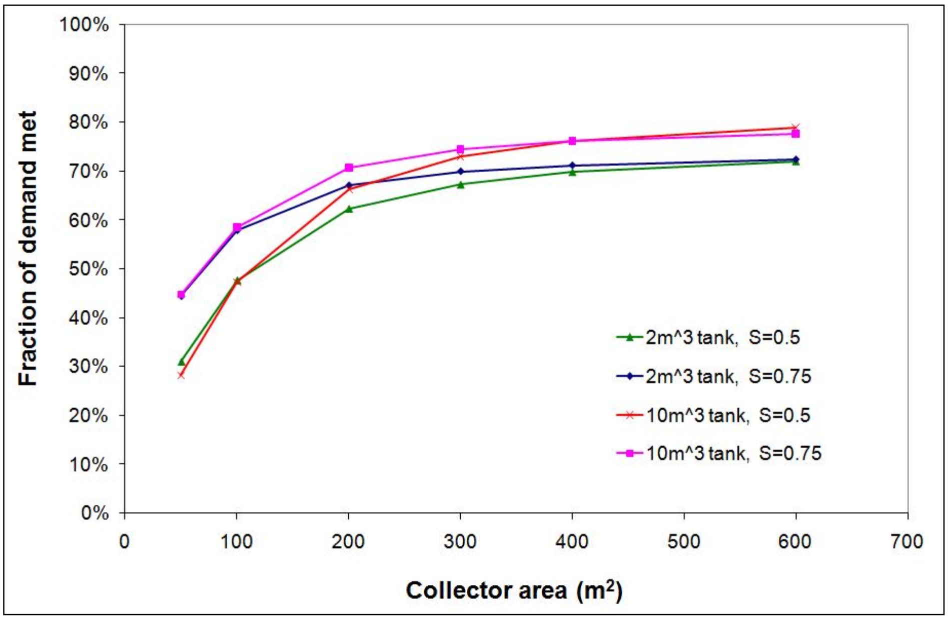


Figure 40: Influence of collector area and storage tank volume on the fraction of demand met

The psychometric chart displaying the variation in instantaneous fraction of cooling demand met is shown in Figure 41 as a function of the ambient conditions. This figure is useful for determining the expected supply conditions given the ambient temperature and humidity. The variation which can exist for any single ambient condition is due to the variations in instantaneous radiation and in the available heat in the storage tank for that particular condition over the course of a year.

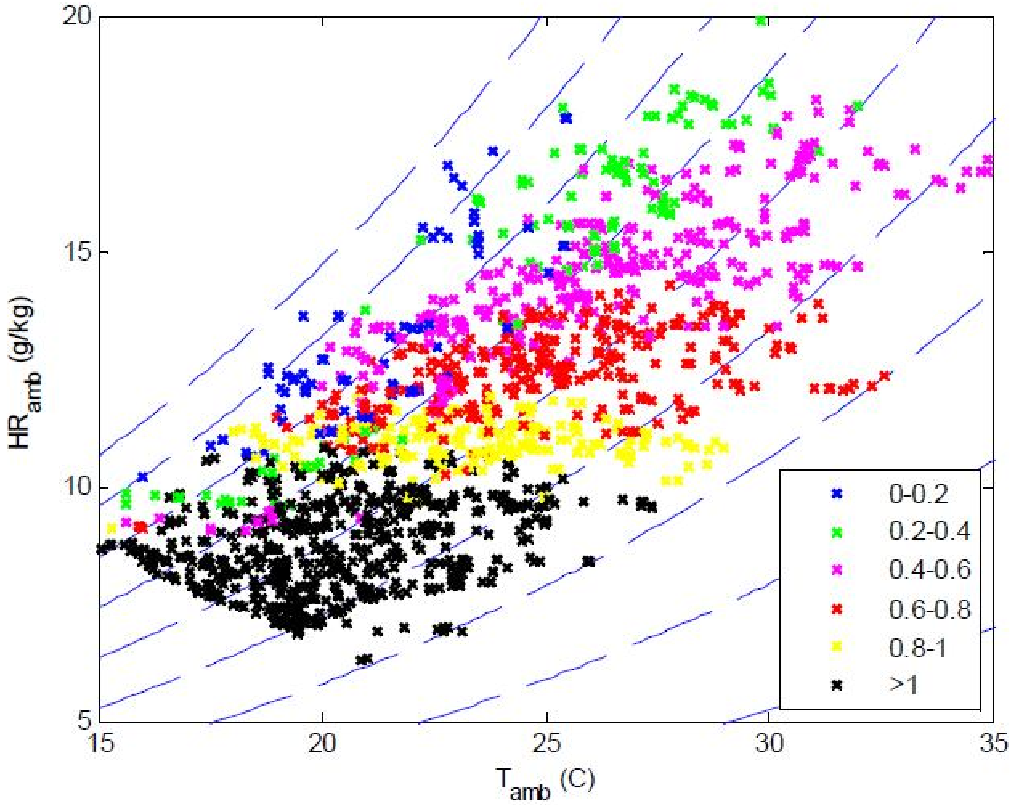


Figure 41: Modelled cooling demand met for a 300 m^2 collector area and a 5 m^3 storage tank

A supply/regeneration flow ratio of 0.5 to 0.75 resulted in an average supply absolute humidity below the 8g/kg requirement, with an average supply temperature of 24°C. The modelling demonstrated that the total cooling supplied was between 50% and 75% of the total cooling demand. The system was relatively insensitive to the hot water storage tank volume. A larger collector field size would yield greater benefits from a larger storage tank. Five cubic meters of storage volume was found to be reasonable for a 200 m^2 collector field.

Integral planning support

Strong collaboration between all stakeholders (Figure 42) was paramount to achieving the successful design and delivery of this project. CSIRO provided the design details for the solar desiccant system using detailed simulations and provided technical advice throughout the project. CSIRO also monitored the performance of the entire system (including the hot water usage and space heating) over a period of two years. Air Conditioning Industries (ACI) was the head contractor for the project. They supplied and installed all the equipment required for the project. NSW public works managed and coordinated delivery of the project along with the host site, TAFE NSW Hunter institute.

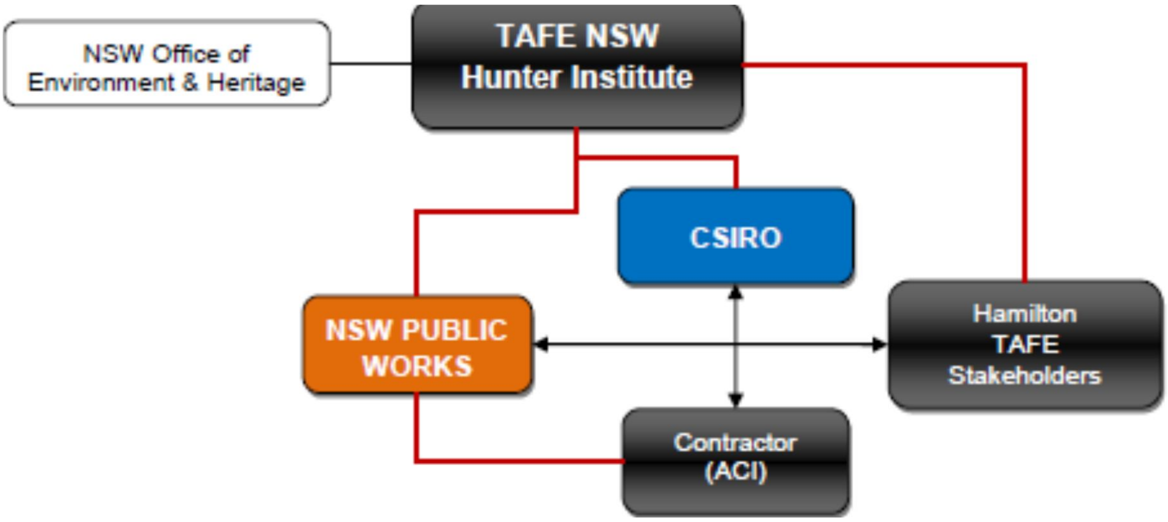


Figure 42: Project stakeholders interaction chart

In delivering the project, the project team overcame engineering and construction challenges including obtaining non-standard components, such as custom thickness desiccant wheels, and incorporating them into AHUs. ACI had to build a new AHU in-house to comply with physical constraints in the plant room. Another challenge centered on finding storage tanks of sufficient capacity that could still fit into the available physical space. This led to the decision to use rectangular storage tanks rather than a stratified cylindrical tank.

5.2.3. OPERATIONAL PHASE

MONITORING DATA AND SDEC SYSTEM PERFORMANCE EVALUATION

Data Monitoring

The entire system was controlled and monitored by a Building Management System (BMS). The BMS monitored and controls the TAC units, the associated hot and chilled water plants and the solar hot water system. The system also allowed users to adjust the set point and dead-band parameters for the serviced rooms.

The desiccant system used the sensors recommended by the IEA SHC Task 38 monitoring procedure. Figure 43 shows the thermal and electrical energy flows monitored by the BMS. The standard Task 38 schematic was adapted to include the unique features of this desiccant system. The energy performance, based on the 2012 to 2013 monitoring period, is provided in Table 11.

Table 12 shows the electrical and thermal COP values of the SDEC system and the overall electrical COP of the system.

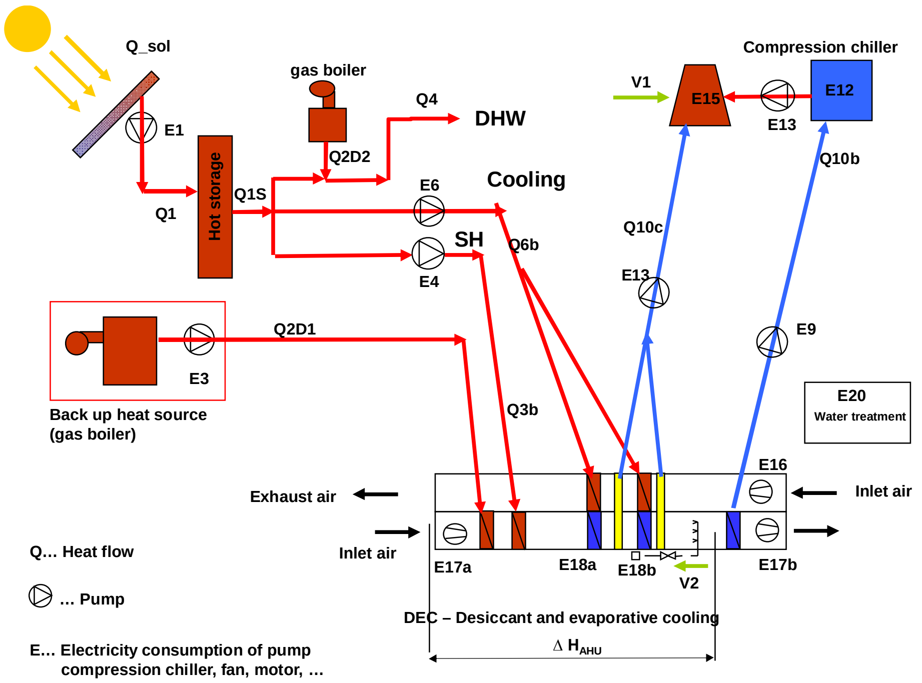


Figure 43: CSIRO desiccant system schematic according to the SHC Task 38 monitoring procedure

Table 11: TAFE SDEC based tri-generation system energy performance from 2012 to 2013

	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
(A) Delta Enthalpy cooling (**ΔH_{AHU}**)	1282	498	794	116	0	0	0	0	76	126	525	1340
(B) Delta Enthalpy heating (**ΔH_{AHU}**)	0	0	0	0	12	324	1405	9836	1936	456	0	0
(C) solar DHW heat produced (Q4-Q2D2)	222	4689	4906	3072	3288	2042	3079	4971	3432	2222	2759	800
(D) total DHW heat produced (Q4)	11967	19969	21034	17573	22423	19984	22686	24078	18752	12921	14225	10005
(E) solar heat delivered to regen coils (Q6b)	9904	3490	4035	1371	0	0	0	0	824	958	3542	9622
(F) Electricity consumption for SDEC	629	231	310	92	0	0	0	0	52	95	266	647
(G) overall electricity consumption	788	356	446	213	124	168	569	1562	586	391	402	810
(H) Solar heat delivered (kWh)	24721	18603	20737	13048	13188	7094	12144	16896	18634	22620	19487	24520

Table 12 TAFE SDEC based tri-generation system energy efficiency performance from 2012 to 2013

	Jan	Feb	Mar	Apr	May	June	July	Aug	Sep	Oct	Nov	Dec
(A/E) COP_{th} (SDEC)	0.13	0.14	0.20	0.08					0.09	0.13	0.15	0.14
(A/F) COP_e (SDEC)	2.04	2.16	2.56	1.26					1.46	1.33	1.97	2.07
(A+B+C+D)/G)COPe (overall)	16.8	57.4	49.0	83.0	180.4	120.6	42.4	21.7	35.4	34.6	36.7	14.0

The data presented in Table 11 and Table 12 was consistent with the expected operation of the system. For example, highest space heating demand happened during the month of August and the highest cooling demand was between November and March. The DHW requirement was relatively consistent over the year, except for during holidays periods (e.g., December to January) where the usage was minimal. The desiccant air conditioning system provided nearly 10% of the total required cooling during the monitoring period.

As shown in Table 12, the system performed with a higher electrical and thermal COP during summer months. The enthalpy change of the process air did not include the sensible cooling provided by the chiller. It was not possible to quantify the cooling provided by the compression chiller and the associated electrical consumption. The overall COP of the system, as defined by to IEA Task 38, was estimated to be a very high value. This was predominantly due to the large amount of heat utilized for DHW that did not require any electrical input. The electrical usage of fans and pumps was not considered so the electrical COP was over 100 all year.

Figure 44 shows the total space cooling energy for both TAC3 and TAC7 per month from April 2013 to March 2014 (calculated as the enthalpy change between the system inlet and before the chiller coil). This data is the combined cooling power provided by all cooling modes.

Over the 12 month monitoring period, the desiccant system accounted 4450Wh of cooling (9% of the total cooling power). The kitchens were the primary source of the cooling load during this period.

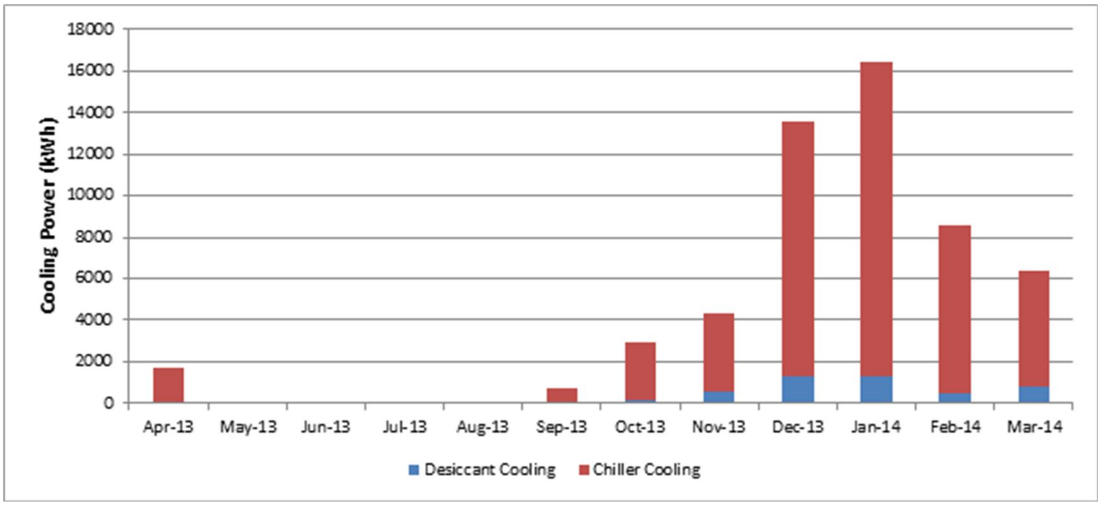


Figure 44: Total cooling demand for TAFE system from April 2013 to March 2014

Desiccant Cooling Mode Performance

The activation of the desiccant cooling system depended on the external conditions. The relative humidity and the solar tank temperature must both meet certain criteria for the system to activate. Three days (the 15th to the 17th of January 2014) for TAC7 have been selected to illustrate the benefits of the system during periods of TAC7's desiccant cooling mode operation. Figure 45 shows the cooling power provided by the chiller and desiccant, combined with the system's operational inputs.

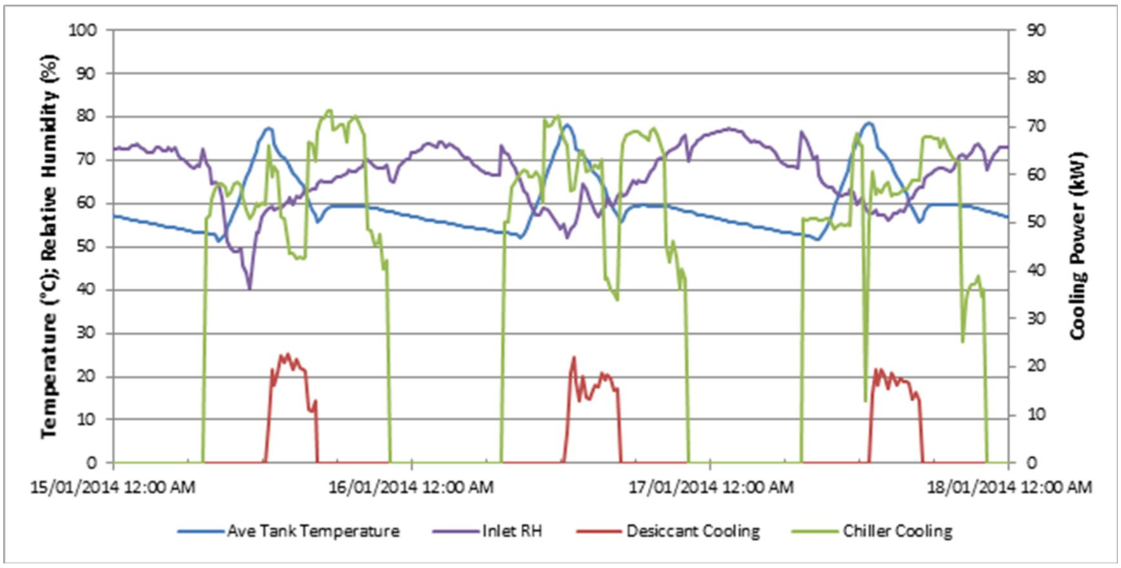


Figure 45: TAC7 desiccant system performance compared to chiller cooling in Jan 2014

Over these three days, the performance of the system was consistent due to the clear weather conditions. The desiccant system operated typically between the times of 12:30pm and 4:45pm. Results indicated that desiccant often operated briefly on Monday mornings (not shown in Figure 45). This occurred because the tank temperature was increased by the solar thermal energy collected during the weekend while the hot water demand was minimal.

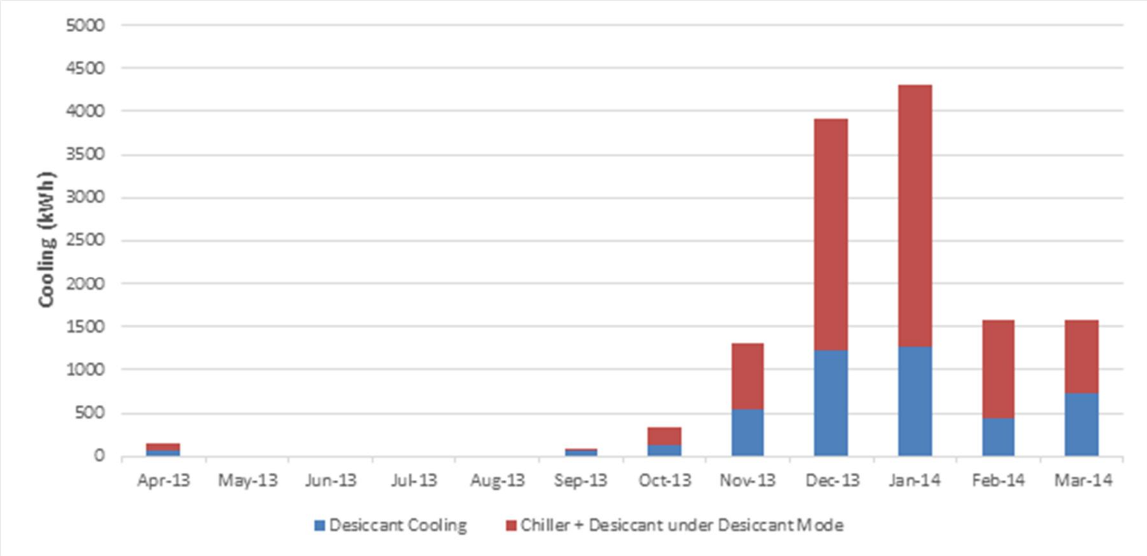


Figure 46: Cooling power under desiccant operation mode from April 2013 to March 2014

This analysis revealed that, when available, the desiccant system components were responsible for a considerable proportion of the cooling. Figure 46 shows the total system cooling provided from both TAC units while in desiccant mode along with the corresponding desiccant component of that total cooling figure. Over the 12 monitored months, the desiccant components provided 34% of the cooling produced while running in desiccant mode. Over this time period, the system operated in desiccant mode for 446 hours compared to the total system run time of 1814 (i.e., for approximately 25% of the total time the system was cooling, it was operating in desiccant mode).

Evaporative Cooling Mode

The system ran in evaporative cooling only mode (without the desiccant or chiller) for 3527 hours of the total recorded 12253 cooling hours during April 2013 to March 2014. This amounted to 29% of the total cooling operating time, contributing to 1382 kWh of sensible cooling. In evaporative cooling mode, the system provided sensible cooling to reduce the air temperature. Enthalpy was not reduced and hence, this did not contribute to the calculated amount of cooling described in previous figures (Figure 44 to Figure 46). When the evaporative cooler achieved the desired supply temperature, it eliminated the need for the chiller to provide cooling.

COSTUMER AND OCCUPANT SATISFACTION

The solar desiccant based tri-generation system has provided an opportunity for the host site to showcase a technology that can sustainably reduce the carbon footprint. The host site is using this system as an educational tool for visitors, students and the local community. Feedback was obtained from the facility management about the system's operation. A few of the comments from the end users are summarized below:

- The users of the facility feel the system provides "adequate" comfort;
- This system is servicing parts of the building that previously did not have access to air-conditioning;
- Lack of temperature set point control by the user in the supply area was cited as a limitation of the system by the facility management and the end users.

The project has already received numerous recognitions including the Green Globe award (2013) from the NSW government.

CONTROL STRATEGY

The TAC units function in one of four operating modes which are controlled by the BMS. The mode of operation is determined by input parameters including the tank temperature, the ambient air temperature, ambient relative humidity and the serviced room temperature. Various operating modes and the component states are provided in Figure 47.

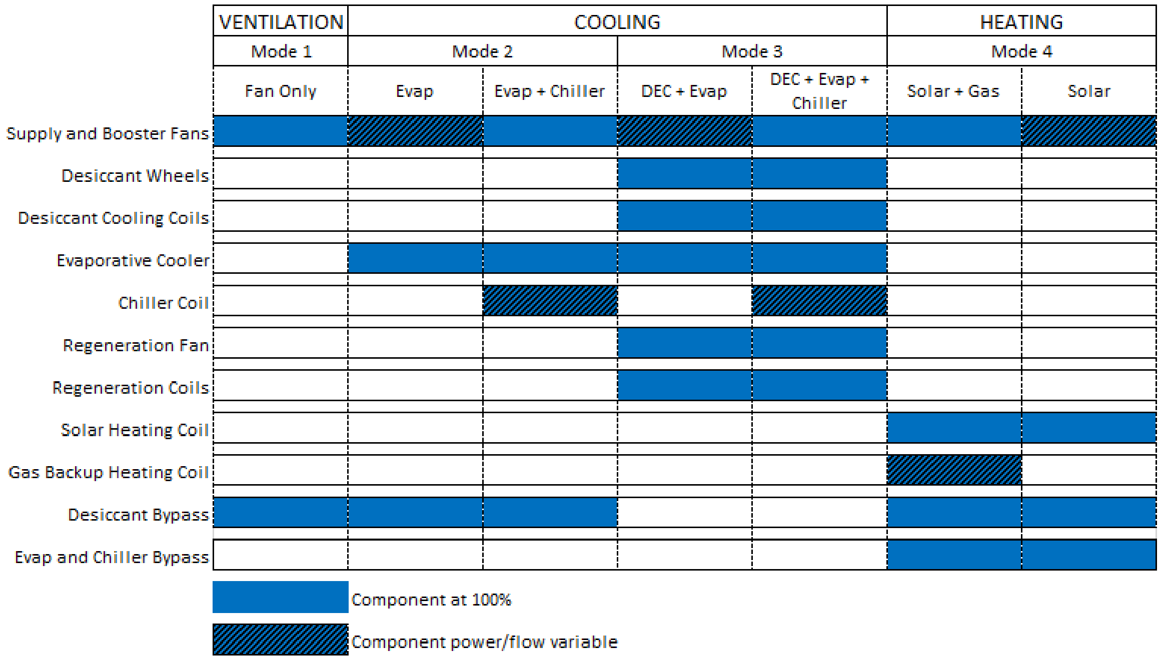


Figure 47: SDEC based tri generation system operation modes

These modes are described below.

Ventilation mode

The supply air fans are programmed to operate from 7am to 10pm on Monday to Friday. Ventilation only will occur when the conditions for other operating modes are not met.

If no call for cooling is made for the dining room, TAC 3 will ventilate the office spaces. If a call is made from the dining room, air supplied from TAC 3 will be redirected and the conventional compressor unit will supply the offices. The dining room request call has a default run time of two hours per call.

Desiccant Cooling Mode

In desiccant cooling mode, the supply air passes through the two desiccant wheels, the two cooling coils, the evaporative cooler and the chilled water coil. The regeneration air is supplied to a solar heated heat exchanger coil prior to entering each desiccant wheel.

For the system to operate in desiccant cooling mode the following conditions must be met:

1. There is cooling demand (i.e., the air-conditioner is operating) and the serviced room temperature is above 21.5 °C;
2. The temperature at the top of the solar energy storage tank is above 70 °C (stops when below 60 °C);
3. The ambient relative humidity must be above 50%.

Temperature is controlled by varying the supply fan speed and the regeneration fan speed. Additional backup cooling is provided by the chilled water coil if the fan speed reaches 100%.

Evaporative cooling mode

In evaporative cooling mode, the supply air bypasses the desiccant wheels and passes through the evaporative cooler and chilled water coil. The space temperature is controlled by varying the supply fan speed. This mode operates only during conditions that enable direct evaporative cooling without dehumidification. This ensures a minimal pressure drop for the fresh air flow and thus, minimal fan energy is expended on supplying air to the conditioned spaces.

For evaporative cooling mode the following criteria must be met:

1. There is cooling demand (i.e., the air-conditioner is operating) and the serviced room temperature is above 21.5 °C;
2. Desiccant cooling conditions are not met (tank temperature insufficient and/or relative humidity below 50%.

Backup chilled water is supplied when the fan speed reaches 100%. The serviced room temperature is then controlled by modulating the chilled water flowrate.

Space Heating

During the operational hours set by the BMS, the system will enter heating mode under the following conditions:

1. The serviced room temperature is below 20°C.
2. The temperature at the top of the storage tank is greater than 40°C (operation stops below 35°C).

The space temperature is controlled by varying the supply fan speed. If the fan speed reaches 100%, the backup, gas powered, hot water system will supply thermal energy to the second heating coil. The air temperature is then controlled by modulating the flow rate to that coil.

IMPROVEMENTS / OPTIMIZATION MEASURES

Continuous system monitoring has helped resolve many operational issues and has enabled improved system performance. Some of these improvements are listed here.

During the initial phases of data monitoring, the desiccant system did not provide any cooling. Flow rates inferred from velocity measurements indicated that the regeneration fan was operating at 60% of the design flow. This was rectified and the desiccant system started showing the necessary temperature drop.

The cooling rate of the desiccant stage was found to be lower than the design values. The intercooling condenser coils used for cooling the process air were supplied with water from the cooling tower that served the institute's existing vapor compression chillers. It was found that the cooling tower set point was a few degrees higher than the design value used during desiccant system design. Since the cooling tower set point was dictated by the vapor compression chiller, other sources for cooling water will be needed.

During system design, the intent was to utilize a cooling set point of 25°C for the controlled zones. This can feel slightly warm to some building inhabitants, however, it was understood to be similar to what the previous TAC units were able to supply. The higher temperature has additional benefits such as reducing the cooling loads and costs in addition to increasing the share of cooling achievable by the SDEC system. After commissioning, the cooling set point was lowered in response to a number of days where the conditioned spaces reached 30°C. The cooling temperature set point has since been raised from 21.5°C to 23°C. This is not a return to the original intended 25°C, but this temperature corresponds to a typical building cooling set point and is expected to save significant electrical power due to the reduced plant cooling output and a greater share of low energy desiccant and evaporative cooling.

Another issue arose when TAC 7 was observed to be in an incorrect mode of operation for a given set of operational points. This is illustrated in the typical weekly behavior observed during September 2013 (Figure 48). It is seen that on three days (23, 26, 27th) the system was in wheel bypass mode, and the condenser water coils were not operating (evaporative cooling only mode), but the regeneration (exhaust) fan was operating and the regeneration coils were heating (full desiccant cooling mode). Heat was therefore being used by TAC-7 for no benefit. Simultaneous operation of regeneration fan (as shown by 100% VSD signal) and condenser coil flow on 25th indicated the desiccant cooling mode was in operation, however, ambient humidity conditions were not high enough to activate desiccant operation on this day. The system operated in the correct mode (desiccant cooling) only during the early morning hours of 30th for the entire week. Table 13 provides a summary of the system's operation during this week.

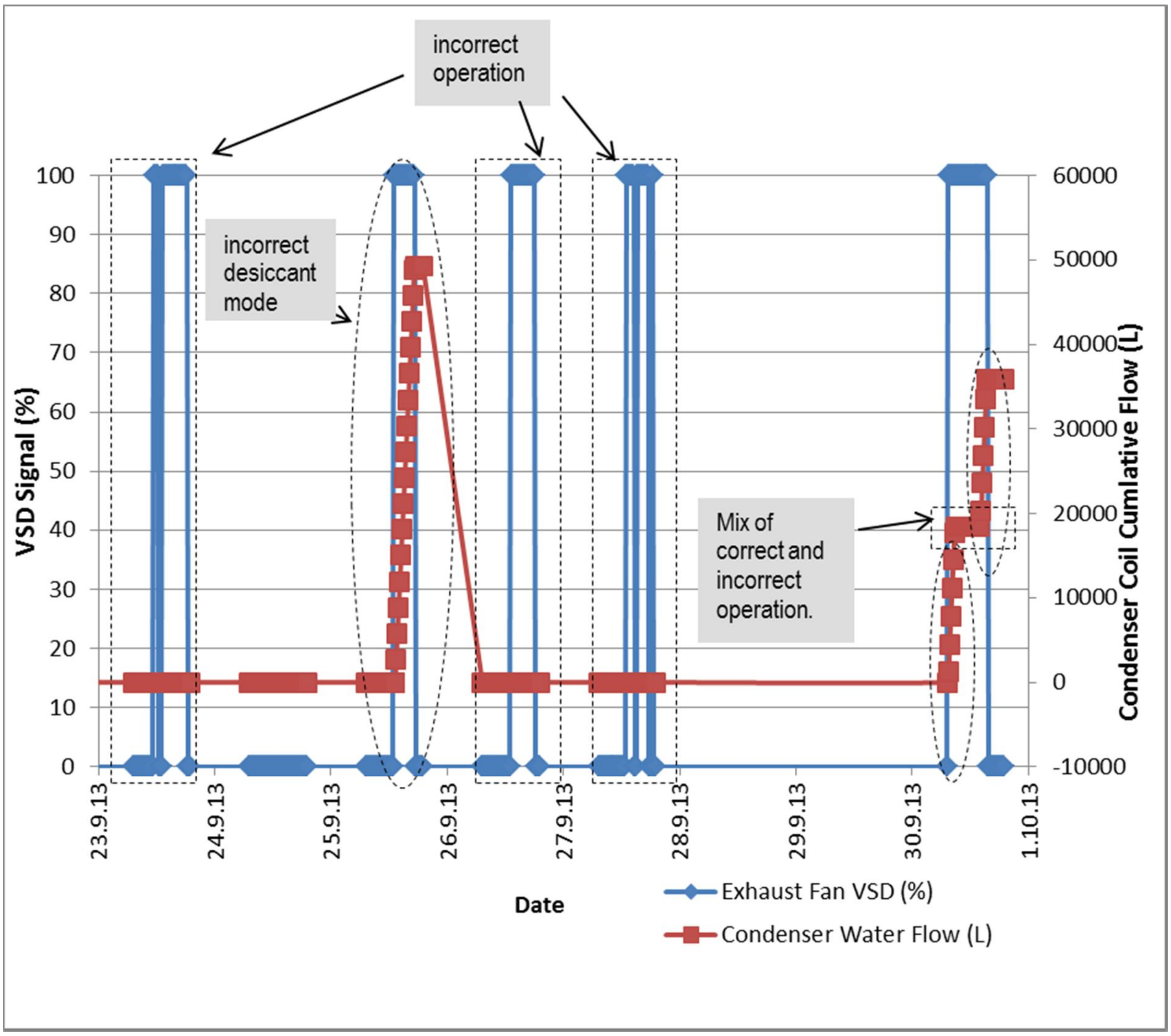


Figure 48: Incorrect operation of TAC7 cooling system

Table 13. TAC-7 operational mode issues

Date	Time Start/ Stop	Amb. Humidity (%RH)	Tank Temp (°C)	Space Max Temp (°C)	Expected Mode of Operation	Actual Mode
23/09/13	11:30	29.99	83.2	22.03	Evaporative only	Mixed
	12:15	21.93	81.88	21.63		
23/09/13	13:15	21.57	80.98	22.37	Evaporative only	Mixed
	18:15	45.52	55.14	21.87		
25/09/13	13:00	44.33	76.8	22.19	Evaporative only	Desiccant
	17:15	47.67	59.76	21.87		
26/09/13	13:15	17.99	75.23	21.99	Evaporative only	Mixed
	18:00	17.41	58.86	21.69		
27/09/13	13:00	34.27	77.31	21.76	Evaporative only	Mixed
	14:30	33.93	69.04	21.59		
27/09/13	15:15	37.66	69.79	22.05	Evaporative only	Mixed
	17:30	42.61	59.95	21.66		
27/09/13	18:30	48.64	61.82	21.85	Either mode (ambient humidity close to 50%RH)	Mixed
30/09/13	7:30	62.22	64.94	21.93	Desiccant mode	Desiccant
	9:00	42.48	59.00	21.85		
30/09/13	9:00	42.48	59.00	21.85	Either mode (ambient humidity close to 50%RH)	Mixed
	14:15	49.87	63.58	22.25		
30/09/13	14:15	49.87	63.58	22.25	Either mode (ambient humidity close to 50%RH)	Desiccant
	19:00	43.48	57.22	22.83		

The issue appeared to have been a logic problem in the BMS where two different desiccant mode signals (one for the solar hot water pumps and the other for the exhaust fans and dampers) were confused. When the tanks had sufficient heat, the solar hot water pump circulated water to the TAC units even when they were not in desiccant mode.

The control logic was corrected and during subsequent tests it was found to be operating correctly. Figure 49 shows condenser coil supplying water whenever the regeneration fan was functioning which indicated the correct desiccant cooling mode.

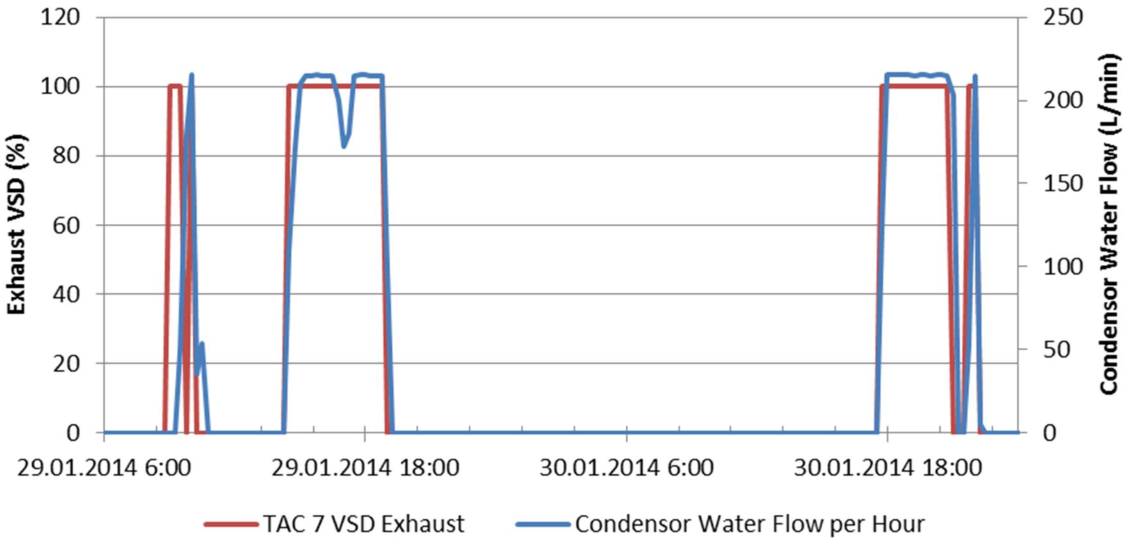


Figure 49: Corrected operation of the desiccant system

The total cost of this project was about 1.3 Million AUD. Main cost component details of the trigeneration system are shown in Figure 50. The solar heat delivery system is the predominant cost that includes the cost of collectors, storage and space heating system. The DEC system cost includes the cost of wheels, regeneration and condensation coil systems and fans. The DEC system installation at TAFE required infrastructure modifications to accommodate the TAC systems. This required extra costs associated with work contracting, decommissioning and demolition. As a result the installation and commissioning contributes to more than 10% of total system cost.

Implementation of solar air conditioning in new buildings is expected to bring down the costs significantly as the existing installation had to accommodate the design within the building limitations.

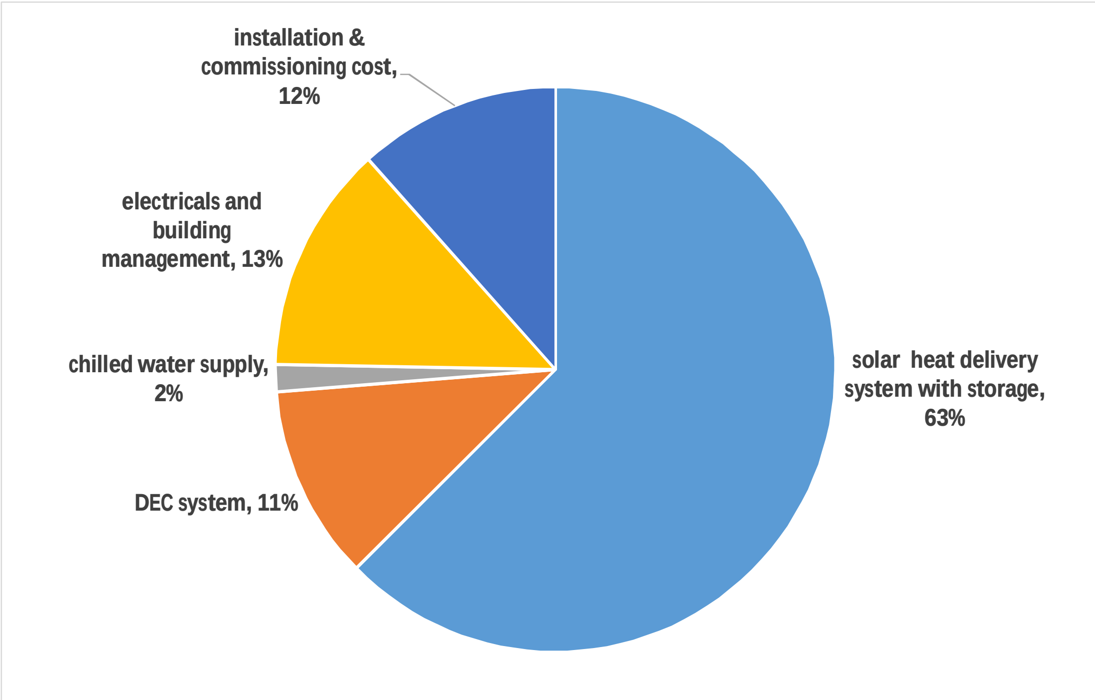


Figure 50: break down of major component cost of SDEC system at TAFE, NSW, and Australia

5.2.4. KEY FINDINGS / LESSONS LEARNED / RECOMMENDATIONS

A two-rotor, solar desiccant cooling & solar DHW pre-heating system has been installed and commissioned at TAFE Hamilton Campus. A summary of the measures that have contributed to the efficient design and operation are listed here.

The SDEC system only used solar heat for regeneration and no thermal backup. This arrangement eliminated the usage of gas during the regeneration of the desiccant. This was possible due to the low tempered air requirements of the serviced areas (i.e., the kitchen).

The usage of solar heat in the tank was maximized by arranging the SDEC heating coils at the top of the storage tank and the DHW coils at the bottom of the tank. As a result, the DHW system could still be pre-heated even when insufficient heat was available for the operation of the SDEC system.

Sharing solar heat between the DHW pre–heating systems and the space heating system has resulted in higher savings for the end user when compared to using the heat only for desiccant cooling. The tri-generation nature of this system has also resulted in the usage of the system all year long.

During the design stage, the modeled results clearly showed a two rotor system providing the required dehumidification and cooling rates while only needing low temperature solar heat. During tests, however, the process air was not cooled enough due to the high temperature set point of the cooling water. Having an independent cooling water supply for this system could result in the system providing a higher cooling rate.

Careful study of a control scheme based on both ambient and indoor conditions has helped improve the system. The system was able to operate in an evaporative cooling mode for nearly 30% of the total cooling time. This provided sensible cooling to the serviced areas while minimizing the electrical power consumption.

Continuous monitoring of the system's performance was found to be important in resolving operational issues with the control system and changes to the cooling set points. Thus we recommend this practice for all SDEC installations for at least one summer period.

Using additional sensors to monitor the desiccant unit's performance has helped identify problems with the intercoolers and the regeneration air flow rate. Careful study of the desiccant system's behavior during the commissioning stage can result in the identification of these issues during early stages of the operation.

Commissioning and integrating this large scale, non-standard system, into an existing public facility would not have been possible without collaboration between all stakeholders. Sound engineering practices along with proper project management enabled the successful realization of this project.

As SDEC technology is new, building managers and plant room maintenance staff may be unaware of many characteristics that require the altering of traditional HVAC system settings and operational procedures. These changes are necessary to maximize the systems performance, efficiency and to maintain comfortable conditions in the serviced areas. In addition to carefully monitoring and analyzing the system during the early stages of the system's operation, communicating with and educating the plant room personnel can help identify and resolve any issues related to the SDEC system's operation in a timely manner.

5.3. POLIMI SDEC

Authors: Frein A., Aprile M., Muscherà M.

5.3.1. BUILDING AND SDEC SYSTEM FACTS

The “Leonardo da Vinci” Residence building in Milan, (Viale Romagna, 62) is a dormitory for students of Politecnico di Milano University. Built in 1934, it is a typical prewar Italian construction, characterized by low insulation levels (both walls and windows), which leads to high energy demands. Prior to the installation of the SDEC system, the incoming ventilation air was treated by a traditional air handling unit. The remainder of the building’s heating and cooling loads were met by a distributed terminal system (fan coils) powered by gas boilers and traditional chillers. After five years of operational failures, the building management staff decided to obtain a new air-conditioning system.

The new demo SDEC plant was installed in 2013 to supply part of the primary ventilation air required by UNI 10339. The system was to be used to assess the capacity of this technology to meet the real needs of a building in the Mediterranean climate, as well as to monitor the reduction of energy consumption.

Figure 51: The Leonardo da Vinci Residence in Milan, Italy

Building Facts

Type of building	Dormitory
Location	Milan, Italy
In operation since	2013
System operated by	Politecnico di Milano
Air-conditioned area	700 m^2 (SDEC)
Solar system used for spac e heating	Yes
Solar system used for DHW	No
Additional Information	
Renewables	110 m^2 of flat plate collectors
Other innovation	

SDEC System Facts

Desiccant cooling units

Technology	Desiccant and evaporative cooling
Nominal volumetric flowrate	6000 m^3/hr
Nominal capacity	~ 36 kW
Brand of cooling units	TECFER
Cooling load subsystem	AHU
Dehumidification	Desiccant rotor (DST RUF 122)
Regeneration power	~ 42 kW (nominal conditions)

Solar thermal collector fields

Collector type	Flat-plate collectors
Brand of collector	Sonnenkraft
Collector area	102 m^2 (net area)
Tilt angle, orientation	35° South-East
Collector fluid	Water-Glycol 40%
Typical operation temperature	20 to 40°C winter (ΔT~10°C); 75 to 85°C summer (ΔT~10°C)
Heating load subsystem	AHU

Heat back-up system

Unit #1

Technology	Gas boiler
Nominal capacity	42 kW (for SDEC operations)
Heating load subsystem	AHU

Unit # 2

Technology	Electrical Heat Pump (HP), two capacity steps (0-50-100%) - two parallel compressors
Nominal capacity	27,4 kW (cooling); 32,9 kW (heating)
Heating load subsystem	AHU

Heat storage system

Number of units	1
Technology	Water tank
Storage capacity	5000 l
Shared with heat back up systems	No
Type of connection	Predisposed for DHW

Climate

Located	Milan, Italy (45°27’ N/ 9°11’ E)
T_{mean} (T_{max}, T_{min})	13,7°C (31,9°C max, -5°C min) *Design T from UNI EN 10349 and UNI EN 12831*
Global solar radiation on horizontal	1253 kWh/m^2 · year (METEONORM)
Global diffuse on horizontal	677 kWh/m^2 · year (METEONORM)
Global direct on horizontal	576 kWh/m^2 · year (METEONORM)

5.3.2. DESIGN PHASE

SDEC Design Principles

The task of the SDEC system was to supply 6000 m³/hr of fresh air to the internal rooms (for hygienic requirements). Standard fan coils placed in every conditioned space treat the internal loads. The neutral conditions are achieved through the treatment of high thermal loads, both in cooling and in heating season.

The supply air temperature was maintained in the range of 20°C to 24°C (climatic control) and the humidity was intended to be limited to 10.5 g/kg. The humidity set point could only be achieved during the summer when sufficient solar heat was available but the target humidity was only intended to be a desired condition and was not a strict requirement. Since the system was being used in a southern European climate, winter humidity control is not required.

The newly installed system integrated a non-conventional DEC AHU with a solar thermal system and an electric-reversible heat pump (HP). During the summer, the HP cooled the supply air (Active Cooling) and pre-heated the regeneration air (when dehumidification was needed). The remainder of the regeneration thermal energy was provided via solar energy. During the heating season, if needed, a heating coil heated the supply air (Active Heating), using thermal energy drawn from the solar tank, the HP or the back-up heater (BUH).

Figure 52: Photos of DEC AHU, solar collector field and the electric heat pump

The main features of the installed SDEC system and the differences from a standard DEC system are listed below:

- <u>Regeneration process:</u> external air is used instead of return air, in a dedicated channel.
- <u>Regeneration heat:</u> provided by the HP condenser (heats the pre-regeneration coil) and the solar array (heats the regeneration coil);
- <u>Hybrid scheme:</u> a traditional cooling coil, cooled by the HP evaporator, replaces direct evaporative coolers;
- Static heat exchanger: no rotary HX;
- <u>Bypasses:</u> The desiccant wheel and the static heat exchanger can be by-passed thanks to PI controlled shutters.

The POLIMI system was designed to achieve the following goals:

- High air quality: a static heat exchanger and a third channel was used for the regeneration process to avoid possible contaminations of the supply air. Furthermore, replacing the supply air humidifier with a cooling coil reduced hygienic risks and avoided water treatment needs;
- Low electricity consumption: by avoiding the pressure drop caused by additional rotary heat exchangers, the electricity consumption of the two by-pass and regeneration channel fans was reduced;
- Simpler control scheme: replacing direct evaporative cooling systems with the cooling coil simplifies the control process.

Solar heat regeneration	Solar energy and heat pump condenser rejection, no thermal back-up
Sorption wheel	Sorption wheel use for dehumidification (only summer operation)
Control function of air handling unit	Supply of primary air

Design set point of ambient air (Summer)	T=20-24°C; HU=10,5 g/kg (summer)
Design set point of supply air	Temperature: 20°C/24°C (climatic control) Humidity: 9,5 g/kg (only summer)
Design of volume air flow	Constant volume flow (6000 m^3/hr) due to the hygienic requirements
DEC configuration	Application of hybrid non-conventional desiccant cooling system (no direct evaporative cooling, separated regeneration duct)
DEC system provider	Single provider delivered the entire plant POLIMI sets control strategy equipment

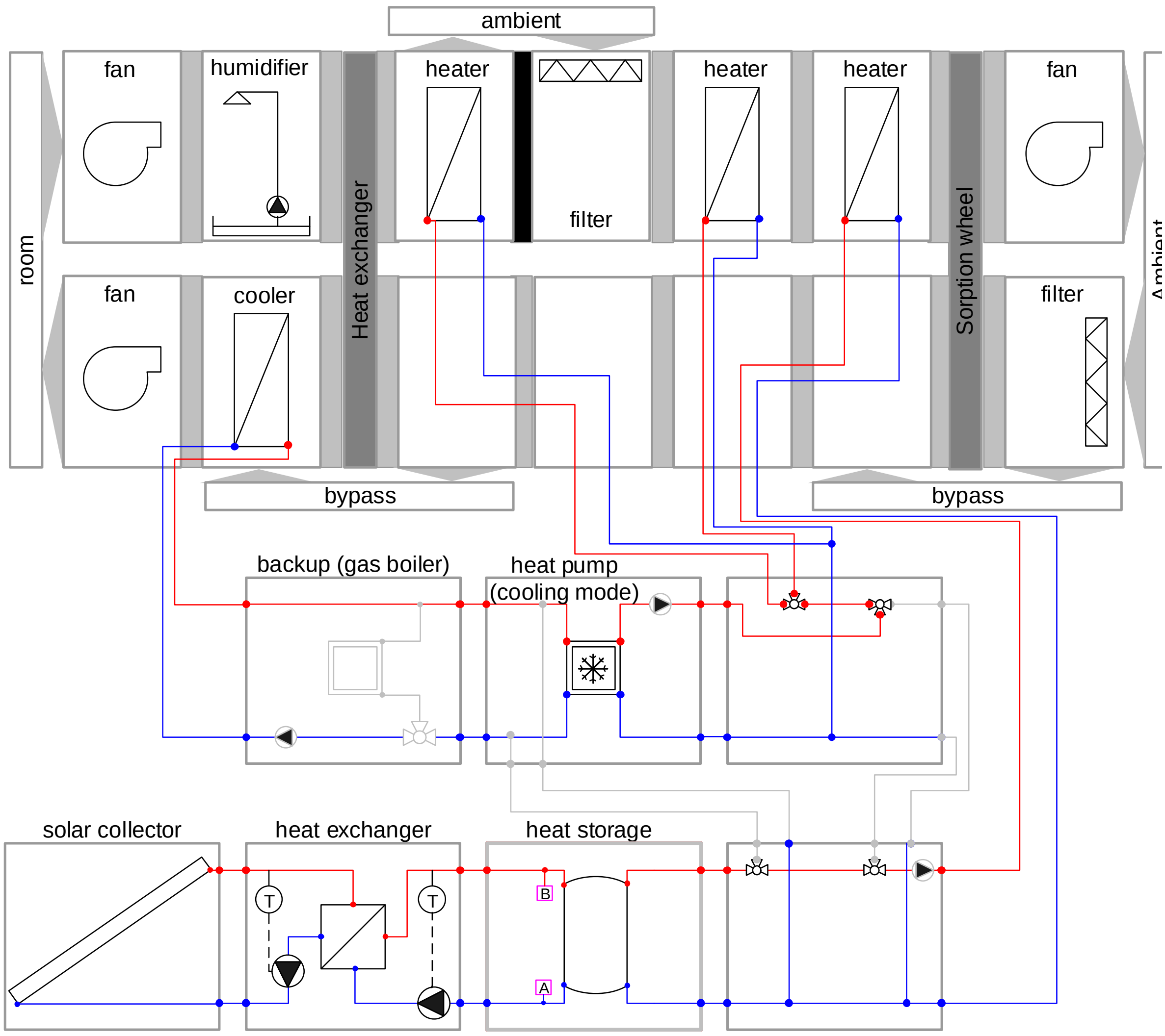


Figure 53: POLIMI SDEC System schematic – summer configuration

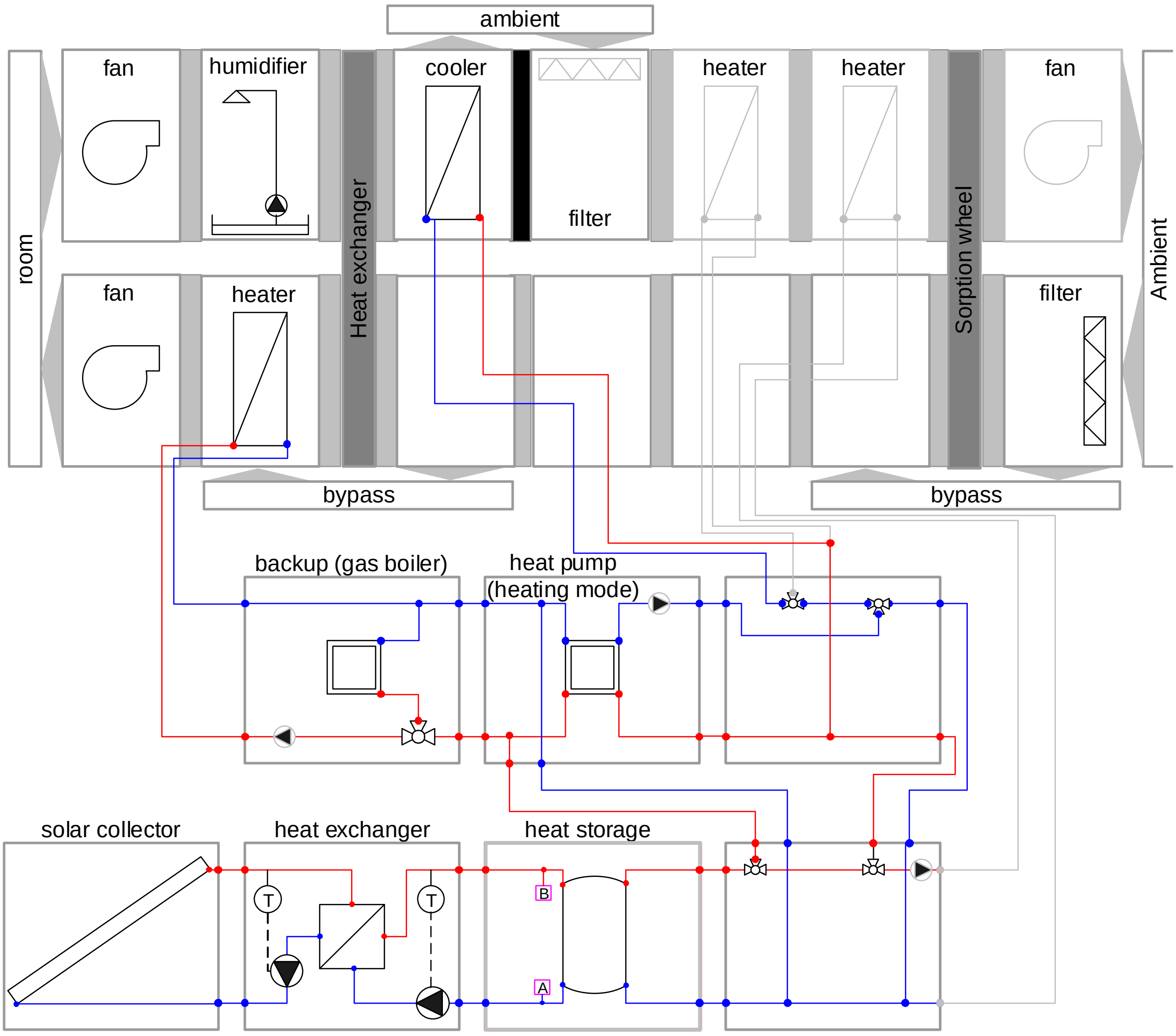


Figure 54: POLIMI SDEC System schematic – winter configuration

Integral Planning Support

The main challenge of the HVAC system upgrade was the integration of new non-conventional components into the existing structure. The installation was intended to be an experimental system, but the real needs of the building's occupants set important limits. Additional constraints were introduced by the non-optimal available physical space, which forced the system's components to be displaced into a non-compact configuration.

The design was based on an extensive simulation[15] of the system. This allowed system's characteristics to be determined, the theoretical concept of the control strategy to be tested and performance calculations were done to help size the system.

First Calculations

Algebraic and differential models of the POLIMI system's components were developed in MATLAB. The dynamic behavior of the system and the first control strategy was evaluated. The desiccant rotor and the heat pump models were based on the manufacturer supplied data (grey box models); the other components were physically defined. The aim of this first model was to optimize the control system and evaluate the potential savings.

The selected control strategy design approach is known as the Finite State Machine approach (FSM). This system activates different operational modes in a sequential order. The operational mode is a package of a predefined number of components' actions. The transition between modes is based on simple logical rules.

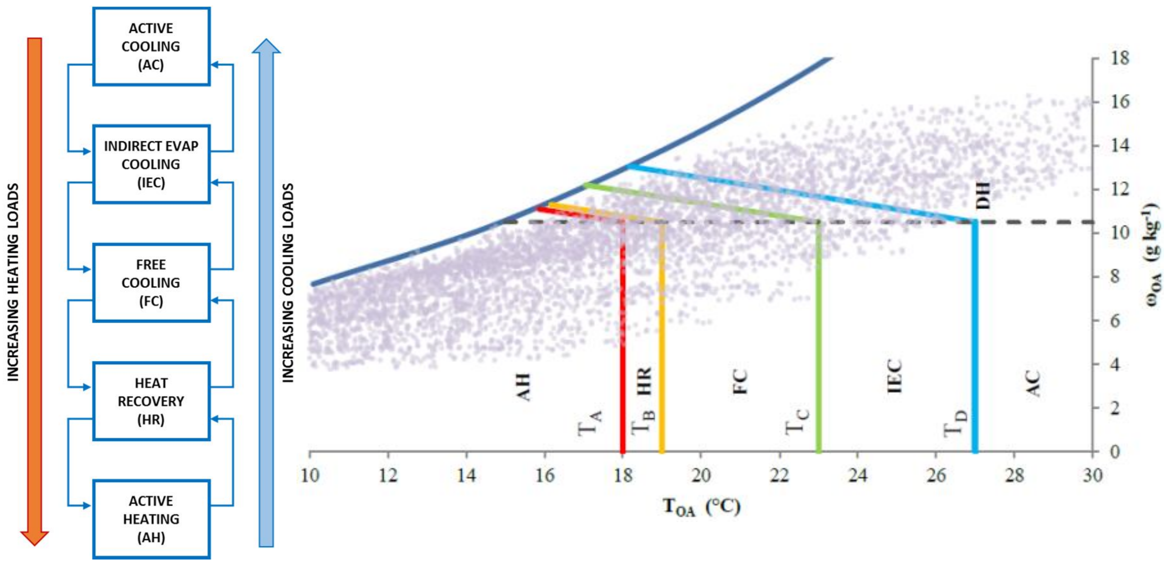


Figure 55: The control logic sequence is displayed on the left and the outdoor air conditions that these modes were operated in are shown for a yearlong simulation

The effect of different parameters on the climatic control systems was investigated. The control

15 Aprile, Scoccia, Motta. Modelling and control optimization of a solar desiccant and evaporative cooling system using an electrical heat pump. IEA SHC 2012.

was based on the outdoor temperature so the supply air maximum temperature was proportionally decreased when the external temperature exceeded 26°C. Similarly, the supply air minimum temperature was proportionally increased when the outdoor temperature fell below 15°C. The maximum absolute increase (in heating) and decrease (in cooling) were varied as shown in Figure 56 according to the following rules:

- CC 2/0: heating increase of 2°C and cooling decrease of 0°C;
- CC 2/2: heating increase of 2°C and cooling decrease of 2°C;
- CC 2/4: heating increase of 2°C and cooling decrease of 4°C.

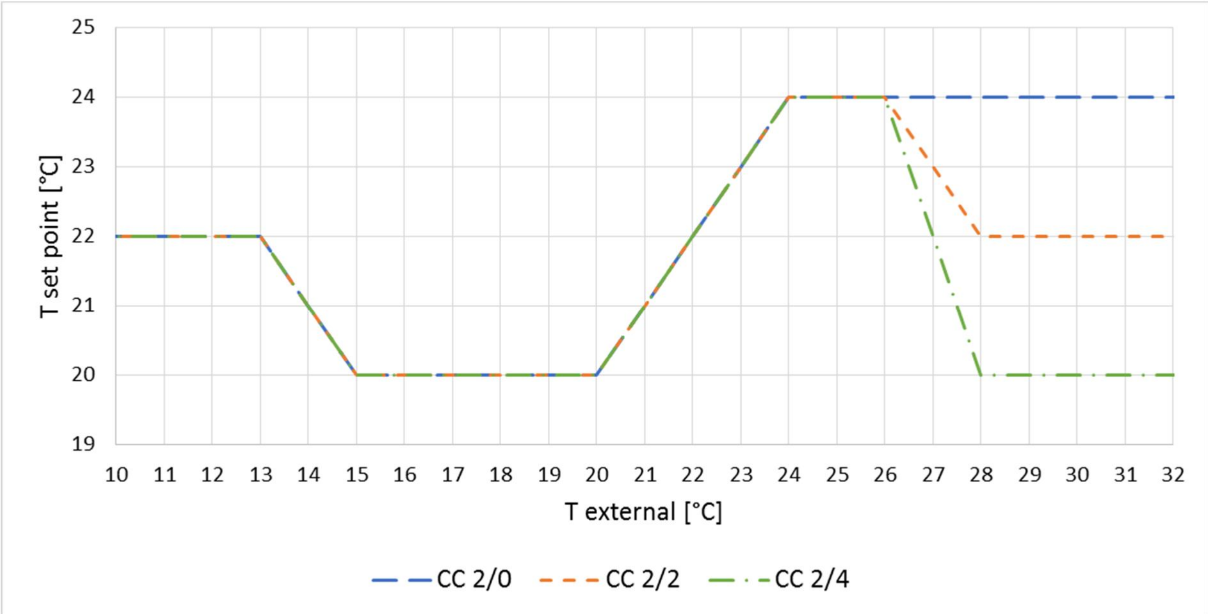


Figure 56: Different options for the supply air set-point temperature

The analyzed performance metrics included the primary energy consumption and the dehumidification capacity. The following graphs (Figure 57, Figure 58 and Figure 59) show the simulation results for a typical summer week when controlled using the CC2/2 control methodology.

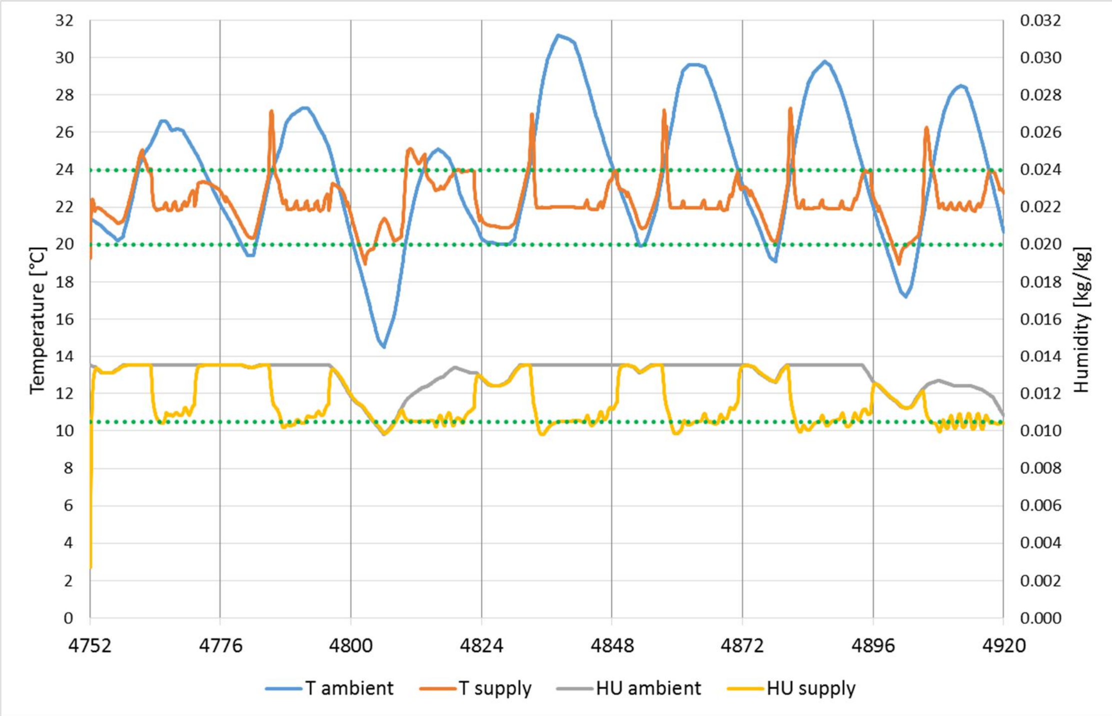


Figure 57: Summer week simulation results (July 18th to 24th) for the Supply air versus the ambient air conditions

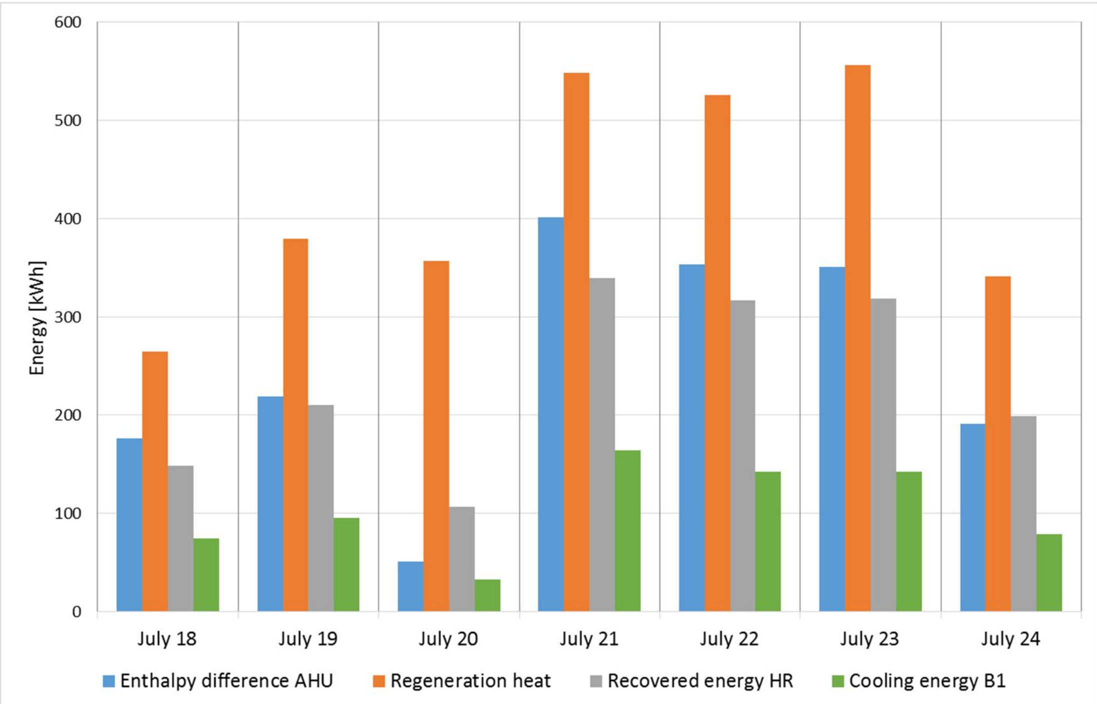


Figure 58: Summer week simulation results (July 18th to 24th)for Global enthalpy gap, total heat for regeneration, heat recovery through heat exchanger and cooling energy delivered by supply air coil B1

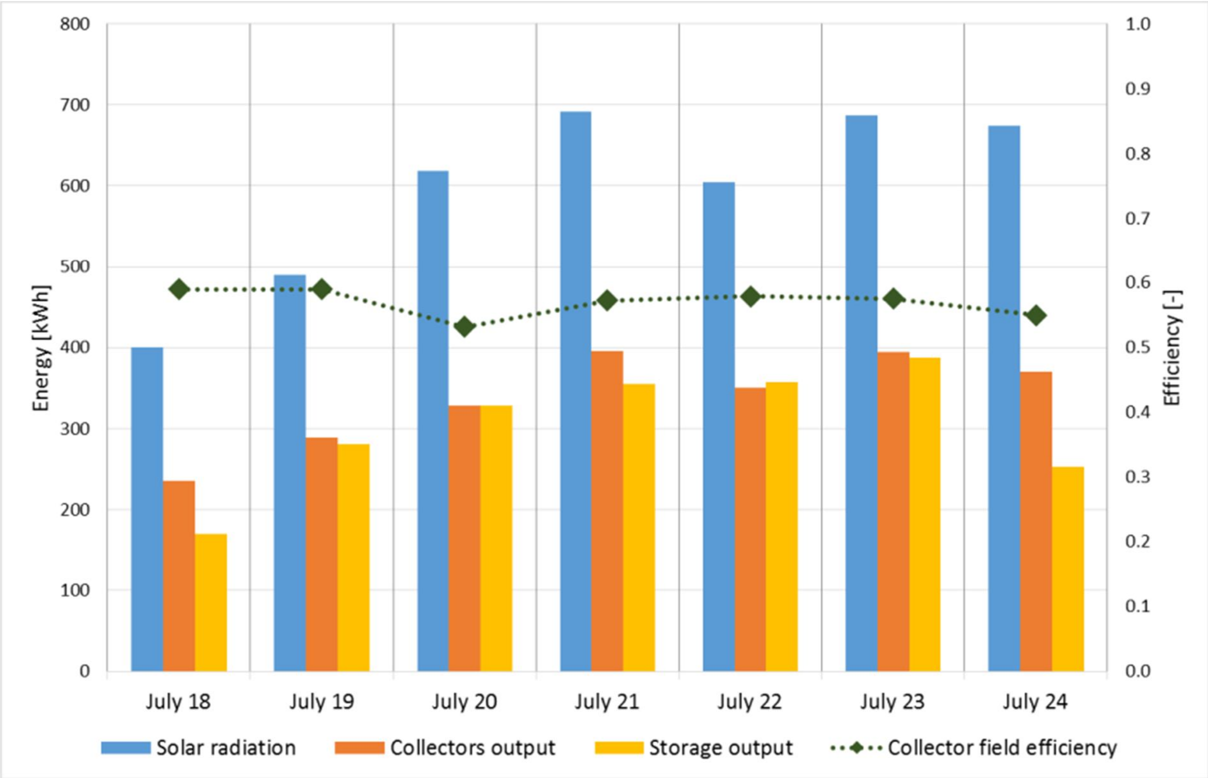


Figure 59: Summer week simulation results (July 18th to 24th) for the solar system's performance.

The model also used to determine the effect on performance of the integration of a second sensible heat exchanger in series with the first one. Despite an increase in primary energy savings of 10%, the decreased contribution of the heat pump caused the dehumidification capacity to decrease by approximately 8,5%. The reduced rate of heat rejection from the condenser reduced the temperature of the air being used to regenerate the desiccant (as the condenser was used to pre-heat the regeneration air).

Key Findings

An analysis of the simulated results led to the following key findings:

- The supply air temperature fell within in the comfort range of 20°C to 24°C for 89% of the week. The average humidity when the dehumidification was active was 10.67 g/kg (set point at 10.50 g/kg), even though dehumidification was only possible when enough heat was available in the solar storage tank. This result shows that in the analyzed period the supply air is well controlled, despite the heavy external conditions.
- The heat rejected by the HP condenser covered the 28% of the total regeneration heat demand. The remaining 72% was supplied by solar energy. This allowed the dehumidifier to run for longer periods as less heat was drawn from the solar storage tanks.
- The sensible heat was removed by the heat exchanger through indirect evaporative cooling operations (this totaled 69% of the total heat removed during the whole week) and by the electrical HP (this totaled 31% of the removed heat, with a weekly electrical consumption of 258 kWh and an average chiller EER of 4.1).
- The collector field efficiency strongly depended on the incident solar radiation and on the dehumidification load (the higher the demand for regeneration, the lower the temperature in the solar storage tank). On sunny days with a significant regeneration demand, the solar collector field efficiency reached 59%.
- If times of limited or no regeneration loads coincided with times of high solar irradiance, the possibility existed for the collectors to stagnate. This is the reason for the implementation of the “anti-stagnation” operating mode (see section control strategy).

5.3.3. OPERATIONAL PHASE

Monitoring Data and SDEC System Performance Evaluation

In order to assess the energy performance, an array of sensors was designed according to the IEA SHC Task 38 guidelines. The Task 38 scheme required modifications in order to match the POLIMI SDEC system's characteristics. The system is shown in Figure 60.

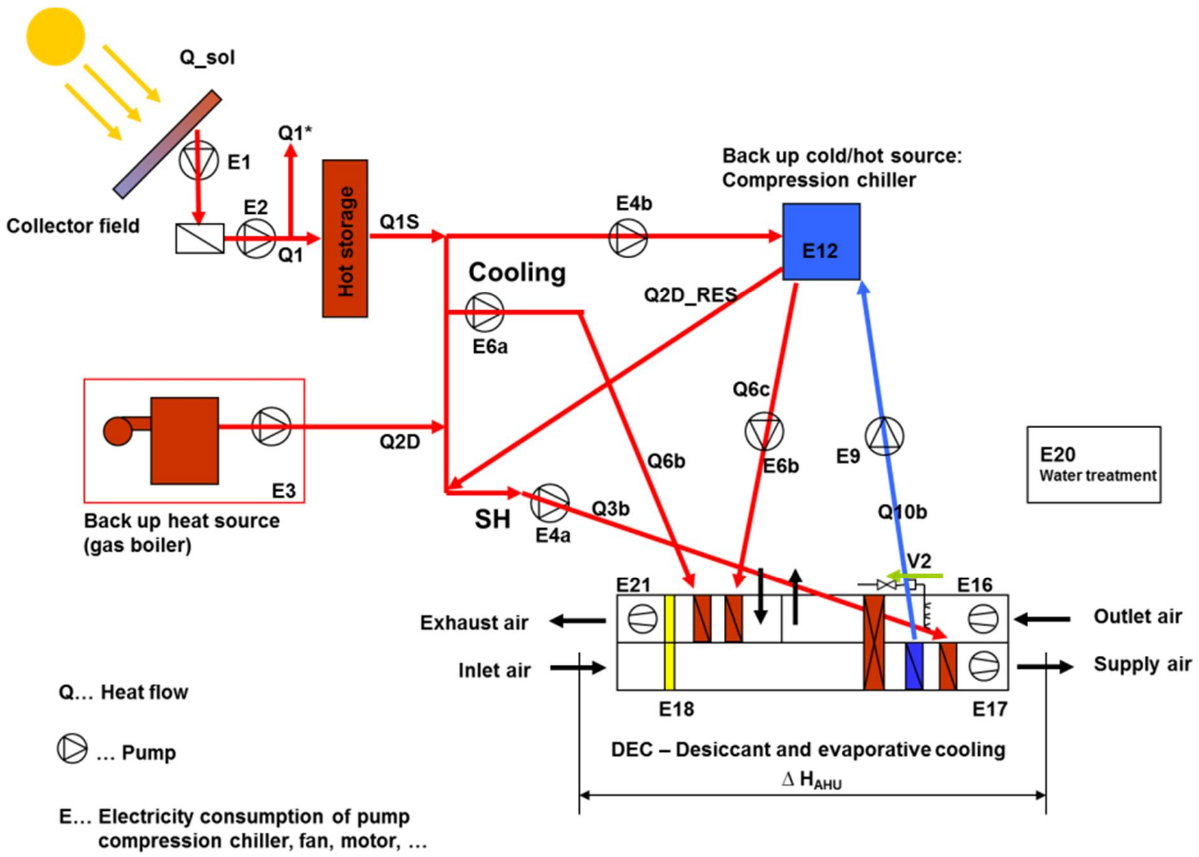


Figure 60: Task 38 scheme, modified to match the POLIMI SDEC system's characteristics

The cooling performance achieved by the system during summer is shown in Table 14 and Figure 61. Following that, the heating performance during winter is shown in Table 15 and Figure 62.

Table 14: SDEC system summer performance

		Apr (s)	May	Jun	Jul	Aug	Sep	Oct (s)
A) Enthalpy difference AHU, ΔH_{AHU}	kWh	120	823	4243	5704	-	-	-
B) DHW production, Q4	kWh	-	-	-	-	-	-	-
C) Cold output backup chiller to AHU, Q10b	kWh	7	266	2895	4392	-	-	-
D) Heat rejection (cooling tower), Q10c	kWh	-	-	-	-	-	-	-
E) Hot storage input to DEC system (regen), Q6b	kWh	0	320	3971	5809	-	-	-
F) Chiller condenser input to DEC system (regen), Q6c	kWh	0	54	2676	4364	-	-	-
G) Solar thermal output to hot storage, Q1	kWh	2574	5061	5512	6238	-	-	-
H) Solar irradiation on total collector aperture area, Q_sol	kWh	14849	17955	17245	17104	14587	13937	8571
I) Electricity consumption (overall)	kWh	1261	1583	3171	3910	-	-	-
I.1) Electricity consumption (total)	kWh	248	425	1609	2118			
J) Electricity backup chiller, E12	kWh	14	138	892	1302	-	-	-
K) Electricity cooling tower, E13b	kWh	-	-	-	-	-	-	-
L) Electricity backup coil (regeneration), E21	kWh	-	-	-	-	-	-	-
Collector field efficiency, G/H	%	**17%**	**28%**	**32%**	**36%**	**-**	**-**	**-**
Total Electrical COP (*), (A+B)/I1	-	**0,48**	**1,94**	**2,64**	**2,69**	**-**	**-**	**-**
Thermal COP_DEC (**), (A-C-D)/(E+F)	-	**-**	**1,49**	**0,20**	**0,13**	**-**	**-**	**-**
Chiller average EER, C/J	-	**0,51**	**1,93**	**3,25**	**3,37**	**-**	**-**	**-**
Summer Solar Fraction, G/(G+J+K+L)	%	**0,99**	**0,97**	**0,86**	**0,83**	**-**	**-**	**-**

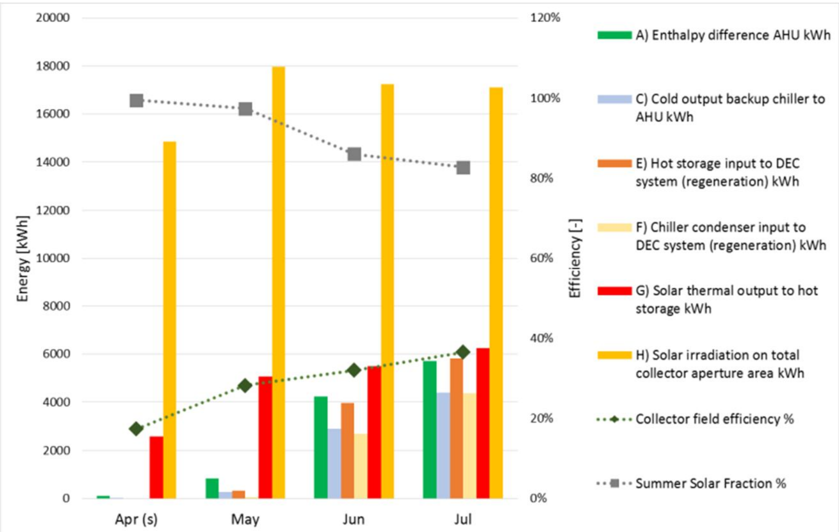


Figure 61: Summary of cooling season performance

(*) The total electrical COP takes into account the total electricity consumption as defined in Task 38 (i.e., the electrical fan consumption was excluded but power for the extra pressure drop due to rotor, regeneration ducts, extra heating coils, and static heat exchangers were included). This allowed for comparison of the system's electrical effectiveness to other systems, because the fans are present in each possible system configuration.

(**) The thermal COP of the DEC system does not consider the cooling contribution of the chiller (through the supply air coil). This was because the aim of this evaluation was to evaluate the DEC system's performance.

Table 15: SDEC system winter energy performance

		Oct (w)	Nov	Dec	Jan	Feb	Mar	Apr (w)
A) Enthalpy difference AHU, ΔH_{AHU}	kWh	-	6590	6271	9560	10975	7842	4292
B) DHW production, Q4	kWh	-	-	-	-	-	-	-
C) Space heating consumption (conventional), Q3a	kWh	-	-	-	-	-	-	-
D) Space heating consumption (ventilation), Q3b	kWh	-	1553	1987	3243	3975	2181	876
E) Hot output backup chiller to AHU, Q2D_RES	kWh	-	392	1115	449	1547	352	0
F) Backup heat into storage, Q2S	kWh	-	-	-	-	-	-	-
G) Backup heat bypassing storage, Q2D	kWh	-	216	851	2366	1879	386	0
H) Solar thermal output to hot storage, Q1	kWh	-	1321	1185	1105	2320	3037	2574
I) Solar irradiation on total collector aperture area, Q_sol	kWh	-	3790	3031	3833	6131	16185	14849
J) Electricity consumption (overall)	kWh	-	1784	1457	2105	1889	1978	1537
J.1) Electricity consumption (total)	kWh	-	212	317	202	513	308	248
K) Electricity backup chiller, E12	kWh	-	113	265	122	422	93	0
Collector field efficiency, H/I	%	**-**	**35%**	**39%**	**29%**	**38%**	**19%**	**17%**
Total Electrical COP (*), (A+B+C)/J1	-	**-**	**31,05**	**19,75**	**47,41**	**21,40**	**25,43**	**17,30**
Chiller average COP, E/K	-	**-**	**3,47**	**4,21**	**3,69**	**3,66**	**3,80**	**-**
Winter Solar Fraction, H/(H+F+G+K)	%	**-**	**0,80**	**0,51**	**0,31**	**0,50**	**0,86**	**1,00**

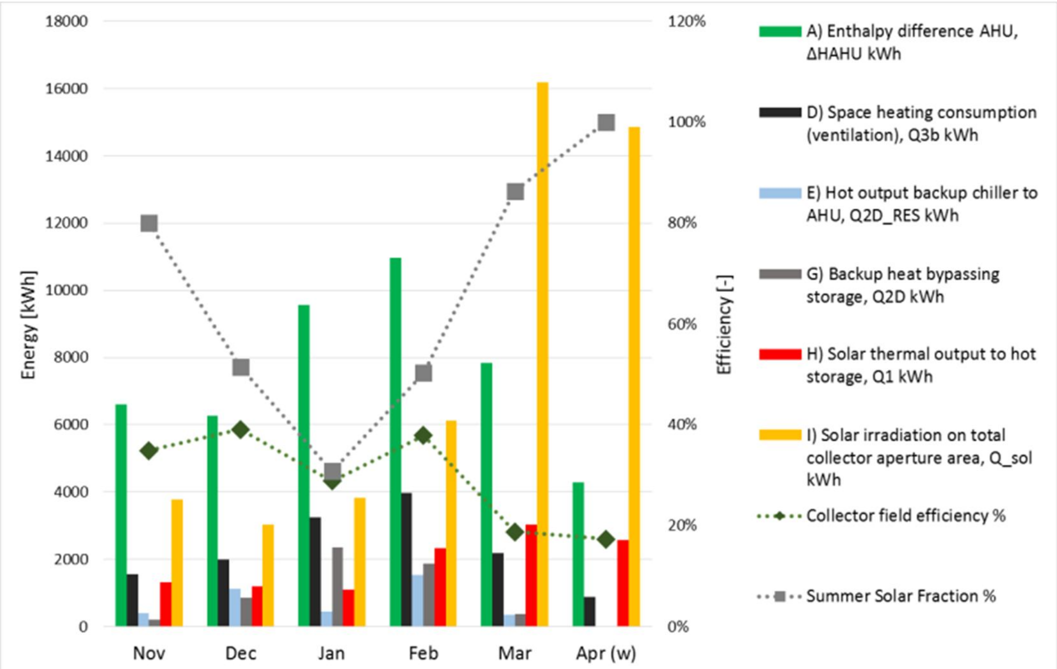


Figure 62: Summary of heating season performances

(*) The total electrical COP takes into account the total electricity consumption as defined in Task 38 (i.e., the electrical fan consumption was excluded but power for the extra pressure drop due to rotor, regeneration ducts, extra heating coils, and static heat exchangers were included). This allows for comparison of the system's electrical effectiveness to other systems, because the fans are present in each possible system configuration.

Data was not presented for August, September and October as a necessary of maintenance period effected the results.

The performance results for the SDEC system were quite low when compared with other DEC applications. The reasons for this were as follows:

- The static heat exchanger had a lower effectiveness compared to a rotary heat exchanger;
- Using external air for the regeneration process instead of return air (heated up by the heat exchanger) required an increased thermal energy to reach the regeneration temperature.

The COP_t was sensitive to the passive process effectiveness (thus, the supply cooling coil is excluded). In the POLIMI system's case, this effectiveness was partly sacrificed by not using a rotary exchanger, to reach high air quality and to avoid fresh air contamination. Furthermore, we have to consider that the thermal energy input was produced by renewable solar energy and a recovered flux (i.e., the heat rejected by the HP's condenser).

The primary energy consumption was calculated both for the SDEC system and for a reference system (designed following Task 38's conventions for a comparative system). It consists in a standard air handling unit, with a cooling condensing coil powered by a traditional chiller (SPF 2.8) and a heating coil heated up by a gas boiler. The AHU is made of two channels with a sensible static heat exchanger. The Task 38 Monitoring Procedure defined four possible scenarios based on the post-heating operation of the reference AHU. For this application the 3rd scenario was selected, where the supply air conditions were exactly the same for both systems. The fractional savings (f_{sav}=1-PE_{SDEC}/PE_{ref}) were calculated for the whole year of operations and presented in Figure 63 and Figure 64.

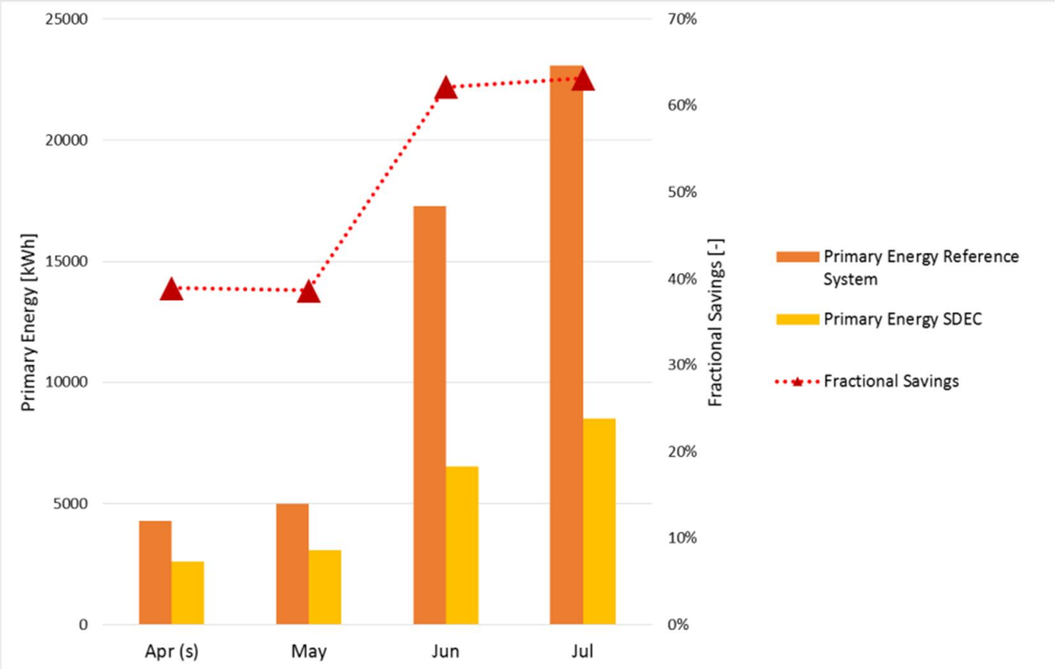


Figure 63: Primary energy consumption and fractional savings of the SDEC system over a reference system during summer

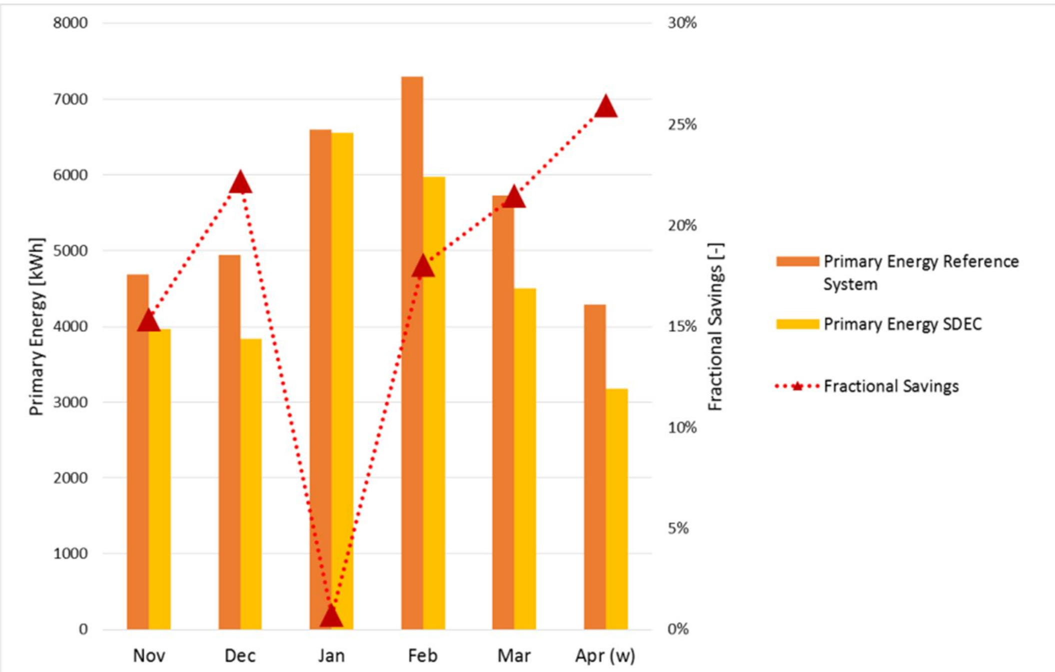


Figure 64: Primary energy consumption and fractional savings of the SDEC system over a reference system during winter

Despite the incomplete data set collected during the summer, a high savings potential was found (52% compared with a conventional AHU). During the winter, the savings potential strongly depended on solar availability as it allowed for heating without the use of the HP. When the system operated using the gas backup, the SDEC's rate of primary energy consumption was higher than the conventional AHU. This was due to the higher pressure drop in the supply and return air channels. For example, during January, 73% of the space heating power consumption was covered by the backup mode, which resulted in poor fractional savings.

The complexity of the system required additional performance monitoring systems to be used in conjunction with those recommended by the Task 38 report. The 2014 summer analysis indicated that the solar collectors and the heat pump's operation were consistent with the manufacture's performance specifications. The sorption wheel, however, showed a significant underperformance. This contributed greatly to the overall system's underwhelming performance[16]. This underlines the commissioning procedures importance and the central role played by the rotor (see section 5.3.4).

CUSTOMER AND OCCUPANT SATISFACTION

The facility's management had the following opinions on the system:

- This SDEC system represented a source of pride due to the implementation of green technologies, the use of renewable energy and the research interest associated with the SDEC system;
- The energy bill was lower even with the existing performance limitations and the planned system improvements (e.g., the extra solar energy dissipated by the anti-stagnation mode could be used to pre-heat DHW during the summer and middle seasons);
- The SDEC plant was much more complicated than the conventional systems that the facility staff was accustomed to. This introduced some logistical, technical and knowledge based issues which were not easy to solve. This problem was addressed through the development of an information transfer procedure that helped to eliminate the knowledge gap (this included the production of a "SDEC manual", predefined maintenance sheets and simplified monitoring tools).

Comfortable conditions for the building's occupants were not met during the initial monitoring period. This was due to heavy maintenance which affected the air conditions.

CONTROL STRATEGY

Figure 55 describes the control strategy approach. During the operational phase, the original control strategy was tuned in order to better satisfy the occupants requirements.

16 Frein, Aprile, Muscherà, Scoccia, Motta. A continuous commissioning analysis and its application to a new installed solar driven DEC system coupled with heat pump. IEA SHC 2014.

Temperature Control

The temperature set point was dependent on the external air conditions; however, the set-point range was limited to 20°C to 24°C.

Free cooling Mode

- When the outdoor air conditions were appropriate, no air treatment was required and the rotor and heat exchanger were bypassed.

Heating Modes

- Heat Recovery: the external air was heated by the exhaust air passing through the sensible heat exchanger. The heat exchanger bypass ratio was PI controlled to adjust the temperature and the sorption wheel was completely bypassed.
- Active Heating: when the external conditions were cold enough to require additional heating, the hot water coils were activated using the following three heat sources in sequence:
    - Direct Solar: the hot water came from the solar hot water storage tank;
    - Active Heating (HP): when the solar system was unable to provide sufficient heat, the HP was activated. The HP could make use of water preheated by the solar system and by the return air;
    - Back Up: when there was still insufficient heat, the backup natural gas boiler heated the water

In all Active Heating modes, the mixing valve for the heating coil was PI controlled to reach the set point temperature.

Cooling Modes

- Indirect Evaporative Cooling: the return air was first evaporatively cooled by the humidifier, and then used to cool the supply air in the sensible heat exchanger.
- Active Cooling: when additional cooling power was required, the heat pump was activated to provide chilled water to the cooling coil. The heat produced at the condenser side is rejected through the return coil or can be used to pre-heat the regeneration airflow.

Humidity Control

The operational humidity set point was fixed at 9,5 g/kg (a slightly lower value than the design value of 10.5 g/kg), in order to maintain comfortable indoor conditions.

Dehumidification mode: when dehumidification was required, the wheel bypass was closed and the sorption wheel was activated. The thermal energy needed to regenerate the wheel was drawn from the solar storage tank (if available) and the heat pump’s condenser (if the heat pump was in the Active Cooling mode).

Safety Control

The control strategy incorporated several additional commands to extend the operational life of the system. These additional rules included the following:

- Antifreeze control: when the external air temperature fell below a certain limit, the circulating pumps were activated at low flowrate (if not already active);
- Heat pump transition mode: used to dissipates heat from the heat pump's condenser to prevent the heat pump's compressors from overheating;
- Collector anti-stagnation mode: during times of high solar irradiance and low dehumidification or heating loads, the hot water in the tank was circulated to the regenerator heating coil and the regeneration fan was activated.

IMPROVEMENTS / OPTIMIZATION MEASURES

The continuous performance monitoring procedure allowed any errors or problems with the systems operation to be quickly identified and corrected. The most significant examples of this are described in the following sections.

Heat pump

The compressor's internal controller was reducing the heat pump's performance by excessively cycling itself on and off. An external function was developed to modify the compressors activation cycles based on the outlet water temperature. This procedure resulted in a significant improvement to the EER values. Two similar days were evaluated, shown in Table 16, to show these performance improvements. The "average ON time" represented the average length of the HP's compressor activations. This value was greatly increased. This data is further confirmed in Figure 65.

Table 16. Performance improvement due to the implementation of the new compressors control scheme

Date	Cooling energy	Electrical energy	Daily EER	Average ON time
	kWh	kWh	-	min
30/08/2013	242	58	4.17	12.2
23/07/2014	232	49	4.73	53.2

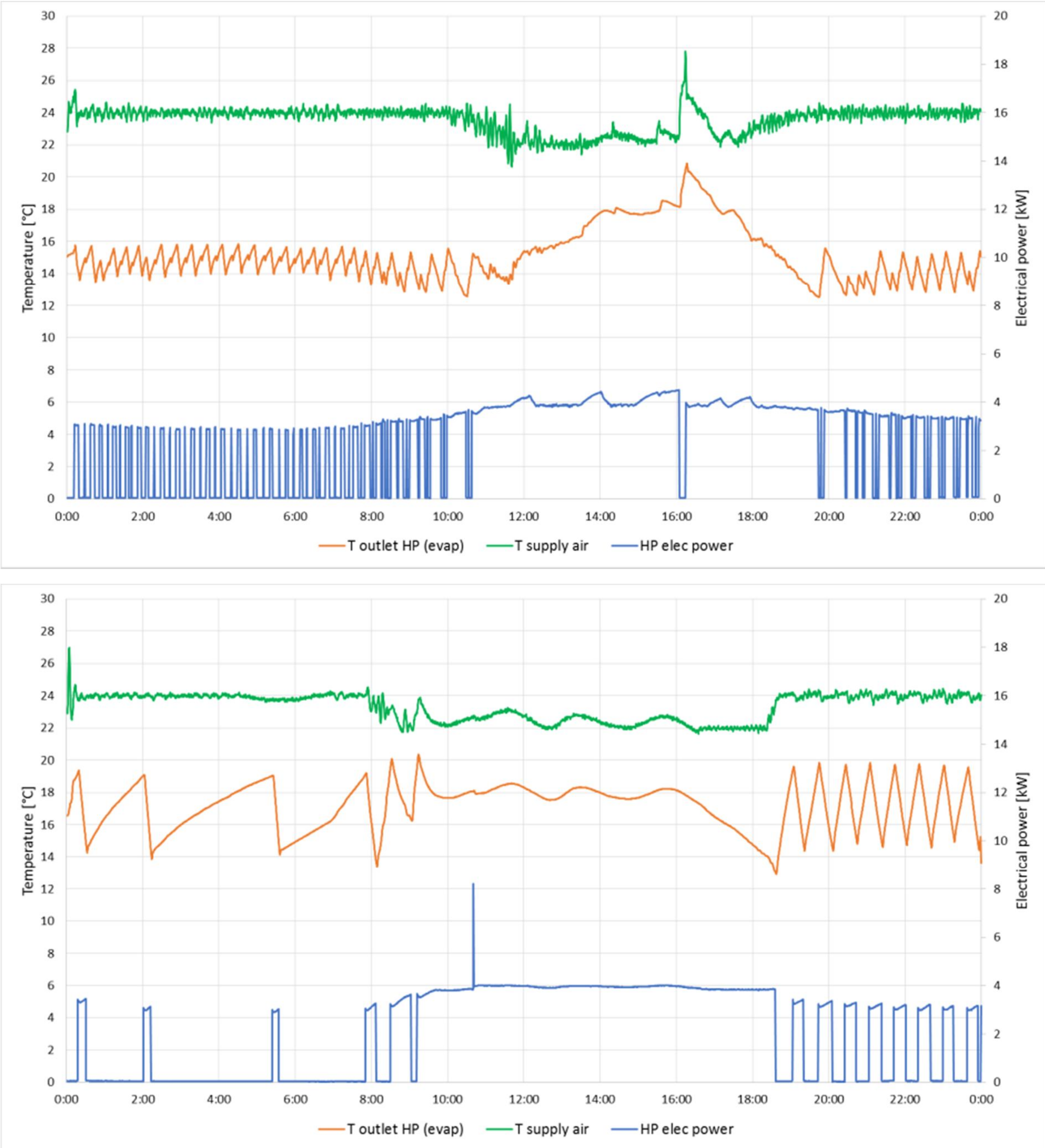


Figure 65: The heat pump behavior before and after the control modification

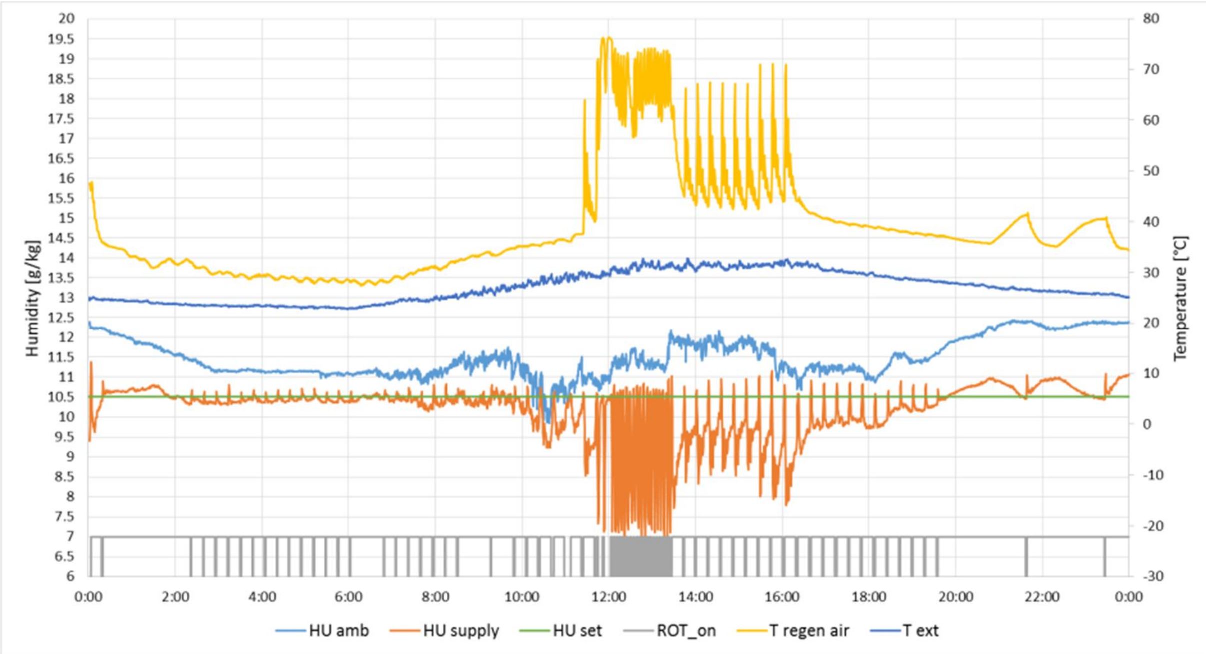
Sorption Wheel

The rotor's operation was irregular. This caused the humidity of the supply air to fluctuate. The difficulty was caused by rapid fluctuation of the supply conditions as the system tried to maintain the set point conditions. The humidity set point was decreased from 10.5 to 9.5 g/kg to allow for longer operational run times. The minimum regeneration temperature was also reduced. These changes resulted in smoother operations of the wheel and in consistent supply air humidity.

The comparison of the sorption wheel's run times before and after the modifications on two days with similar conditions is shown in Table 17. A more detailed analysis is shown in Figure 66.

Table 17. Operational run time of the sorption wheel before and after the control modification

Date	Average ON time
	min
05/09/2013	17.0
13/06/2014	106.5



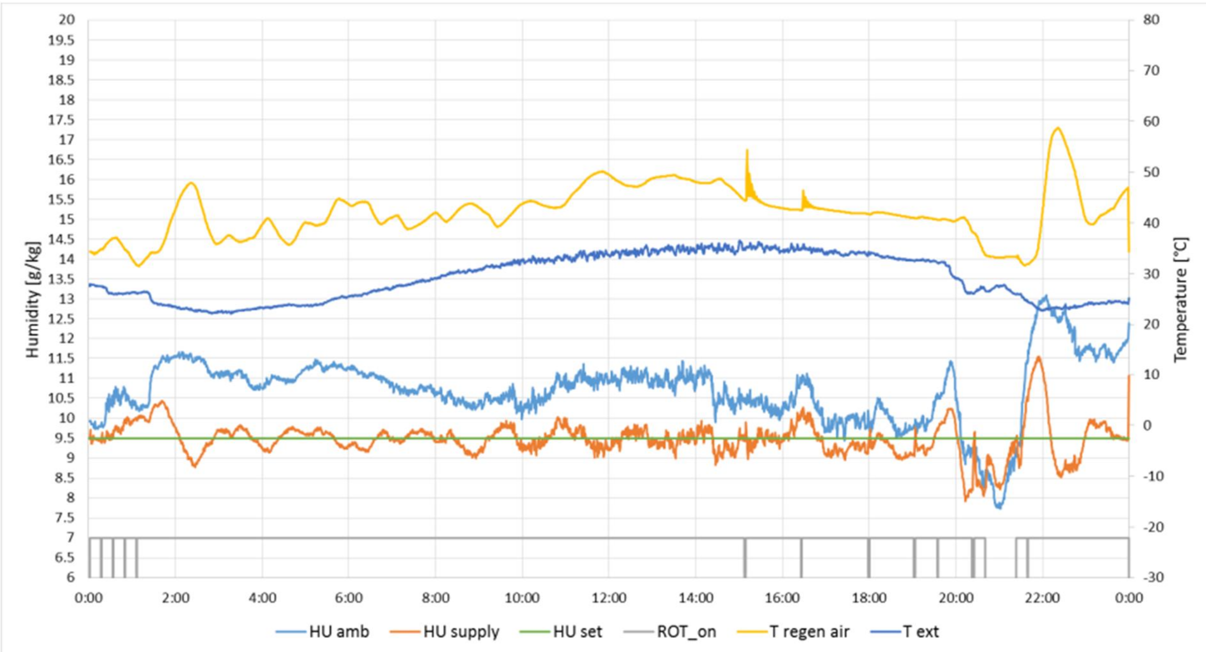


Figure 66: Sorption wheel behavior before and after the control modification.

A possible improvement for the POLIMI SDEC system could be the integration of the solar system into the DHW system. This could reduce the collectors' stagnation time during the middle seasons and improve the solar energy utilization. Improvements to the desiccant wheel could significantly increase the performance figures and the energy and cost savings.

5.3.4. KEY FINDINGS / LESSONS LEARNED/ RECOMMENDATIONS

The first monitoring period was used to gain experience and to identify trouble areas with the new HVAC system and its integration into an older building. This experience can be used by designers and engineers to help determine if a similar system could be applicable for their own projects. These recommendations will hopefully allow the designer to start the design and operational phases at more advanced levels and avoid making errors due to a lack of knowhow and experience.

- **MEASUREMENT ISSUES**

A well-designed sensor suite must be used to monitor the operation of the system. While the water streams are easy to monitor, the air flows may require more effort to ensure accurate measurements while keeping costs low. The following references are recommended to aid in the design of the data acquisition system:

- The Dehumidification Handbook, Munters Corporation, 2002.
- S.J. Slayzak, J.P. Ryan, NREL/TP-550-26131: Desiccant Dehumidification Wheel Test Guide, National Renewable Energy Laboratory, 2000.

- **MAINTENANCE ISSUES**

The complex and unconventional equipment used by the plant introduces the need for a continuous monitoring procedure, especially for the first operational period. Simple and effective tools are required to clarify the necessary procedures for the people working with the system. Furthermore, due to the large number of involved components, it is necessary to set up a rigid maintenance program. For example, regular air filter changes are crucial to avoid extra electrical consumption due to the extra pressure drop. It is very important to notice that these simple maintenance operations can be more effective than complicated optimization measures. The lack of a maintenance procedure could affect the success of the system.

- **CONFIGURATION ISSUES**

A compact system layout should be used to reduce energy losses, to simplify monitoring and maintenance procedures and to minimize costs. Reducing distances between the subsystems and designing proper maintenance spaces can significantly reduce the effort required to operate the system. This can be difficult when retro fitting an existing system, as in the POLIMI case. The designer has to design around the existing space which, was not originally built to host a large SDEC system. This issue can interfere with maximizing the performance and cost effectiveness of the plant.

- **SUBSYSTEM EVALUATION**

The large number of components causes difficulties in the performance evaluation of the whole system. In the POLIMI case, as mentioned in 5.3.3, a performance evaluation procedure for each of the main subsystems (heat pump, solar system and desiccant rotor) was developed. The subsystem's performance should be monitored to ensure they are operating properly at the expected values (i.e., manufacturer supplied performance ratings). This can enable easier trouble shooting and the determination of system weak points. In the POLIMI case, particularly, the sorption wheel behavior is was found to be the most critical to the influential on the performance of the whole system operation. The rotor did not work as expected and the system's performances decreased significantly.

- **ROTOR INFLUENCE**

As discussed above, the main design/commissioning effort should be focused on the desiccant rotor subsystem. The observed underperformance of the POLIMI system's rotor significantly decreased the SDEC's energy saving potential. The lack of a clear test/certification procedure for manufacturers of this type of component (see chapter 5) makes it difficult to predict the nominal performance of the rotor during the design phase.

- **INSTALLATION MISTAKES**

During the installation phase, an oversized recirculation pump for the humidifier was mounted. This pump needlessly increased the system's electricity consumption and introduced too much water into the system. The performance of other sections indicated the presence of water puddles in the AHU which could produce hygienic risks. This highlights the importance of choosing and installing properly sized components.

- **FREEZING PROTECTION**

During the wintertime, some system sections (regeneration pipes and coils) were not in use and if the water had not been replaced by a water/glycol solution, these components could have been seriously damaged.

Economics

One of the principle reasons to consider the use of an SDEC system is the cost savings over a conventional system. An economic analysis of the POLIMI SDEC system has been completed to highlight this. It should be noted that this is a demonstration SDEC system that has been retrofitted into an elderly building. The pre-existing structure's layout and available space set very strong limits during the design and the construction phase. This significantly impacted the economic performance of the project. Other systems, especially when designed into the building from the start, could see improvements in this area. In order to show the saving potentials, the real situation and an ideal situation were analyzed.

The following features characterize the real scenario:

- Higher costs and greater pressure and energy losses from longer piping and ducting due to the non-optimal available space.
- Removal cost of the old HVAC system.
- High costs of the control equipment, due to the demonstration concept of the system.

In the ideal scenario, the cost distribution is quite different from the real POLIMI case (Figure 67 and Figure 68), with an investment cost approximately 30% lower than the POLIMI system. This situation is achievable through improved knowhow and the application of the lessons learned examples outlined in this document.

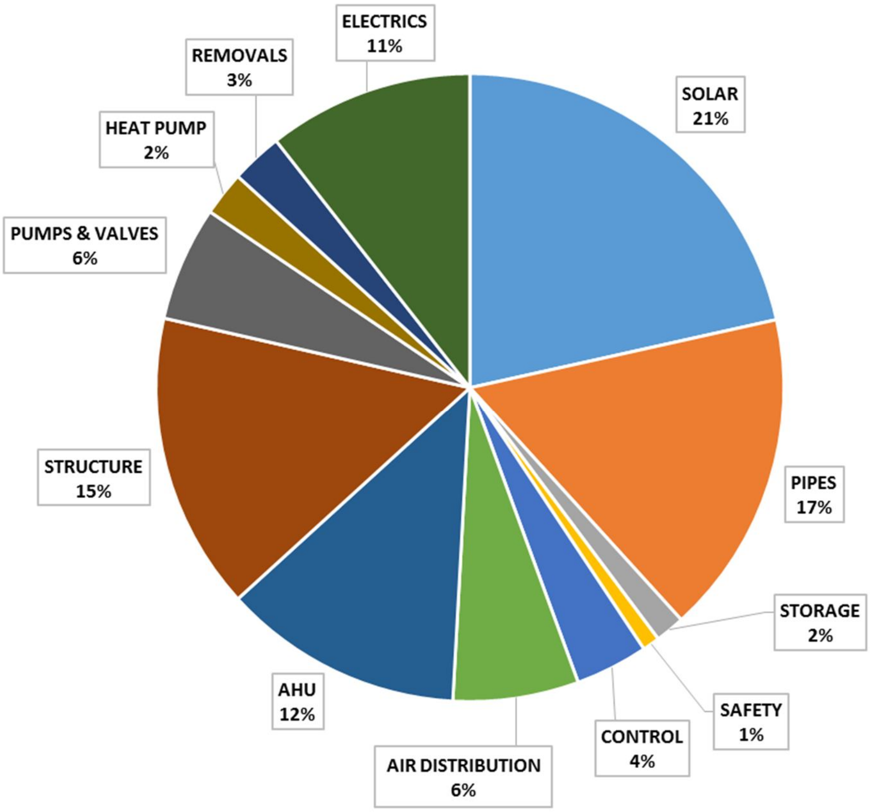


Figure 67: Investment cost distribution for the real scenario.

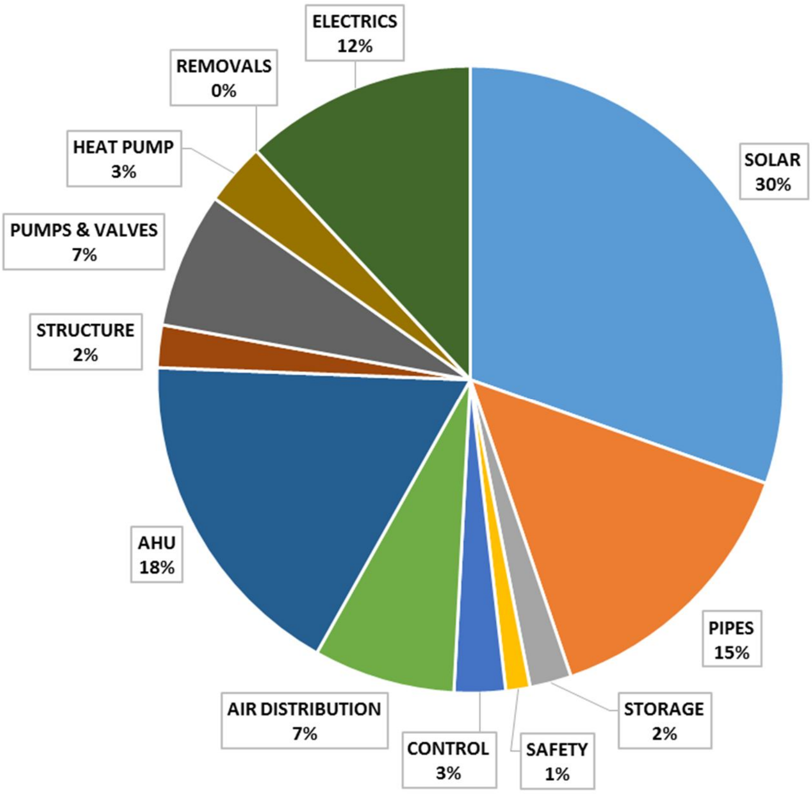


Figure 68: Investment cost distribution for the ideal scenario.

The full savings potential of an SDEC system could be reached on new buildings, where the SDEC system designer has the ability to collaborate with the building's designers to maximize cost effectiveness and to ensure comfortable conditions for occupants.

6. LIST OF FIGURES

7. LIST OF TABLES

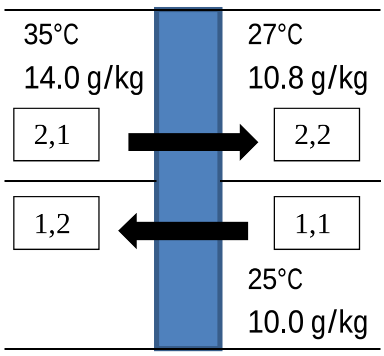
8. ANNEXES

8.1. QUALITY LABELS OF DESICCANT WHEEL AND OF SOLAR DEC

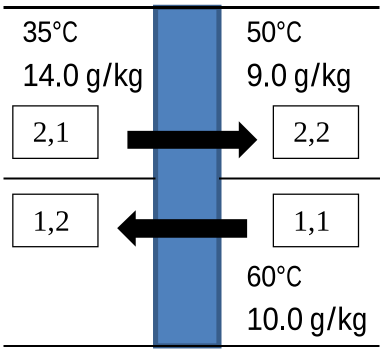
8.2. NOMENCLATURE

$\dot{V}_s$	[m^3/hr]	[cfm]	Volume air flow under standard conditions
$A_{tot} = \pi \frac{D_o^2 - D_i^2}{4}$	[m^2]	[ft^2]	The free face area between outside and inside diameter
$\dot{V}$	[m^3/hr]	[cfm]	Air volume flow
v_s	[m^3/kg$_{DA}$]	[ft^3/lb$_{DA}$]	Specific volume of air
w	[g$_w$/kg$_{DA}$]	[grains/lb$_{DA}$]	Humidity ratio
MRC	[kg/hr]	[lb/hr]	Moisture removal capacity
RE	[W]	[Btu/hr]	Regeneration heat input
RSHI	[kJ/kg]	[Btu/lb]	Regeneration specific heat input

8.3. EXAMPLES

	Examples of Nominal conditions in cooling mode	**Effectiveness**
Passive dehumification wheel		$\eta_T = \frac{27 - \mathbf{35}}{\mathbf{25 - 35}} = 0.8$ $\eta_x = \frac{10.8 - \mathbf{14}}{\mathbf{10 - 14}} = 0.8$

Active dehumification wheel



$$\eta_T = \frac{50 - \mathbf{35}}{\mathbf{60 - 35}} = 0.71$$

$$\eta_x = \frac{9 - \mathbf{14}}{\mathbf{10 - 14}} = 1.25$$