Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b:
Monitoring Results

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

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Contents

1 Introduction

This report summarizes the monitoring activities on small-scale solar heating and cooling systems carried out within subtask A of IEA-SHC Task 38. All systems were monitored according to the monitoring procedure developed with Task 38. However, the level of detail varies from system to system. The monthly monitoring data has been filled into the Monitoring Excel Tool of Task 38.

In the first part of the report, a short description of all 13 systems that were monitored is given including the characteristic data of the application and the installation, a few pictures and a short written summary of the monitoring results.

In the second part, an overview of the key performance figures of 11 of the systems is given and the results are compared with each other (For 2 systems there were not enough reliable data available to include them in the comparison). The shown performance figures include different coefficients of performance of the chillers and the whole systems as well as the reached fractional primary energy savings, the collector yields and the water consumption of the cooling towers.

Finally, for most systems, there is a more detailed report on the monitoring results available in the appendix.

2 General Description of Monitored Systems

2.1 Austria: SOLution Headquaters, Sattledt

Description of the application

Type of building office building	
Location Sattledt, Austria	
In operation since 2005	
System operated by SOLution Solartechnik GmbH	
Air-conditioned area 150 m^2	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

The solar thermally driven absorption cooling system is designed to control the air temperature of the SOLution Solartechnik GmbH headquarter office (150 m² useful area) in summer and winter. The system configuration allows a solar-autonomous cooling operation, e.g. the driving heat for the absorption chiller is only generated and provided by the solar system. A gas driven boiler allows test routines and as well a significant heating support in winter.

Since 2008 the company of SOLution Solartechnik GmbH has moved to a new headquarter building. Therefore the solar absorption cooling system has been disassembled and the system is not anymore in operation.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	15 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	EAW WEGRACAL SE 15
Chilled water application	Chilled ceiling
Dehumidification	no
Heat rejection system	Wet open (EWK 036/06 Axima)

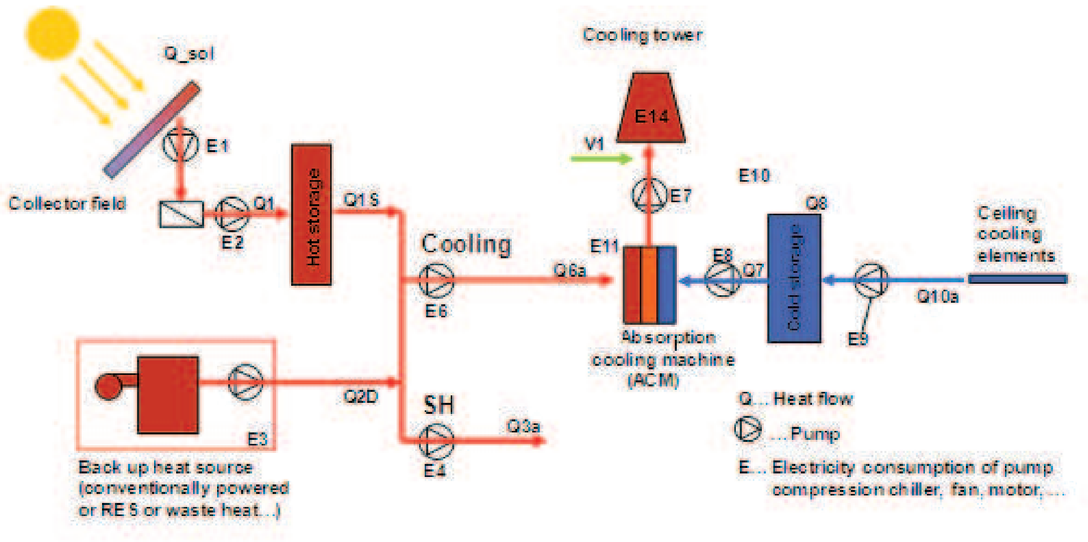
Solar thermal

Collector type	flat-plate
Brand of collector	SOLution
Collector area	40,50 m^2 gross
Tilt angle, orientation	21°, South
Collector fluid	water-glycol
Typical operation temperature	85°C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	0.8 m³ water
Auxiliary heater	Gas boiler, 9 kW
Use of auxiliary heating system	Space heating in winter
Auxiliary chiller	none

System scheme



System performance

The small-scale solar heat driven cooling (SHDC) system has been put into operation in 2005. The monitoring campaign carried out by Austrian Institute of Technology allows the assessment of the energy system performance from September 2006 till September 2008. Based on the measurement data following conclusion regarding the energy system performance can be stated:

- Around 1,026 kWh chilled water has been produced by the SHDC system during the entire monitoring period. This corresponds to around 70 full load hours. The solar system including flat-plate collector and hot water storage did provide around 2,356 kWh heat to the adsorption chiller which leads to a seasonal performance factor of SPF = 0.435. The annual electricity generation efficiency with reflects the ratio of produced energy of chilled water and the consumed electricity of the overall SHDC system is calculated to 3.39.

- The SHDC system energy performance can be distinguished between two different categories:

  - ***High chilling capacity and energy efficient system performance:*** The solar thermally driven absorption system operates with coefficient of performance COP factors corresponding to manufacturer data. Cold water temperatures are in the range of 12°C and the absor ption chiller operates in the capacity range of 5 to 15 kW_{cold}. In all three hydraulic circuits of the chiller nominal temperature level are achieved.

  - ***Low chilling capacity and inefficient system performance:*** The absorption chiller operates with COPs significantly lower than nominal COP according to manufacturer data. Especially during the starting phase the nominal chilling capacity of 15 kWcold is not achieved. The required temperature levels in all three water circuits around the chiller are insufficient and the chiller operates mainly under low part load conditions.

- Heat rejection – the applied wet open cooling tower operates efficiently und supplies sufficient cooling water temperatures to the absorption chiller. Till September 1,700 liter fresh water have been consumed this corresponds to 70 l fresh water per hour of full load operation. The open wet cooling tower did supply cooling water temperatures below 20℃ a couple of days, e.g. the direct use of this cooling water for the chilled ceiling circuit is technically reasonable.

- The solar thermal system - composed of 40.50 m² flat-plate collectors and 2000 liter stratifying hot water tank - generates sufficient driving temperatures for the absorption chiller. Typically the driving temperature starts with 75℃ and decreases to 55℃. The low flow operation strategy for the flat- plate collectors in the rage below 20 l/h m² supplies sufficient driving temperatures.

System reliability and overall success of the installation

The monitoring system did operate reliably. Some measurement data logging gaps appeared because of reboot troubles of the data logging system in case of temporary grid blackouts. The measurement signals of the pyranometer did not deliver reasonable values and the technical problem could not be solved on site.

During the monitoring period once the absorption chiller produced too low cold water temperatures which led to crystallization phenomena of the working fluid. The manufacturer did maintenance work which means mainly refilling of lithium-bromide and recreation of vacuum.

Office room air temperatures were measured as well and during the monitoring period and only some few hot days were identified with office room air temperatures exceeding 26℃. Consequently the thermal user comfort lead to a high acceptance of the office users.

Photos

	
Office building of SOLution headquarters Sattledt/ Austria	Technical room equipped with EAW WEGRACAL SE 15 absorption chiller, hot and cold water storage tank
	
Chilled ceiling device	Wet open cooling tower

	(EWK 036/06 Axima)

Monitoring Data

Measured period	09/2006 till 09/2008
Monitoring level (according to Task 38 procedure)	2
Person responsible for monitoring	Tim Selke (Austrian Institute of Technology, AIT) Phone: +43 (0)50550 6651 Email: Tim.Selke@ait.ac.at

2.2 Austria: Bachler, Gröbming

Description of the application

Absorption chiller with open wet cooling tower to cool 700 m² office building with a peak cooling load of 8 kW.

Type of building office building	
Location Gröbming, Austria	
In operation since 2007	
System operated by Bachler HVAC company (own office building)	
Air-conditioned area 700 m^2	
System used for space heating? Yes	
System used for DHW preparation? Yes	

General description of the system

Closed cycle absorption with ammonia/water as working pair. The system has biomass as auxiliary heating for heating and for the absorption machine.
The building is an office building for a HVAC company with 700 m² cooled office area. The peak cooling load of the building is 9 kW and the building is cooled via concrete core activation.
The building has mechanical building ventilation with heat recovery.
The heat rejection is wet, open EWK Axima refrigeratoin with 25 kW.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	12 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	PINK chillii PSC 12
Chilled water application	Chilled ceilings
Dehumidification	no
Heat rejection system	Wet open

Solar thermal

Collector type	flat-plate
Brand of collector	“Neuma-Solar” in Kremsmünster, Austria
Collector area	46 m^2 aperture area
Tilt angle, orientation	45°, south
Collector fluid	water-glycol
Typical operation temperature	80 °C (driving heat for cooling application)

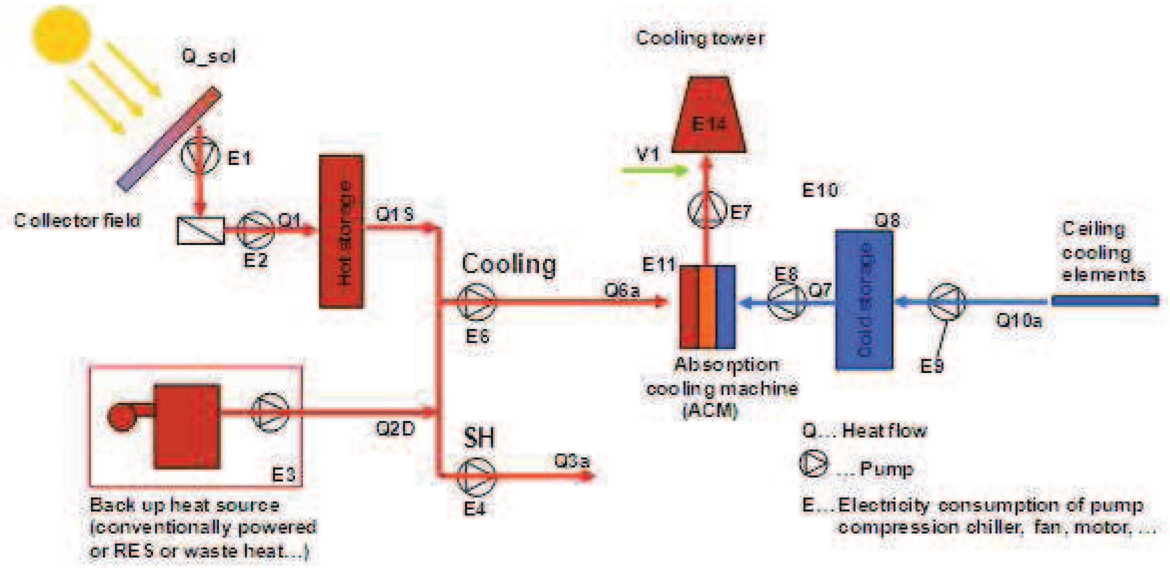
The solar collectors are partly mounted on the ground, partly installed on the façade of the building. The system is operated with a variable flow control.

Configuration

Heat storage	3*1.5m³ water
Cold storage	none
Auxiliary heater	Biomass, 150 kW

Use of auxiliary heating system	Space heating in winter, DHW and cooling
Auxiliary chiller	no

System scheme



System performance

An average thermal COP for the period between the 21st of August and the end of September of 0.565 could be reached. Electrical COPs range in daily values from 0.5 to 5 and achieve an average value of 3.1. In the period between the 11th and 18th of September no results were monitored due to a data processing problem of the computer system.

System reliability and overall success of the installation

In the time where the plant was running, it worked quite well. Nevertheless the average values over a longer time period are still improvable. In the beginning of the cooling season problems did occur due to the low volume flow of the solar cycle, therefore the mode was changed to maximum speed. Now the solar pumps are working in summer times without rpm-regulation.
During the monitoring period some problems did occur including monitoring and control system problems. The wet cooling tower works very well in the Gröbming climate. At a glance the PSC12 Pink chiller fulfilled the promised thermal COPs at the different driving, cooling and recooling temperatures. Considering the other parts of the system, such as the heating of the swimming pool, the hot storages or the district heat the complete hydraulic scheme gets overloaded and not clearly arranged at the first look.

Summarizing five main practical suggestions for improvement have been found:

- adapting the control system in order to make clear division between heating and cooling periods
- including the cooling cycle of the DHW priority rule
- excluding the logical test room temperature >24℃ of the control strategy in order to use the concrete core activation as a storage and make the control system easier

- changing the switching operations between summer and winter to automatically by including the valves V3 and V4 in the controller
- possibly rising the starting driving temperature in the control strategy if there are still problems with a clocking behavior of the chiller

Photos

	
Office building with solar collector field	Technical premises
	
Wet cooling tower and Pink chillii PSC12 in front of the building	Pink chillii PSC 12

Monitoring Data

Measured period	Aug 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Neyer (University of Innsbruck UIBK, Austria) Phone: +43 512 507-6618 Email: daniel.neyer@uibk.ac.at

2.3 Austria: SOLID Office Building, Graz

Description of the application

The office building of SOLID was renovated in 2004 and the solar cooling device was installed in the year 2008. The façade of the office rooms are south and west oriented. To reduce the solar gains external shading devices are installed at each glazing. Because of internal gains and ventilation via windows, active cooling is indispensable. The solar cooling equipment is installed in a so-called “Cooling Cabin” placed under a pergola which is situated in front of the office building as a nice main entrance. The solar collectors are installed on the roof of this pergola. The hybrid cooling tower is placed on the flat roof of the office building. The cooling load of the office rooms is taken out via ceiling cooling elements.

Type of building	office building
Location	Graz, Austria
In operation since	2008
System operated by	SOLID
Air-conditioned area	435 m^2
System used for space heating?	Yes
System used for DHW preparation?	No

General description of the system

A closed absorption cycle for generating cooling energy is employed. Autonomously solar thermal generated heat by high temperature flat plate collectors is used to regenerate the process. Within the absorption cooling machine water is used as refrigerant and lithium bromide is used as solvent. The cold water generated by the absorption cooling machine is used to cover internal and external heat gains and also the heat income caused by window-ventilation in the office rooms. In winter the solar collectors are in assistance to the space heating. In summer and winter the solar generated heat is stored in one buffer storage and all energy demand is taken out of it. The additional heat from the district heating is not stored in the tank but directly carried to the space heating system. A special application is the usage of the hybrid cooling tower for direct cooling via the ceiling cooling elements.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	17.5 kWcold, 21 kW peak
Type of closed system	Absorption
Brand of chiller unit	Yazaki WFC SC5
Chilled water application	Chilled ceilings
Dehumidification	no
Heat rejection system	Hybrid cooling tower

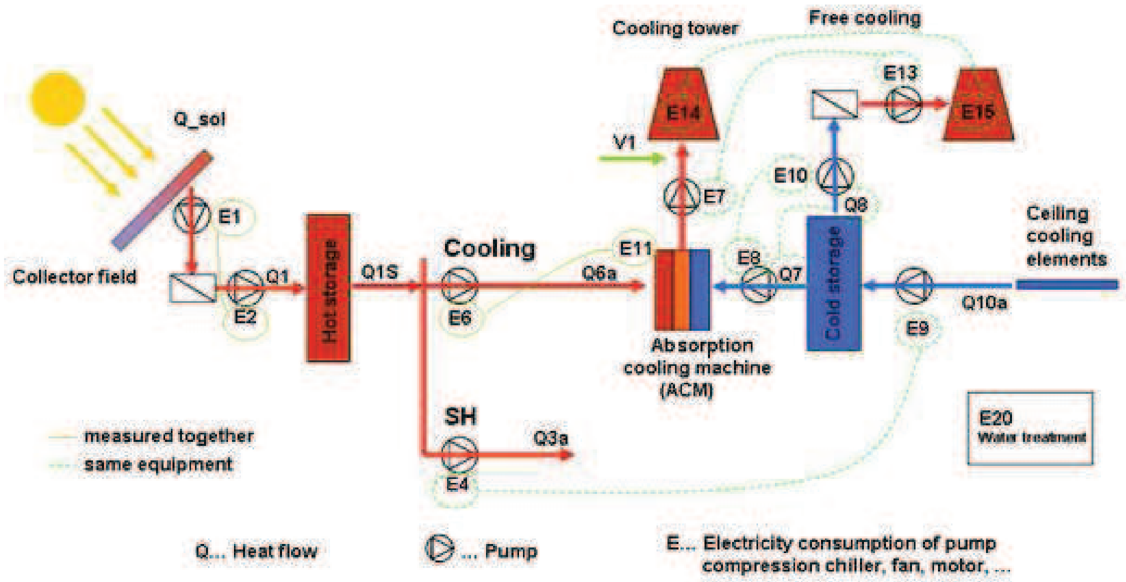
Solar thermal

Collector type	High temperature flat-plate
Brand of collector	ÖkoTech Gluatmugl HT
Collector area	60 m^2 gross area
Tilt angle, orientation	11°, south
Collector fluid	water-glycol
Typical operation temperature	88 °C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	0.2 m^3 water
Auxiliary heater	District heat
Use of auxiliary heating system	Space heating in winter
Auxiliary chiller	no

System scheme



System performance

Over the summer period 2009 (July-Sept) a thermal coefficient of performance of 0.6 was observed. About 1/3 of the cooling demand was delivered by free cooling, 2/3 by solar cooling. The electrical coefficient of performance for free cooling was COPel = 1.8, for solar cooling a COP_{el} = 1.6 was measured. The overall coefficient of performance for the summer period was approximately COP_{el} = 1.7.
In the winter period from October 2009 to April 2010 the average coefficient of performance for free cooling was about $COP_{el.}$ = 8.2, no solar cooling was observed.

System reliability and overall success of the installation

The whole system is running reliably but with a poor overall performance, especially the electrical consumption is very high. The chiller was running very reliably throughout the whole summer. It also brought the expected power level and reached a feasible COP as well. Free cooling mode is activated mainly in night times and running on a poor power level. The two largest electrical loads are the cooling tower and the pump towards the cooling tower. These two consumers are responsible for more than 75% of the electrical consumption. The reason for the high electrical consumption of the cooling tower is its high fan power which is not speed-regulated. Due to this a high potential in the improvement of the control system can be observed, mainly through an adequate control of the cooling tower. Small changes in hydraulic design can also decrease the electrical consumption, especially for free cooling.

Photos

“CoolCabin” placed under a pergola

Technical premises in the “CoolCabin”

Monitoring Data

Measured period	Jun 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Neyer (University of Innsbruck, UIBK, Austria) Phone: +43 512 507-6618 Email: daniel.neyer@uibk.ac.at

2.4 Austria: Municipal Administration, Vienna

Description of the application

The MA34 solar cooling system was put up in April 2009. It consists of an Adsorption chiller with dry cooling tower to cool the office building with a peak cooling load of 7.5 kW. The hybrid cooling tower and the solar collectors are placed on the flat roof of the building.

Type of building office building
Location Vienna, Austria
In operation since 2009
System operated by
Air-conditioned area
System used for space heating? Yes
System used for DHW preparation? No

General description of the system

Main item of the plant is an adsorption chiller with a nominal cooling capacity of 7.5 KW, type SOL ACS 08. The heat rejection is done by a dry cooling tower, type RCS 08, with EC-fan technology and an additional fresh water spraying system. The drive heat for the chiller is generated by 12 universal panel collectors with a total gross surface of 32.40 m ². The flat-plate collectors are installed in approximately 40 ° inclination and south orientation on an existing roof. In the concept no reheating is intended by other boilers. The solar heat is stored in a 2000 l solar buffer - the heat transfer from the primary to the secondary solar cycle is made by a layer loading unit. In the cooling mode the chilled water is stored in a water buffer with the capacity of 800 l. Depending on the cooling requirement of the different rooms the available fan coils are supplied by the water tank. In winter mode the hydraulic interconnecting makes a solar thermal support for heating purposes of the areas possible.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	Sortech AG / SOL ACS 08
Chilled water application	Fan coil
Dehumidification	no
Heat rejection system	Dry cooling tower with water straying system

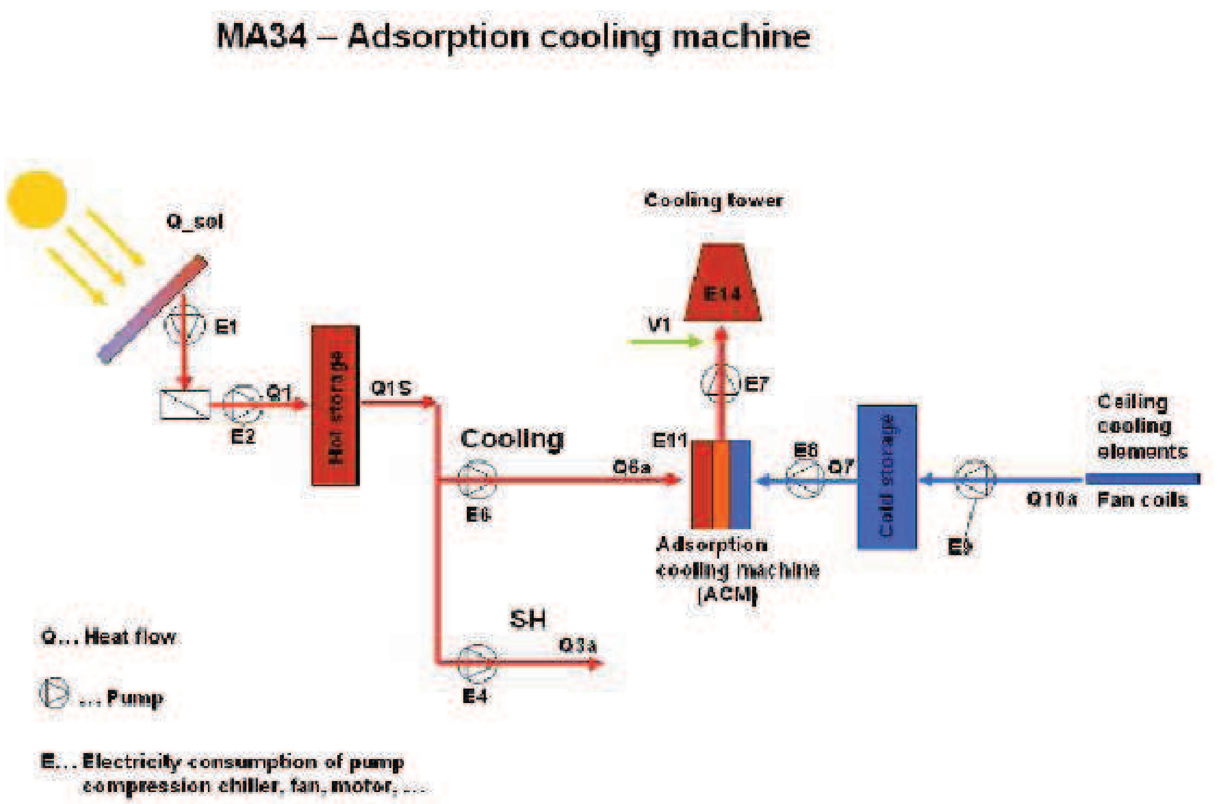
Solar thermal

Collector type	flat-plate
Brand of collector	SOLution
Collector area	62.4 m^2 gross area
Tilt angle, orientation	40°, South
Collector fluid	water-glycol
Typical operation temperature	65°C – 70°C driving temperature for chiller operati on

Configuration

Heat storage	2 m^3 water
Cold storage	0.8 m³ water
Auxiliary heater	None
Use of auxiliary heating system	--
Auxiliary chiller	None

System scheme



System performance

The solar cooling system worked steadily in summer 2009 and 2010 without major failures concerning the whole system or individual components. It was possible to keep the office room temperature below 24 °C. The average daily thermal COP kept with 0,3 in the same range in summer 2009 and 2010, only when the heat rejection temperature going into the adsorption cooling chiller was below 27°C the nominal thermal COP of 0,56 was reached. The average daily electrical COP increased from 2,3 in August 2009 to 4,00 in August 2010. The highest part of the electricity demand (50-60 %) was caused by the fans of the heat rejection and the heat rejection pump. As the heat rejection pump was changed to a variable speed high efficient pump in September 2010 and an adaptation of the whole control system will be implemented before summer 2011 the electrical COP should be better in the next cooling season. The solar thermal system worked very efficiently in both summers using only around 3 % of the electricity demand. The cooling capacity used from the Fan-Coils varied between 2,2 and 3,2 kW which caused a continuous part load behavior in the adsorption cooling system.

System reliability and overall success of the installation

In summer 2009 the monitoring system didn't work properly which led to many gaps in the monitoring recording. This problem was solved between summer 2009 and summer 2010. It was already possible to implement improvements between the first and the second summer by variation of the set points for the desorption temperature and the cold water temperature as well as by inventing the water spraying mode in the heat rejection device. The biggest electricity consumers are still the fans in the heat rejection which couldn't be reduced essentially by using variable speed control of the fans in summer 2010. An adaptation of the whole control system also taking load dependent operation into account will be the next step for improvements. Overall a know-how gain concerning the interrelation between the several components of a small scale adsorption system during operation to identify improvement strategies was given.

Photos

Monitoring Data

Measured period	April 2009 - ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Anita Preisler (AIT, Austria) Phone: Email: anita.preisler@ait.ac.at

2.5 Austria: Office Building Feistritzwerke, Gleisdorf

Description of the application

The office building of the Feistritzwerke STEWEAG GmbH, a municipal energy and water supplier, was renovated in 1995 and was equipped with a solar heating and cooling device in June 2010 for air conditioning both floors of the office building. Before that, no air conditioning system existed except a conventional split unit for a special server room. Air ventilation was and is done via manual window openings. The long back side of the office building is oriented +236° (south is 0°) and severa l windows are equipped with external, manually operated shading devices. Cold water is generated by an ammonia-water absorption chiller and is distributed via chilled ceilings. Space heating occurs with those same new installed ceiling elements and with existing radiators. Some technical installations already existed in this building and were used in the new heating and cooling concept. Five heat storages with 2 m³ volume each and heat generation devices like a condensing natural gas boiler, three combined heat and power plants (CHP) powered by vegetable oil and natural gas as well as a high temperature ground source heat pump were integrated in the system.

Type of building	Office building
Location	Gleisdorf, Austria
In operation since	15.06.2010
System operated by	Feistritzwerke
Air-conditioned area	1,000 m²
System used for space heating?	Yes
System used for DHW preparation?	Yes

General description of the system

Solar thermal energy is produced by a 64 m² collector field and is stored in five 2 m³ heat storage tanks. One of these tanks is switched in series to the other four tanks and is working as a high temperature tank, equipped with a stratifier unit and can be operated independent of the other four tanks. They are switched in parallel to each other and are used to store surplus heat generated by the solar collectors or the CHPs.

During the heating season the three CHP's, the condensing natural gas boiler and the heat pump serve as backup. Except the Feistritzwerke office building also a local district heating net has to be supplied with heat.

The absorption chiller's generator heat is taken out of the high temperature tank. The cold water generated by the chiller is transported directly to the cold distribution system without any cold water storage in order to avoid an extra cold water pump to reduce the electricity effort. A dynamic cooling power control configuration is achieved for the chiller. The mass flows and relevant temperatures of the particular hydraulic cycles of the chiller (generator-, recooling- and cold water cycle) are varied in that way, that an infinitely variable control of the produced cooling power is possible. Heat rejection of the absorption cooling process is realised with an open, wet cooling tower combined with an electrolytic water preparation device with the advantage of reduced electricity consumption.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	19 kW_{cold}
Type of closed system	Absorption (Ammonia-water)
Brand of chiller unit	PINK
Chilled water application	Chilled ceiling
Dehumidification	No
Heat rejection system	Wet, open

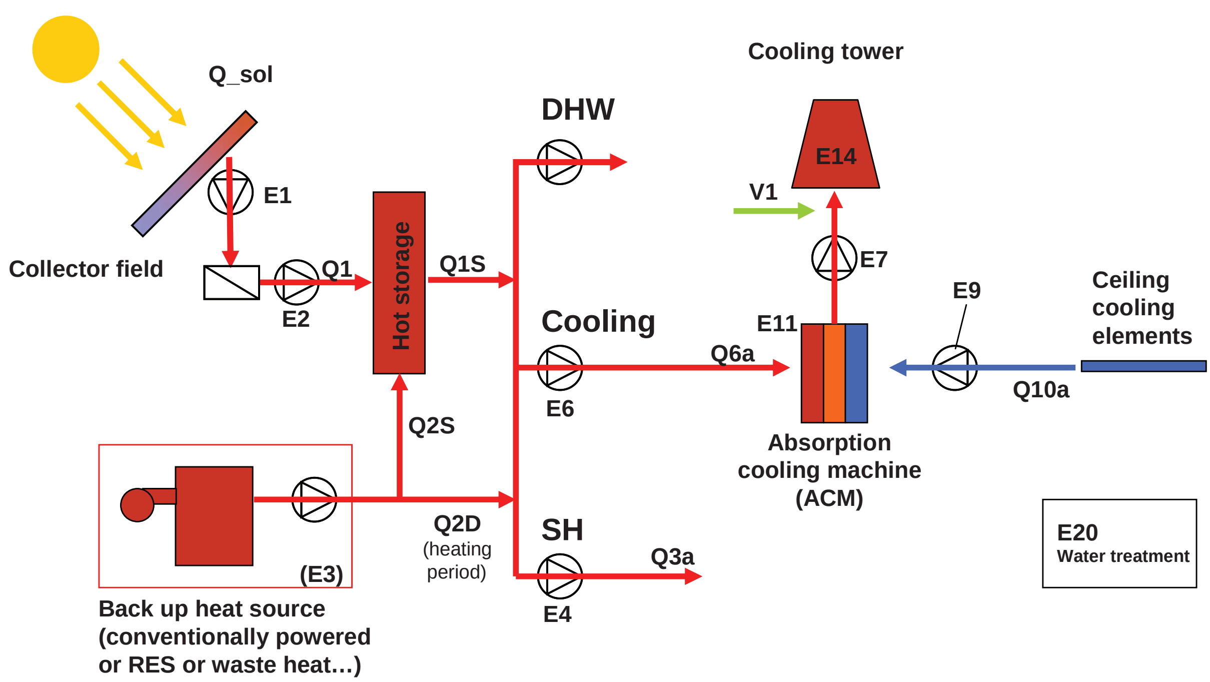
Solar thermal

Collector type	Double glazed flat plate collectors Gluatmugl HT
Brand of collector	ÖkoTech
Collector area	64 m²
Tilt angle, orientation	40°, 34° east
Collector fluid	Water-glycol
Typical operation temperature	85°C

Configuration

Heat storage	5 x 2 m³
Cold storage	None
Auxiliary heating support	If insufficient solar energy is available (into heat storage)
Auxiliary heater	3xCHP (35+18+16 kW_{th}), natural gas boiler, heat pump
Auxiliary chiller	Split units in individual rooms like the control room for the local electrical grid

System scheme



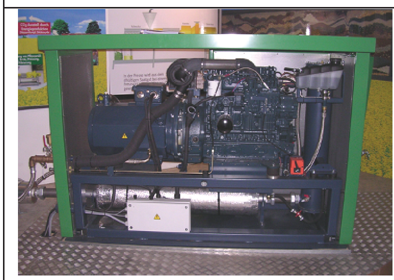
System performance

The monitoring data showed thermal coefficient of performances (COP_th_chill,m) of 0.55 and 0.53 as well as electrical coefficient of performances (COP_el_coldSol,m) of 5.75 and 5.23 in July and August 2010.

System reliability and overall success of the installation

Due to the fact that detailed monitoring data was not available for the whole first cooling period, no concrete statement to the system reliability can be given. But the fragmented measurement data showed good operation behaviors of the solar thermal system and the cold distribution system. Additionally the feedback of the Feistritzwerke's employees was pretty positive in view of room comfort conditions. A huge optimization potential could be detected in the storage charging control strategy. Currently, all five heat storage tanks are charged (Solar, CHP) - with resulting storage losses of 65% respectively 59%! Improvements will be reached by operating only the one high temperature tank. Nevertheless, for the circumstance that this system was installed and put in operation with no carried out optimization measures, the monitoring data showed quite acceptable values.

Photos

		
Absorption chiller	Chilled ceiling, water treatment	Cooling tower
		
Vegatable oil driven CHP	High temperature storage tank	External plate heat exchanger of the solar thermal system

Monitoring Data

Measured period	July 2010 - ongoing
Monitoring level (according to Task 38 procedure)	Level 3
Person responsible for monitoring	Alexander Thür, AEE INTEC Phone: +43-3112-5886-26 e-mail: a.thuer@aee.at

2.6 France: Résidence du Lac, Maclas

Description of the application

The targeted building welcoming the solar cooling application is the Résidence du Lac, a building dedicated to retired people. This building is located in the small town of Maclas in the Rhône Alpes area, close to Lyon. The town is in altitude, nearly 450 m high. The building was created in the seventies and has an average quality of energy efficiency. Only one small part of the building is cooled, the leisure space/restaurant which is compulsory since summer 2003 in retired buildings. This area is of 210 m² and includes a veranda oriented in the Southern direction. Efforts were made to increase the solar protection level in the veranda by adding dark thin protection films. Till 2007, the building owner used electric compression chillers (3 monosplits). Two of them were out of order in 2007 and the management took the decision helped by the SIEL (Syndicat Intercommunal d'Energie de la Loire), to go for a solar cooling system. The owner of the system is the SIEL itself.

Type of building Retired people residence	
Location Maclas, France	
In operation since 2007	
System operated by SIEL	
Air-conditioned area 210 m^2	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

The system is based on an absorption chiller of 10 kW coupled with evacuated tube collectors. The system works as a quasi solar autonomous cooling system because only a small electric compression chiller (split type) is used in case of failure of the solar system. The load of a part of the building is based on the following scenario: cooling demand from June to mid September and heating demand from mid October to end of May. The solar system uses fan coils for the cooling and heating modes. In addition, thanks to buffer storage, the energy is valorized in the heating mode through the central heating network of the Résidence du Lac. The heat rejection system is done by a drycooler located in the northern facade of the building.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	Sonnenklima
Chilled water application	Fan coils
Dehumidification	no
Heat rejection system	dry

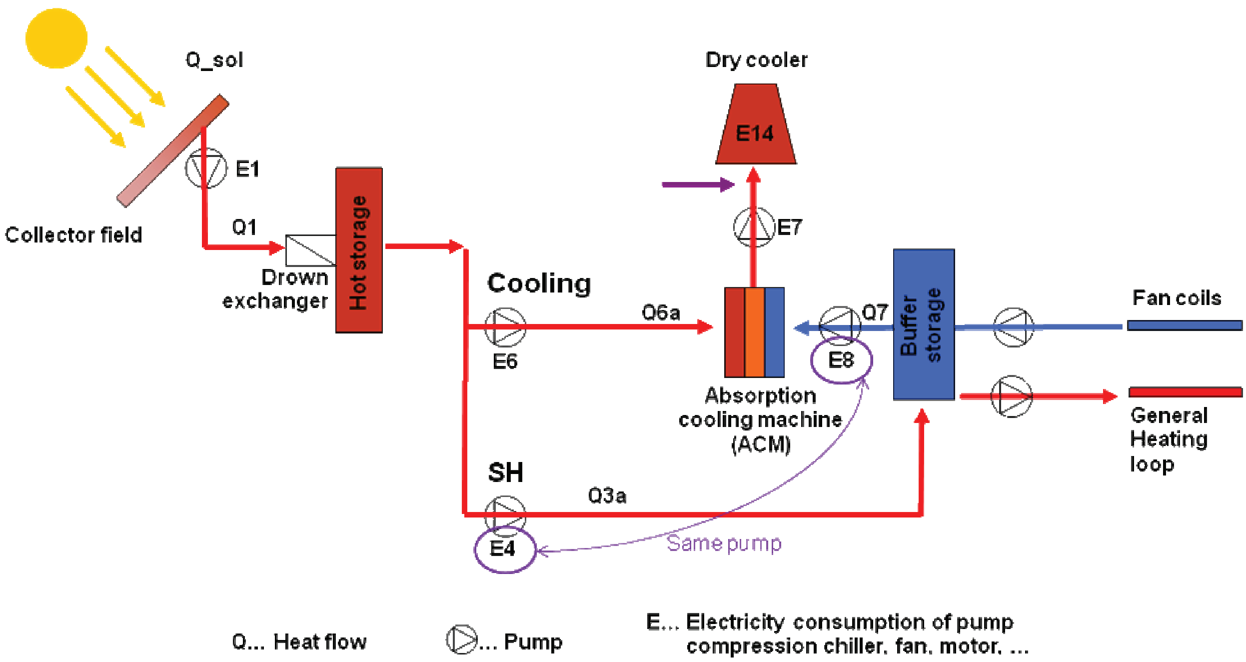
Solar thermal

Collector type	Evacuated tube
Brand of collector	Thermomax Mazdon 20
Collector area	24 m^2 absorber
Tilt angle, orientation	30°, 15° west
Collector fluid	water-glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	0.5 m^3 water
Cold storage	Buffer water (80 liters)
Auxiliary heater	none
Use of auxiliary heating system	-
Auxiliary chiller	yes
- type	el. Compression chiller
- capacity	3 kW_{cold}

System scheme



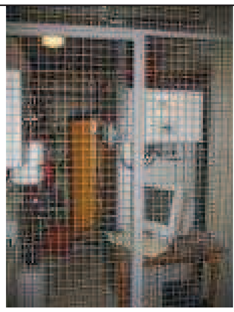
System performance

For one cooling season monitored in summer, an average thermal coefficient of performance of COPth = 0.54 was observed while the collector yield is about 40%. And the electrical coefficient of performance for the same period is about COPel = 2.5. Focusing on a day with especially good climatic conditions, the COPth can reach 0.63, and the COPel reached 3.08.

System reliability and overall success of the installation

The main issue for the Maclas installation was actually an “administrative” one. Indeed, the data transfers were a bit chaotic since they were transferred first by the chiller manufacturer, and then sent to the institution in charge of the monitoring. And because of the solvability situation of the chiller manufacturer, it was not possible to access the data to calculate performances or to check if the installation is working properly. However, this partial monitoring led to several conclusions. The strengths of the installation were: the ability to cool down the building from 12pm to 18pm (when it was needed), the compatibility with a dry cooler which avoid legionella problems which is crucial for a retired people residence, the installation had also good media coverage. This monitoring permits also to enlighten the points which have to be improved: the electric performances (COPel) and of course the monitoring system reliability.

Photos

		
Solar collector field	Technical premises	Dry cooler
		
Cooled space	Outside view of building	Fan coils

Monitoring Data

Measured period	June – September 2009
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.7 France: CNRS PROMES Research Center Office, Perpignan

Description of the application

The targeted building welcoming the solar cooling application is the CNRS PROMES research center office. It is dedicated to research works and offices in the technical area TECNOSUD of Perpignan located in Languedoc Roussillon area (South of France). The building is a large building of more than 5,000 m^2 made of 3 levels and the solar cooling system is located on the ground floor and producing energy only for a small proportion of the building. The general orientation of the building is North/South (30° tilt, 45° south east) and the collector field is oriented in the same direction on the roof. The building was created in 2000 and is of good quality level for the energy efficiency.

Type of building Office building

Location Perpignan, France

In operation since July 2008

System operated by Neotec

Air-conditioned area 180 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

The system is based on an adsorption chiller of 7.5 kW coupled with 24 m^2 double glazed flat plate collectors. The system is producing independently energy in parallel of a general multi split compression chiller system. The distribution system for the solar cooling system is an independent chilled/hot water network using fan coils working at 14/18 °C temperature level. The heat rejection is carried out by a drycooler assisted by a spring water spraying device, only used in case of very hot days.

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption
Brand of chiller unit	SORTECH
Chilled water application	Fan coils
Dehumidification	no
Heat rejection system	dry cooling tower with adiabatic spaying

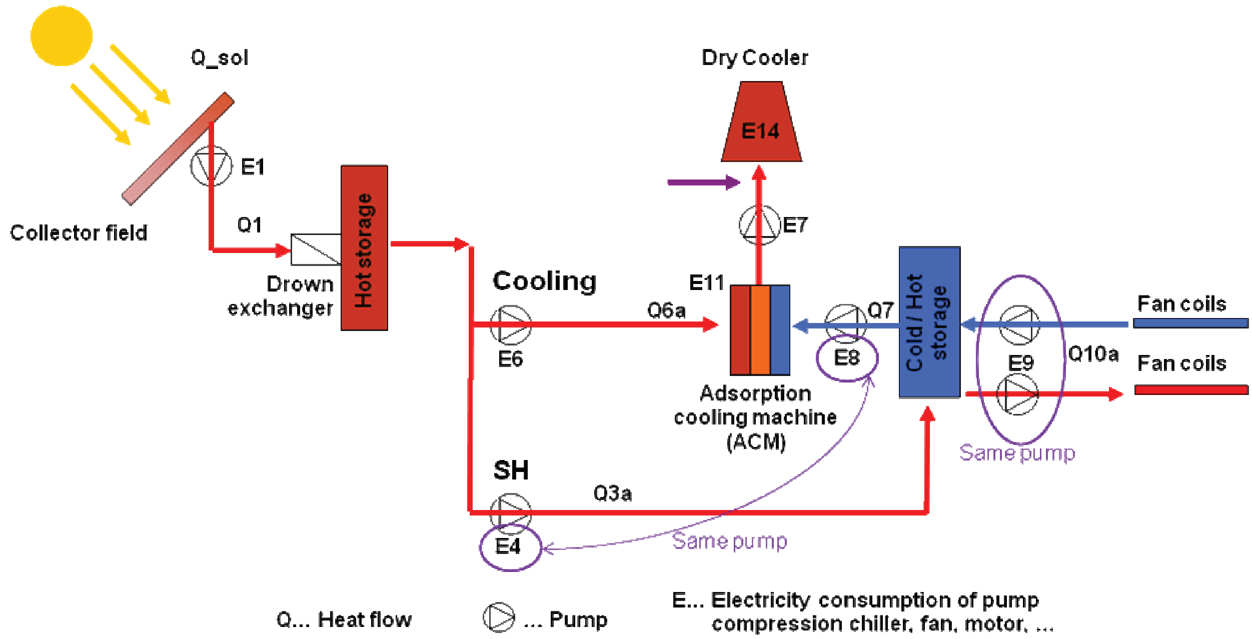
Solar thermal

Collector type	Double glazed flat plate collectors
Brand of collector	Schüco
Collector area	25 m^2 absorber
Tilt angle, orientation	30° tilt, 45° east
Collector fluid	water
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	0.3 m^3 water
Cold storage	0.3 m^3 water
Auxiliary heater	none
Use of auxiliary heating system	-
Auxiliary chiller	Yes, el. Compression chiller

System scheme



System performance

The energy production for the monitored season 2008-2009 was:

– Cooling: 2 500 kWh/year (5 months on 12)

– Heating: 4 000 kWh/year (6 months on 12)

Energy savings:

– Cooling & heating: 10 ç€/kWh (ESEER = 2; average quality multisplit split)

– Electricity consumption: 580 kWh = 58 €/year

TOTAL= 540 €/year (on the basis of an average increase of energy price of 5%/year)

System reliability and overall success of the installation

The system has been working properly for more than 1.5 years on cooling and heating mode. The overall electrical COP (electrical efficiency of the solar system) has reached an average of 10 on a yearly monitoring duration. The building owner is satisfied by the solar cooling and heating system. However, nearly 10% of the annual solar resource cannot be valorized because the targeted building has no need of heating nor cooling in April (example in 2009) however, the drainback system on the collectors' field is protecting the system from overheating risks.

Photos

		
Solar collector field	Building	

Monitoring Data

Measured period	2008-2009
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.8 France: INES Research Center Offices, Chambéry

Description of the application

The targeted building welcoming the solar cooling application is the PUMA3's INES research office. The INES (National Institute of Solar Energy) was created in 2006 by the public institutions to promote and develop solar technologies in France. To reach these objectives, the INES is divided into two centers: research, development and innovation on the one hand, and training and education on the other hand. The INES is located in the "Savoie Technolac" area which is very close to Chambéry in Rhônes-Alpes area (close to the French Alps and Lyon). The PUMA3 building is large, but only 3 mezzanine offices are cooled down by the solar system. The building was created recently so it has a good level of energy efficiency.

Type of building Office building
Location Chambéry, France
In operation since April 2009
System operated by INES
Air-conditioned area 21m²
System used for space heating? Yes
System used for DHW preparation? Yes

General description of the system

The system is based on an absorption chiller of 4.5 kW coupled with 30 m² flat plate collectors. The installation cools the building in summer, heats it in winter, and it is also able to produce a small amount of domestic hot water all year long. The 400 liters heat storage tank is included in a packaged device supplied by Clipsol (SSC BlocSol RSD 120). Included in this device there is also an electric backup and all the security devices related to the heat storage. Another tank is installed in the system which is ensuring the cold storage in summer and a second heat storage in winter. The heat rejection is carried out by a horizontal geothermal field.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	4.5 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	ROTARTICA
Chilled water application	Fan coil
Dehumidification	No
Heat rejection system	Geothermal field, exchange area about 138m²

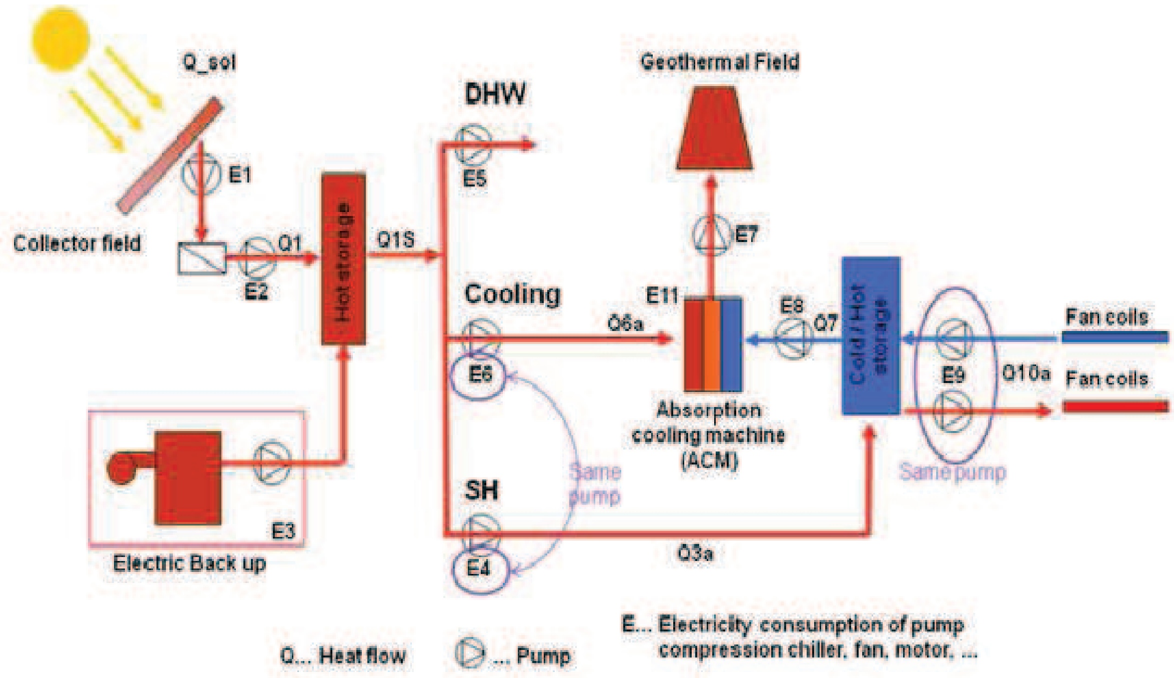
Solar thermal

Collector type	Flat plate collectors
Brand of collector	CLIPSOL
Collector area	30m²
Tilt angle, orientation	30°, 10°
Collector fluid	Water glycol
Typical operation temperature	80°C

Configuration

Heat storage	0.4 m^3 water (part of:SSC BlocSol RSD 120 CLIPSOL)
Cold storage	0.3 m^3 water
Auxiliary heater	Electric heater (part of:SSC BlocSol RSD 120 CLIPSOL)
Use of auxiliary heating system	Hot backup (heating the heat storage tank)
Auxiliary chiller	none

System scheme



System performance

For one year of monitoring, from October 2009 to September 2010, the average Electrical COP is about 5,37 which is a good value. Meanwhile, the collector efficiency was about 22%. In summer of this year, the thermal COP of the chiller was very good (high and stable) and was about 0,74 in average for the entire cooling period. Focusing on the performances of one very good and sunny day in cooling period (28th June 2010), the daily solar collector efficiency reached 32%, the thermal COP was 0,75, and the electrical COP reached 5,68. And for the heating period, the daily electrical COP can go up to 15 for very sunny winter days.

System reliability and overall success of the installation

The system has been working properly for more than 1.5 years on cooling and heating mode. The overall electrical COP (electrical efficiency of the solar system) has reached an average of 5.37 on a yearly monitoring duration. The building owner is satisfied by the solar cooling and heating system.
The main conclusions obtained regarding the performances analysis carried on are:
- Excellent performances of the chiller.
- Good behavior of the geothermal horizontal probes.
- Good performances of the collector field considering they are flat plate collectors, but a lot of days are not sunny enough to start the solar system.
- Difficulties to reach a high electrical COP due to the “experimental state” of the installation:

explained by the presence of numerous pumps, of devices consuming more electricity than usually (hot tank, chiller), and of a large set of sensors (more numerous than in a “basic” solar cooling installation).
- The average Primary Energy Ratio is higher than 1, but it’s still a bit low. It’s explained by the low electrical COP, and by the use of an electric boiler as a backup which is disadvantageous when the calculations are based on primary energy.

Photos

		
Solar collectors	Technical premises	Geothermal field

Monitoring Data

Measured period	May 2009 – ongoing
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Daniel Mugnier ; Romain Siré (TECSOL, France) Phone: +33468681640 Email: daniel.mugnier@tecsol.fr ; romain@tecsol.fr

2.9 Germany: Technical College Butzbach

Description of the application

In the low-energy building of the Technical College, demand for summer air conditioning arises due to high occupation rates and frequent use of computer equipment. The building is used throughout the summer season. Two existing ventilation systems (supply/return air with heat recovery, 1,250 m³/h air volume flow rate each) are not sufficient to remove the sensible and latent cooling loads in summer. For this reason, a solar thermally driven chiller plant was added and supported in the frame of the German Solarthermie 2000plus funding programme. The ventilation units were extended by cooling coils, and additionally chilled ceilings and cooling panels were installed.

Type of building College building, low energy building	
Location Butzbach, Germany	
In operation since 2008	
System operated by Technical College Butzbach	
Air-conditioned area 335 m^2	
System used for space heating? Yes	
System used for DHW preparation? No	

General description of the system

Two absorption chillers are installed and thermally driven with solar heat alone. The condensing boiler is used for space heating in winter only. Different operation modes are possible with the system:

- only one chilled water temperature level, e.g. for the removal of sensible cooling loads, is required. In case both chillers are in operation, the driving circuits of the chillers are usually connected in parallel;
- one chiller operates at low chilled water temperature level for supply air dehumidification and cooling, while the other provides chilled water at a higher temperature level for the chilled ceilings. In this mode, it is possible to connect the driving circuits of the chillers in series with the effect of an increased temperature difference in this circuit, resulting in a more favourable collector system operation.

The evacuated tube collector system contains pure water as collector fluid. A special freezing protection control runs the collector loop in short intervals at low ambient temperatures. No heat exchanger is installed in the total heat supply system.
System concept and overall system control: Hindenburg Consulting (www.hindenburg-consulting.com)
Detailed planning: IGT GmbH

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	2 x 10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	SK Sonnenklima: Suninverse
Chilled water application	Supply air cooling and dehumidification, chilled ceilings

	and a cooling panel
Dehumidification	in supply air system and in the cooling panel
Heat rejection system	Wet open

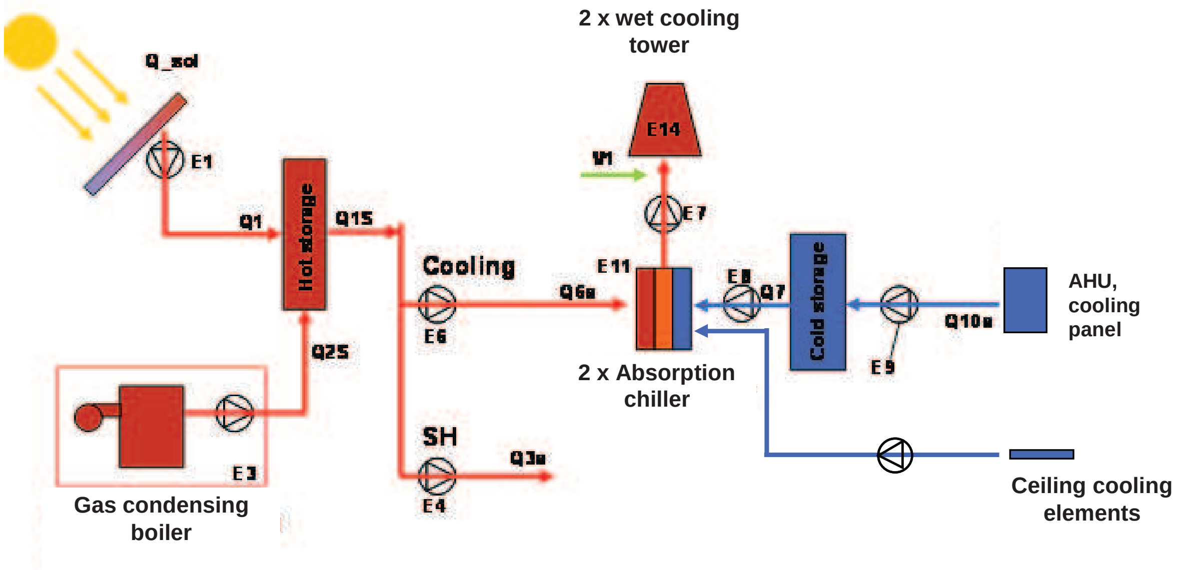
Solar thermal

Collector type	evacuated tube with CPC-mirror
Brand of collector	CPC Star azzurro
Collector area	60 m^2 aperture area
Tilt angle, orientation	30°, south
Collector fluid	water
Typical operation temperature	70 °C driving temperature for chiller operation

Configuration

Heat storage	3 m^3 water
Cold storage	1 m^3 water
Auxiliary heater	gas condensing boiler, 28 kW
Auxiliary heating support	Space heating in winter
Auxiliary chiller	none

System scheme



System performance

Data for complete monitoring period 2009:

Radiation sum horizontal	1080	kWh/m²	
Radiation sum in collector plane	1216	kWh/m²	
Specific collector yield *	337	kWh/m²	(collector output, incl. expenses for freezing protection)
Collector efficiency *	28	%	

* on base of collector output; diminished through heat return to the collector for freezing protection

Annual thermal COP of Absorption chiller 1: 0.52

Annual electric COP of Absorption chiller system 1: 4.7
(chiller, heat rejection, collector and hot water driving pump)

Annual electricity demand collector system only: 75 kWh thermal per kWh el.

Solar fraction on total annual heat supply (heating and cooling: approx. 37 %

System reliability and overall success of the installation

One of the two absorptions chillers was in the complete year 2009 not in operation, caused by a vacuum leakage. The problem could not be solved in the first year due to the insolvency procedure of SK Sonnenklima. The chiller was repaired in 2010 and since end of July 2010, both chillers are in operation.

The solar system and chiller no. 1 was working with high reliability in 2009. Some stagnation periods in the collector system were passed without difficulty. The collector yields are a little below expectations, mainly caused through shading (trees and a scaffold in 2009 in front of the building).

The high temperature flexibility in the driving circuit of the chiller (operation between 90℃ and 60℃ driving temperature) allowed often a conti nuous operation of the chiller for 6 hours or more. The chiller thus meets well the requirements for a solar thermal autonomous operation.

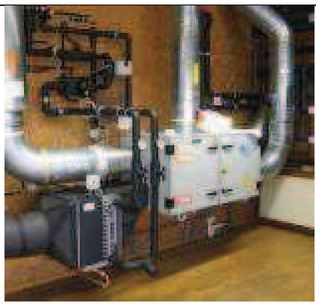
Problems occurred:

- Pollution in the open heat rejection circuit, caused by dust, pollen etc. This lead often to decreased mass flow and thus high heat rejection temperatures. Consequently, for several operation periods the performance of the chiller was low. The problem was solved through the installation of an automated cleaning procedure in the heat rejection circuit;
- A measurement error occurred in the heating season in the mass flow of the gas boiler heating circuit. Thus, the heat input of the boiler could not be calculated accurately, but estimated through assumptions of storage heat losses.

Overall success: although in 2009 only one of the two chillers was in operation, this (part) system worked with high reliability. The comfort in the seminar rooms has clearly increased, the users (students and teaches) are very satisfied in general. The solar autonomous operation mode matches well the occupation profile and cooling demand profile.

Since 2010, both chillers are in operation and during the first operation period, their driving circuits are connected in series (different chilled water temperatures for supply air dehumidification and chilled ceilings). This mode works reliable, the temperature difference in the solar loop is increased by this measure.

Photos

		
Collectors system, installed at the main college building	Chilled ceilings	One of the supply air systems

Monitoring Data

Measured period	Jan 2009 – Aug 2010
Monitoring level (according to Task 38 procedure)	3
Monitoring by	ZfS Rationelle Energietechnik GmbH, Hilden
Person responsible for evaluation	Edo Wiemken (Fraunhofer ISE, Germany) Phone: +49 (0)761 4588 5412 Email: edo.wiemken@ise.fraunhofer.de

2.10 Germany: Fraunhofer ISE, Freiburg

Solar air-conditioning with adsorption chiller of the canteen kitchen area at Fraunhofer ISE, Freiburg, Germany

Description of the application

The Fraunhofer Institute for Solar Energy Systems (ISE) institute building is an energy efficient building with passive cooling measures. An exception is the canteen kitchen area, where due to high internal loads active cooling of the supply air is appropriate. This is done by means of a small size thermally driven chiller.

Type of building kitchen area of institute

Location Freiburg, Germany

In operation since 2007

System operated by Fraunhofer ISE

Air-conditioned area 42 m^2

System used for space heating? Yes

System used for DHW preparation? No

General description of the system

The system technology is a closed cycle chilled water system with an adsorption chiller. Heat is provided by a solar thermal system and by the heat network of the institute which is fed by a CHP unit. During summer, the system runs in cooling mode. The medium temperature heat of the chiller is rejected by three ground boreholes of 80 m each. In winter, the heat pump function of the machine is activated and the ground tubes act as low-temperature energy source. The system thus cools the supply air into the kitchen (max. flow rate 3000m³/h) and pre-heats the air in the main channel of the air handling unit (max. flow rate 9000m³/h).

Central air-conditioning unit

Technology	closed cycle
Nominal capacity	5.5 kW_{cold} (08/2008 – 07/2009) 7.5kW (since 11/2009)
Type of closed system	Adsorption
Brand of chiller unit	SorTech ACS 05 (till 08/2009) ACS08 (since 11/2009)
Chilled water application	supply air cooling
Dehumidification	occasionally
Heat rejection system	dry, ground coupled boreholes

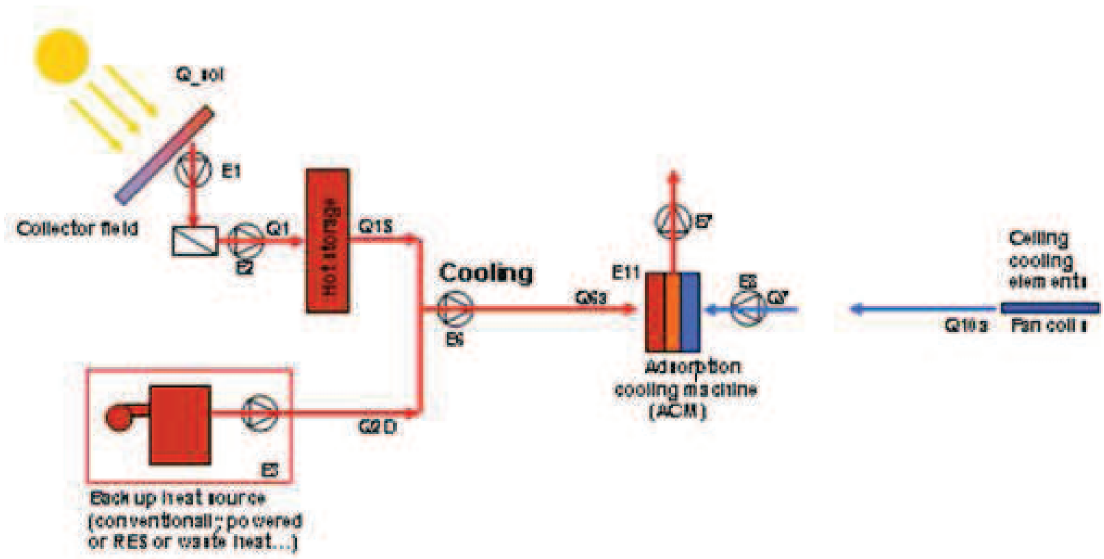
Solar thermal

Collector type	flat-plate
Brand of collector	Solvis FF 35s 3/2 FKY
Collector area	22 m^2 aperture
Tilt angle, orientation	30°, south
Collector fluid	water-glycol
Typical operation temperature	75 °C driving temperature for chiller operation

Configuration

Heat storage	2 m^3 water
Cold storage	none
Auxiliary heater	Institute heat network, operated by CHP and gas boiler
Use of auxiliary heating system	Auxiliary driving source for chiller, auxiliary driving source for heat pump operation in winter
Auxiliary chiller	no

System scheme



System performance

For one year of operation (August 2008 until July 2009) an average seasonal Coefficient of Performance of COP = 0.43 was observed. The solar thermal coverage of the total heat input for cooling was around 45% while for heating it was 6.7%. It has to be remarked that the solar energy in winter is only used to drive the adsorption heat pump if it reaches 70℃ and is not used directly for heating. The chiller was newly evacuated before the monitoring period. During winter, the machine runs in heat pump mode; the resulting COP in this operation mode for the whole heating season was 1.25.

In summer 2009 the adsorption chiller ACS05 was exchanged by the model ACS08, which started its operation in November 2009. In the heating season 2009/2010 (Sept. 2009-May 2010) a thermal COP of 1.27 was achieved while the solar fraction in the driving heat was 3.2%. This means that 3.2% of the driving heat was provided by the solar system. The summer period 2010 (April – Sept. 2010) a cooling COP of 0.58 and a solar fraction of 65.2% was obtained.

It has to be mentioned that the systems was operated every day from around 8 to 17 o'clock whenever the temperatures in the supply air were above 20℃ (Cooling mode) and below 14.5℃ (heating mode). Solar energy was used if the temperature in the upper part of the storage was above 73℃. Further, it has to be menti oned that the driving temperatures from the CHP driven network were very irregular and often in the range between 55 and 70℃.

System reliability and overall success of the installation

After a decrease in performance during the first operation months (inert gasses), the chiller ACS05 was newly evacuated in August 2008. Apart from this, the operation was very reliable. The new chiller (ACS08) did not require any service in the operation period nor showed any degradation in its performance. The system concept is promising: no cooling tower is required, and a fraction of the rejected heat in summer in the borehole may be available in the low-temperature heat source during the heat pump operation mode in winter. Another advantage is the noiseless operation of the chiller. The use of the solar resource could be improved by using the solar heat in winter directly instead of driving the adsorption machine. This would be advantageous as lower temperatures could be used.

Photos

		
22m^2 of Solvis flat-plate collectors on the instute's rooftop	SorTech ACS 05 chiller unit	Fraunhofer ISE institute building. In the foreground: installation of ground tubes

Monitoring Data

Measured period	Aug 2008 – July 2009 (ACS05); Nov. 2009 – Sept. 2010 (ACS08)
Monitoring level (according to Task 38 procedure)	3
Person responsible for monitoring	Tomas Núñez (Fraunhofer ISE, Germany) Phone: +49 761 4588 5533 Email: tomas.nunez@ise.fraunhofer.de

2.11 Germany: ZAE Bayern Office Building, Garching

Description of the application

Within the framework of the German “Solarthermie 2000plus” Programme a pilot installation of an innovative solar heating and cooling system has been erected at the office building of the ZAE Bayern in Garching, Germany in 2007. The installation simultaneously serves as a field test project for a compact water/LiBr absorption chiller with 10 kW nominal capacity. Further more the performance and durability of a low temperature latent heat storage based on salt hydrate is tested.

Type of building Office building	
Location Garching, Germany	
In operation since 2007	
System operated by ZAE Bayern	
Air-conditioned area 400 m²	
System used for space heating? Yes	
System used for DHW preparation? Yes	

General description of the system

In conventional absorption cooling installations, wet cooling towers designed for coolant supply/return temperature 27/35°C are applied. When a dry air-cooler is to be used, cooling water temperatures have to be increased. As a consequence of the increase of the cooling water temperature, the temperature level of the driving heat supplied to the regenerator of the absorption chiller has to be increased accordingly. By integration of a latent heat storage into the heat rejection system of the absorption chiller, a part of the reject heat of the chiller can be buffered during the operation of the solar cooling system, allowing for lower coolant temperatures during peak load operation of the chiller. The stored reject heat then can be discharged during off-peak operation or night time when more favourable ambient conditions, i.e. lower ambient temperatures or electricity tariff, are available. During heating operation the latent heat storage balances heat generation by the solar system and other heat sources and the supply to the consumer. Thus a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	10 kW_{cold} (basic load)
Type of closed system	Absorption
Brand of chiller unit	Sk Sonnenklima: Suninverse
Chilled water application	Ceiling panel
Dehumidification	No
Heat rejection system	Dry cooler supported by a latent heat storage

Solar thermal

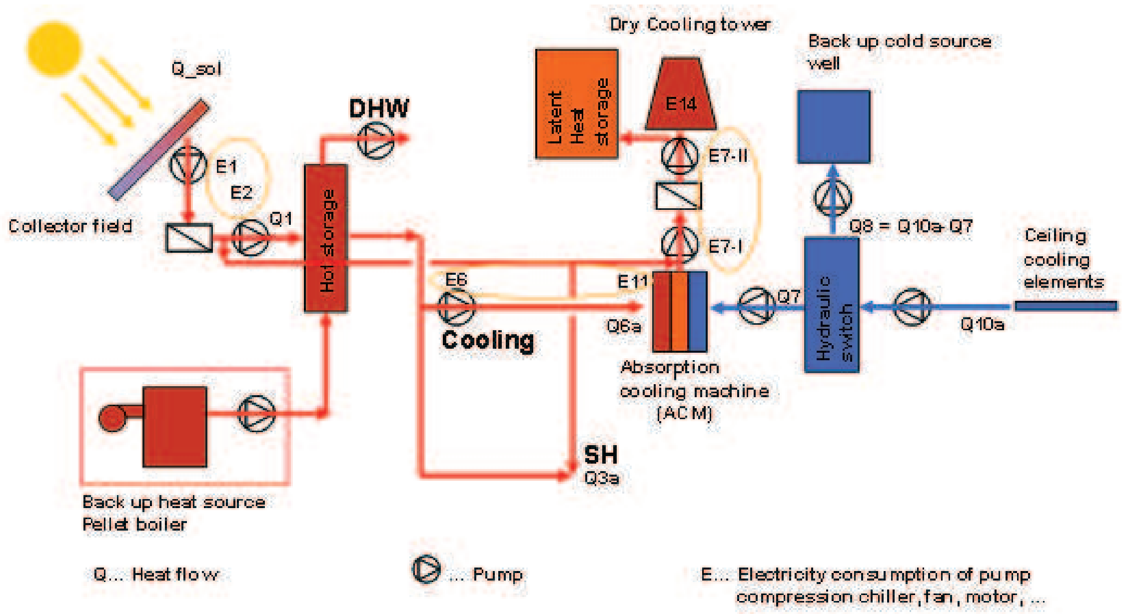
Collector type	Flat plate
Brand of collector	Wagner EURO C20 AR
Collector area	57.4 m²
Tilt angle, orientation	40°, south +10°

Collector fluid	Water-glycol
Typical operation temperature	92 ℃ driving temperature for chiller operation

Configuration

Heat storage	2x1 m³ water tank (serial) and 1,6 m³ latent heat storage
Cold storage	No
Auxiliary heater	Pellet boiler
Use of auxiliary heating system	Yes
Auxiliary chiller	Well

System scheme



System performance

The system has been in operation and monitored for three complete years (2008-2010). Due to the low system temperatures for heating and moderate chilled water temperatures supplied to the activated ceilings in summer a high collector yield of 400 kWh / (m² a) is obtained.

The absorption chiller itself has been operated for about 600 hours a year, producing cold at 15 ℃ by means of solar heat about 90 ℃ with an av erage thermal Coefficient of Performance of 0.69 up to 0.72. By replacing some ineffective pumps with high efficiency components in spring 2009 the overall electrical COP increased significantly (about 20 %). The average total electrical COP of the solar cooling system was 6.6 in 2009. Almost 60% of the primary energy has been saved compared to a conventional compression cooling system supplying the same amount of cold.

During the monitoring period the two latent heat storage modules have undergone over 800 loading and unloading cycles under real conditions. Due to the use of the latent heat storage the cooling water return temperature did not exceed 33.5 ℃ despite dry air cooling and ambient temperatures above 32 ℃. In heating mode a bout 15 % of the overall heat demand has been provided by the latent heat storage.

System reliability and overall success of the installation

After installation in 2007 followed by an optimization phase of the control strategy, the solar cooling and heating system has been operated in automatic mode since January 2008.

- Solar thermal system and hot water buffer

The solar thermal system has been working properly. An overheating of the solar thermal panel system is avoided by using the dry air cooler as heat sink for surplus solar heat. The system performance can be improved by adding an irradiation related speed control of the primary loop pump and an enlarged heat exchanger.

- Absorption chiller

The thermally driven chiller worked quite well. Only once the automatic anti-crystallization routine has been activated due to inert gases caused by corrosion. Even after standstill during winter the chiller has been available for automatic operation without manual intervention. Yet, the chiller operated at reduced thermal COP due to the effect of inert gases accumulated during standstill. After a manual evacuation the chiller reached nominal values again.

- Reject heat loop with dry air cooler and latent heat storage modules

The dry air cooler has been cleaned once a year in spring to remove pollen of the heat exchanger surface. No further maintenance has taken place. By implementing a 3-way valve in the hydraulic loop of the latent heat storage a speed control of the fan is no longer required in further installations. The latent heat storage modules have proved their functionality and reliability. Up to now no degradation in performance or storage capacity has been detected. In summertime cooling water temperature has been reduced significantly and in wintertime the efficiency of the solar system has been increased due to low and constant temperatures of the latent heat storage.

In consequence of the good performance and reliability of the system the monitoring will be continued and further improvements are going to be implemented.

Photos

		
Libr/water absorption chiller: Suninverse	Heat reject system: dry cooler +latent heat storage	Solar panels and office building, ZAE Bayern

Monitoring Data

Measured period	Jan. 2008 – ongoing
Monitoring level (according to Task 38 procedure)	1
Person responsible for monitoring	Martin Helm (ZAE Bayern, Germany) Phone: +49 89 329442 33 Email: helm@muc.zae-bayern.de

2.12 Germany: Radiological Practice, Berlin

Description of the application

In the Radiological Practice in Berlin cooling demand occurs throughout the day and throughout the year for permanent cooling of the tomografic equipment. Originally, the cooling demand is covered by a chilled water network, serving also other medical services in the building. An electrically driven compression chiller is operating the network.

Since summer of 2008, an additional small solar autonomous cooing system was installed for covering daily cooling peak loads of the practice. The approach made is interesting, since

- pre-installed roof-top system: the chiller, hot water storage and all hydraulic components are installed in a small size container and placed on the roof-top of the building, since no further technical area inside the building was available;
- dry heat rejection with an air-cooler, which is used in winter for direct chilled water production without chiller operation (free cooling mode at sufficient low ambient temperatures);
- beside a constant ground level of cooling demand (approx. 8 kW), daily peaks of cooling demand up to 30 kW matches very well the the operation time of the solar thermally driven chiller

A monitoring system was operated in 2009.

Type of building heritage protected commercial building	
Location Berlin, Germany	
In operation since 2008	
System operated by SK Sonnenklima, Berlin	
Air-conditioned area 120 m^2	
System used for space heating? No	
System used for DHW preparation? No	

General description of the system

An absorption chiller was installed and thermally driven with solar heat alone. The solar system is used only for the chiller; due to building related properties, it was not possible to connect the solar system to the building heating network. The solar thermal system was prepared to be operated with pure water and a corresponding control was installed. However, for test reasons, the system was filled with water-glycol.

The plant is pre-cooling the chilled water, returned from the practice. In case the temperature of the chilled water is not sufficient low (approx. 8℃) leaving the absorption chiller, it is sub-cooled passing a heat exchanger in the building chilled water network.

During winter, chilled water can be prepared directly through the air cooler. This mode was already widely used in 2008 and 2009. As a consequence, the heat rejection circuit as well as the cold water circuit are filled with water-glycol. Also the hot water circuit is using water-glycol (foreseen to be replaced with pure water later).

In order to simplify the installation and maintenance of the roof-top system, dry heat rejection was chosen.

The system is a pilot installation to gain experiences with

- solar pre-cooling in a process cooling network;
- dry heat rejection in combination with the installed absorption chiller;
- free cooling mode in winter
- pre-installed contained roof-top system

Detailed planning: SK Sonnenklima

Central process pre-cooling unit

Technology	closed cycle
Nominal capacity	10 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	SK Sonnenklima: Suninverse
Chilled water application	Process cooling (medical equipment); pre-cooling
Dehumidification	none
Heat rejection system	dry air-cooler

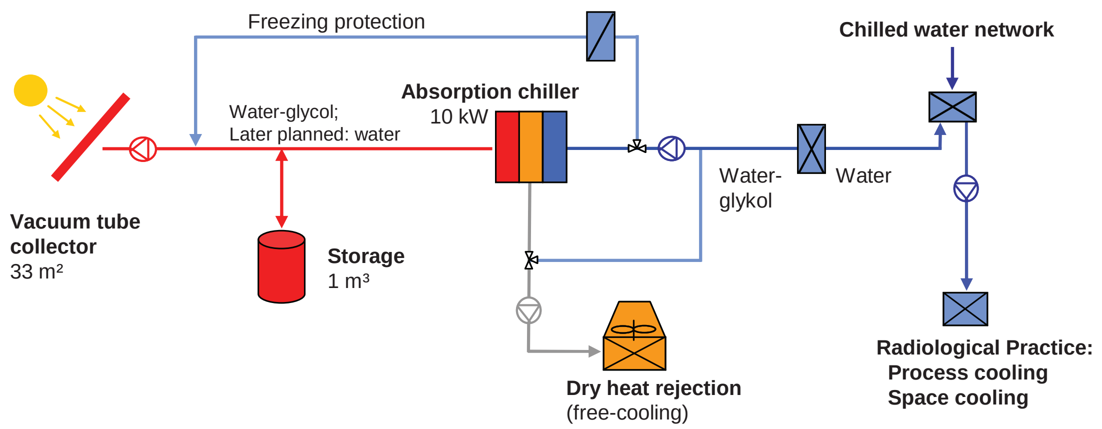
Solar thermal

Collector type	evacuated tube with CPC-mirror
Brand of collector	Phönix Sonnenwärme AG
Collector area	33 m^2 aperture
Tilt angle, orientation	45°, south
Collector fluid	water-glycol
Typical operation temperature	80℃

Configuration

Heat storage	1 m^3 water-glycol
Cold storage	none
Auxiliary heating support	none
Auxiliary heater	none
Auxiliary chiller	backbone chilled water network (el. compression chiller)

System scheme



System performance

For reasons given below, no performance data can be given.

System reliability and overall success of the installation

The system went into operation in late 2008, monitoring data were available since February 2009. In 2009, several adjustments and tests were made on the system. A continuous system operation with effective cooling was available in July and August 2009. However, the system was still not optimized and the monitored performance was below expectations (e.g., monthly thermal COP of below 0.3; chilling capacity too low). Some monitoring signals were not reliable; e.g., the electrical performance could not be calculated, further monitoring errors occurred in the heat rejection circuit. Before solving the remaining problems, the company SK Sonnenklima went into a insolvency procedure. The system operation stopped end of August 2009 and was not restarted since then (also due to the ongoing insolvency).

In the available operation period of the chiller July and August 2009, the system has shown apart from optimizing problems the general ability to reduce the peak-load cooling demand from the chilled water network; pre-cooling was done reliable. Thus, the concept is still promising. Furthermore, the free cooling mode via the cooling tower was also working reliable.

Since the insolvency procedure was not closed until August 2010, the further use of the system is not clear yet.

Photos

		
Collector; the container with the chiller is in the background	Absorption chiller in the container	Dry heat rejection, attached to the container

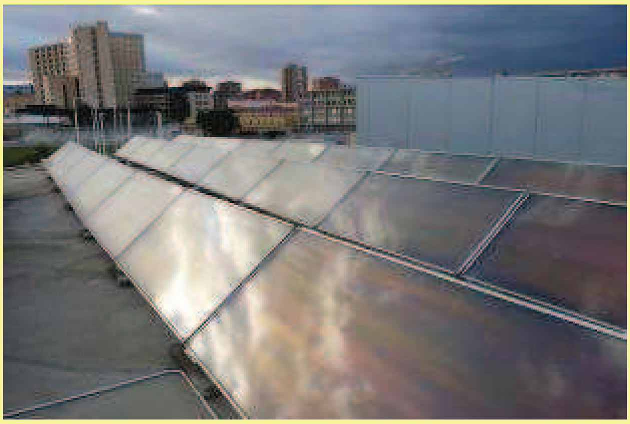
Monitoring Data

Measured period	Feb 2009 – Aug 2009 (commissioning phase)
Monitoring level (according to Task 38 procedure)	3 (planned)
Monitoring by	Technical University Chemnitz
Person responsible for evaluation	Edo Wiemken (Fraunhofer ISE, Germany) Phone: +49 (0)761 4588 5412 Email: edo.wiemken@ise.fraunhofer.de

2.13 Spain: Gymnasium of the University of Zaragoza, Zaragoza.

Description of the application

The installation is located in Zaragoza (Spain) at the indoor sports centre of the University of Zaragoza and it is used to cool a gymnasium. This installation was designed as a consequence of the overheating in the existing solar collectors. In summer, the solar field was oversized because solar power was higher than needed.

Type of building Gymnasium	
Location Zaragoza, Spain	
In operation since 2007	
System operated by University of Zaragoza	
Air-conditioned area 215 m^2	
System used for space heating? No	
System used for DHW preparation? No	

General description of the system

This solar cooling installation was designed as a consequence of the overheating problems of the existing solar filed used to contribute to the domestic hot water supply of the building. In the summer, to solve this problem and to use this solar waste energy, the chosen solution was the installation of an absorption chiller. Therefore, the solar collector field of the solar air-conditioning system has 37.5 m^2 of useful area. Solar radiation is absorbed and transformed into thermal energy to feed a 4.5 kW absorption machine by Rotartica. The installation contains 700 liters of hot water storage and two gas boilers as an auxiliary system, but both have been never used. Two fan coils transfer the chilling power from the evaporator of the absorption machine to the gymnasium.

Initially, a dry cooling tower was installed to reject the produced heat of the absorption cycle to the outdoor air. After the analysis of the two first years, and due to the negative influence of the ambient temperature on the COP of the chiller, an open geothermal system was installed as a heat rejection system, to improve the COP and the performance of the chiller.

Central air-conditioning unit

Technology	Closed cycle
Nominal capacity	4.5 kW_{cold}
Type of closed system	Absorption
Brand of chiller unit	Rotartica 045
Chilled water application	Fan coil
Dehumidification	No
Heat rejection system	Dry cooling tower and an open geothermal system since 2009

Solar thermal

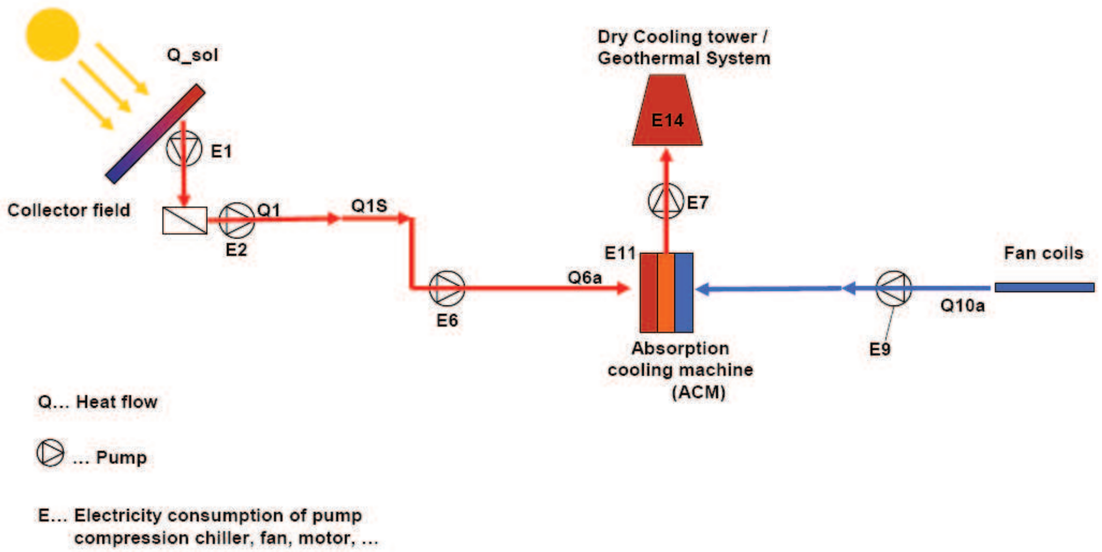
Collector type	Flat-plate
Brand of collector	Viessmann
Collector area	37.5 m^2 aperture area
Tilt angle, orientation	30°, South

Collector fluid	Water-glycol
Typical operation temperature	80 ℃ driving temperature for chiller operation

Configuration

Heat storage	0.7 m^3 water
Cold storage	no
Auxiliary heater	yes (not used)
Use of auxiliary heating system	no
Auxiliary chiller	none

System scheme



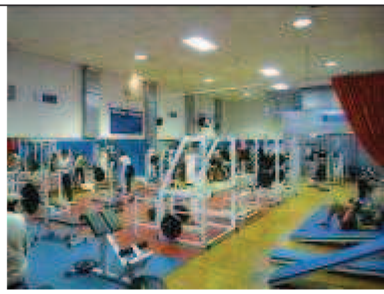
System performance

In the last three years, 2007, 2008, 2009 and 2010, the chiller has been operated under different scenarios of performance. In 2007 the chiller worked with two different solar collector areas (20 m^2 and 37.5 m^2) in order to analyze the influence on the chiller performance The average COP value operating with 20 m^2 was 0.51 whereas with the 37.5m^2 the COP value achieved 0.49, although in the first case the average chilling power 4.0 kW, in the second one, the chilling power increased to 5.3 kW. In both cases, the chiller rotary drum speed was 300 rpm. In the next scenario the chiller worked with the whole solar surface and its rotary speed was increased till 400 rpm to improve the heat and mass process of the absorption cycle. The performance of the system improved (COP: 0.57, W_{ch}: 5.78 kW), but the installation kept on having a strong influence of the outdoor temperature. In the year 2008, the average outdoor temperature was 12% higher than the year before, a fact that was displayed in the results of that year (COP: 0.51, W_{ch}: 4.4 kW). Finally, to eliminate the outdoor temperature influence on the heat rejection system and taken the opportunity of using a closed water well with a constant temperature of 17ºC, in 2009, an open geothermal cycle was installed. The obtained results presented constant values, such as the COP as well as the chilling and heat rejection powers, but they were lower than the expected (COP: 0.52, W_{ch}: 4.2 kW), due to the higher operational temperature of the water well (25ºC instead of 17 ºC used in the design process).

System reliability and overall success of the installation

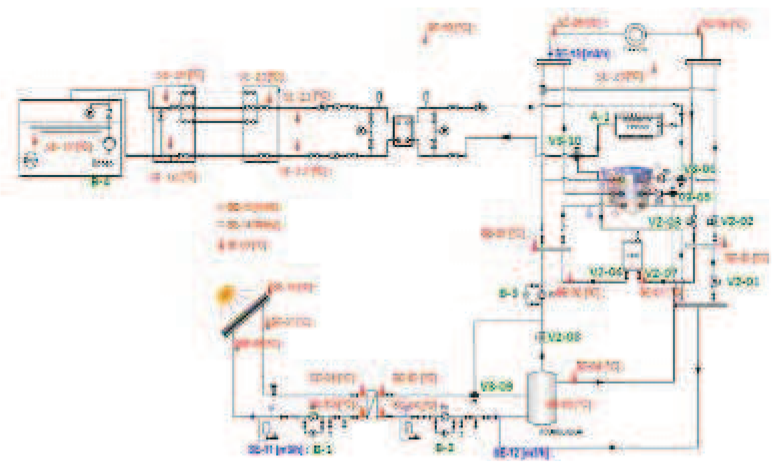
In general, the operation of the installation works reliable. The principal objective of the solar cooling system has served the purpose of resolve the overheating problems of the solar field. Besides that the users of the gymnasium are satisfactory because the ambient conditions in the gym have been improved.

Photos

		
Solar field	Machinery room	Gymnasium
		
Fan coil	Initial heat rejection system	Geothermal heat rejection system (2009)

Monitoring Data

Measured period	2007 - ongoing
Monitoring level (according to Task 38 procedure)	2
Person responsible for monitoring	Fernando Palacín (National Renewable Energy Centre (CENER)) Phone: +34 948 252 800 Email: fpalacin@cener.com
	Carlos Monné (University of Zaragoza) Phone: +34 976 762 042 Email: cmmb@unizar.es



Scheme of the solar cooling installation in Zaragoza

3 Results

In this chapter, the monitoring results of 11 of the 13 systems described above will be presented and compared with eachother. For two systems, no results can be presented because there were too many operational and/or monitoring problems during the monitoring period.

For 6 of the 11 systems, at least one year of data is available. For all the other systems, between two and six months of data have been recorded and will be presented here.

The 11 presented systems can be subdivided into different groups of systems:

Group 1: Systems that were only used for cooling (Zaragoza and Maclas, 4 months of data)

Group 2: Systems where winter backup was not monitored, the measured space heating consumption is therefore only the part that was produced by the solar thermal system (Perpignan).The system in Perpignan uses a compression chiller as cold backup.

Group 3: Systems that use the hot backup system only for winter operation. In summer, the system is operated either as solar autonomous system or with a cold backup system (Sattledt, Chaméry, Butzbach, Garching, Graz).

Group 4: Systems that use the hot backup system for both summer and winter operation.

3.1 Thermal COP of the Chiller

In all eleven monitored systems, the thermal coefficient of performance (COP) of the thermally driven chiller was determined by measuring the produced cold and the driving heat. Only two systems include an adsorption chiller (Perpignan and Freiburg). In all other systems, absorption chillers are used.

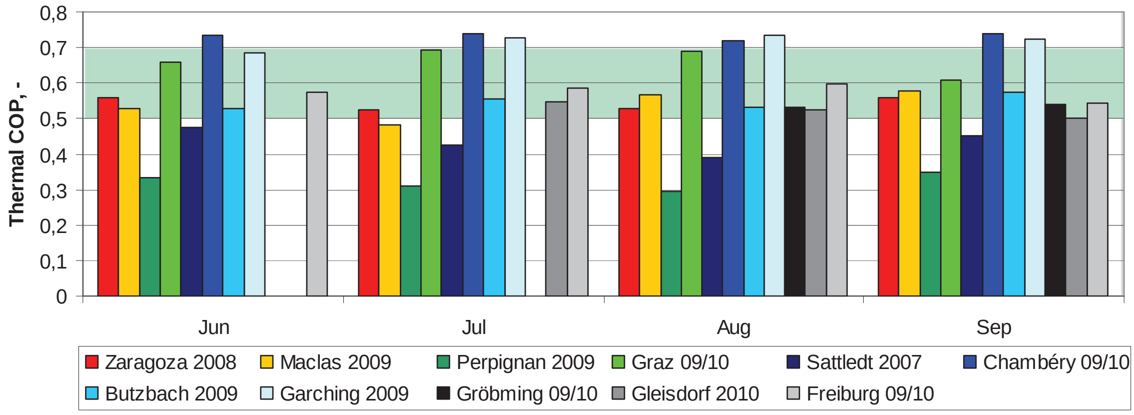


Figure 1: Measured thermal COP of 11 small-scale solar cooling systems.

The results show that almost all chillers have thermal COPs in a range between 0.5 and 0.7, i.e. close to the manufacturers' specifications but of course depending on the operating conditions in the specific system. Only two systems show significantly lower values. One is the adsorption chiller in Perpignan that has to operate under unfavorable heat rejection conditions (high ambient temperatures and most dry heat rejection). The other one is the system in Sattledt where the driving temperature was relatively low (55-75℃).

3.2 Electrical COP of the Chiller

The next step is to look at the electricity consumption of the chiller itself, i.e. the solution pump or any internal valves. This value was measured separately only for 4 of the analyzed systems. The electrical COP of the chiller is defined as the produced cold divided by the electricity consumption of the chiller by itself.

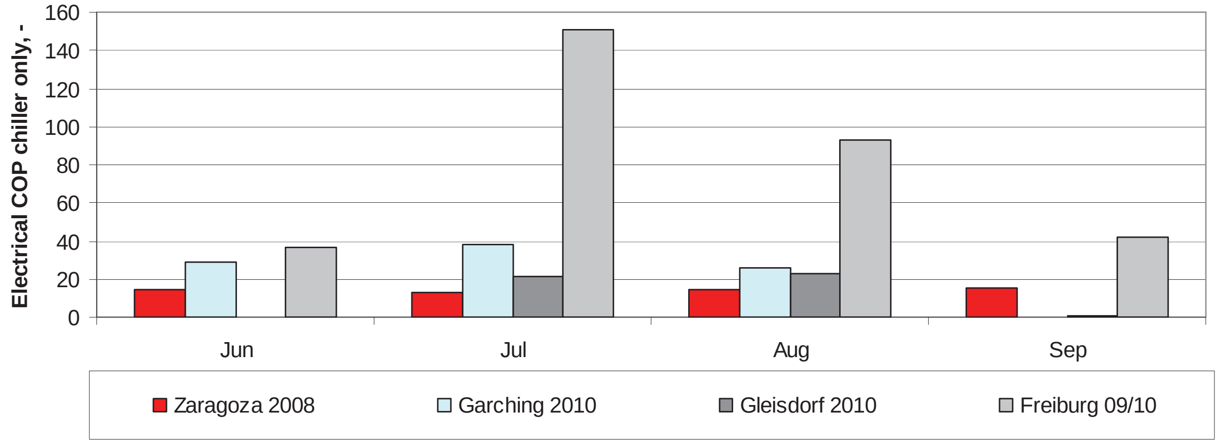


Figure 2: Electrical COP of the chiller only for 4 small-scale solar cooling systems.

The results in Figure 2 show that the electricity consumption of the adsorption chiller in Freiburg has very little electricity consumption. The reason is that there is no solution pump necessary. The electricity consumption of the chiller alone is almost independent of the cold production: it only depends on the switch-on time and this time is almost the same for all months. In July 2010 on the other hand a high amount of cold was produced, increasing significantly the electric COP of the chiller alone. The absorption chillers are all in the same order of magnitude. The best one is the one in Garching. The results shown here are values of 2010 because the electricity consumption of the chiller was not measured separately in 2009. In addition, only the electricity consumption of the solution pump was included and not the pump for the generator circuit which is also included in the chiller. The system in Zaragoza uses a Rotartica absorption chiller which consumes more electricity than other absorption chillers due to its rotating technology.

3.3 Electrical COP Cold Production

The electrical COP of the cold production includes in addition to the electricity consumption of the chiller itself, the electricity consumption of the heat rejection system (pump and fan) and the pump in the generator loop. If there is a cold storage tank, it includes also the electricity consumption of the pump between chiller and cold storage tank. The results for the seven systems where this value was measured are shown in Figure 3. Obviously, the values are significantly lower than for the electrical COP of the chiller by itself. The best system in this monitoring campaign reaches a value of 8. A number of systems lie in a range of 5 to 6 which are acceptable values but still leave room for improvements.

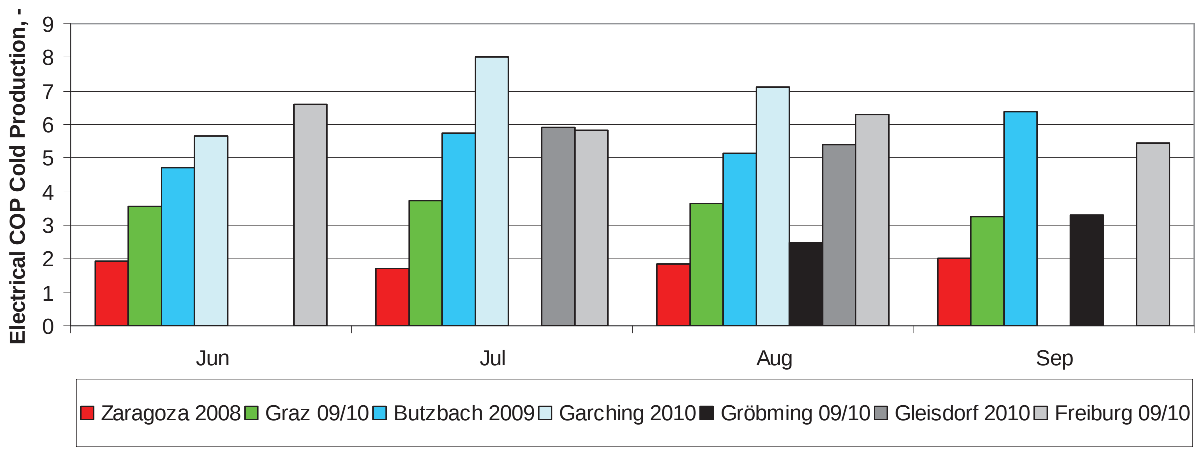


Figure 3: Electrical COP of the cold production

A few systems show values below 3 which means that the system probably consumes more electricity than a conventional compression chiller would.

A general conclusion is that in most systems the electricity consumption of the external components of the cold production (pumps, fans) has not been optimized and is therefore too high. In the best system in Figure 3 (Garching) the pump in the generator loop has been exchanged against a more energy efficient one. In addition it is planned to replace the fan of the dry cooler in the near future which would increase the electrical COP even further.

Another topic is the part load operation of the system. If a system is operated a lot in part load but the external pumps are still operated at full speed, this will lead to low electrical COPs. Optimized control strategies for this operation scenario are mandatory to ensure primary energy savings of the system.

3.4 Total electrical COP of the Solar Cooling System

As a next step, the total electrical COP of the system can be calculated. This includes in addition to the electrical consumers mentioned before, the electricity consumption of the pumps in the solar circuit(s) and the electricity consumption of the auxiliary heater and the associated pumps if applicable.

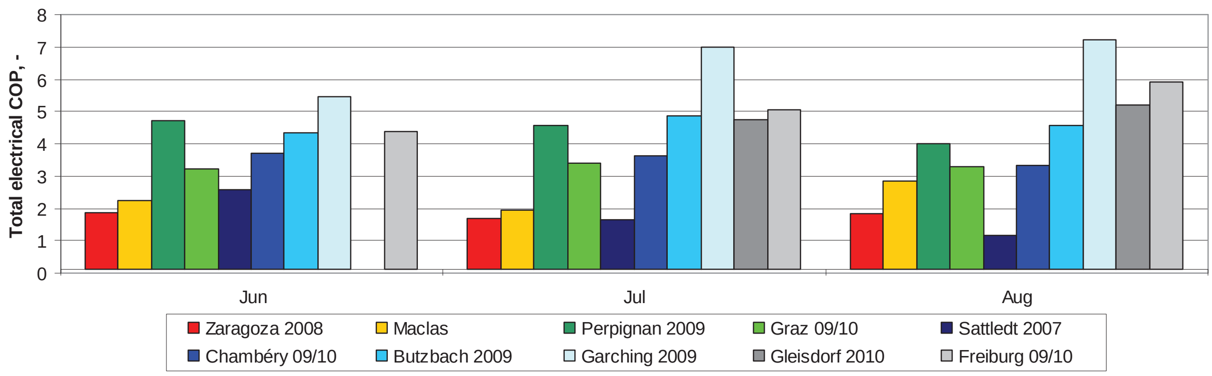


Figure 4: Total electrical COP of 10 solar cooling systems

Figure 4 compares the total electrical COP of 10 systems only for the summer months. The value for the system in Gröbming is not shown because the system was also used for DHW preparation and space heating. Therefore, the value is significantly higher but not comparable.

For the other 10 systems the trend is the same as described in the previous section. But the total electrical COPs of the systems are obviously somewhat lower than for the cold production itself, but the more critical factor seems to be the electricity consumption of the pumps in the chiller circuits and the heat rejection system. These are the components that need to be optimized in terms of component selection and control strategies.

3.5 Fractional Primary Energy Savings

Now, let's take a look at the probably most important monitoring result of a solar heating and cooling system: The primary energy saved by the system.

For this purpose, the primary energy ratio was calculated for both the monitored solar heating and cooling system and for a conventional reference system delivering the same amount of heating, cooling and domestic hot water.

The following assumptions were taken for the conventional reference system:

Space heating is done by a fossil fuel boiler without storage tank. The primary energy ratio for space heating is calculated as follows:

$$PER_{ref,space\_heating} = \frac{Q_{SH}}{\frac{Q_{SH}}{{}_{boiler} \cdot {}_{fossil}} + \frac{0.02 \cdot Q_{SH}}{{}_{elec}}} = \frac{1}{\frac{1}{0.95 \cdot 0.9} + \frac{0.02}{0.4}} = 81.99\,\%$$

with

${}_{boiler}$	Mean annual efficiency of fossil fuel boiler:	0.95
${}_{fossil}$	Primary energy factor for fossil fuel:	0.9 kWh_{final}/kWh_{PE}
${}_{elec}$	Annual electricity generation efficiency:	0.4 kWh_{el}/kWh_{PE}

The electricity consumption of the fossil fuel boiler was assumed to be 2% of the generated heat of the boiler. All these assumption lead to a reference primary energy ratio for space heating of 81.99%. The numbers listed above are mean European values and have been chosen in order to compare different systems. Local values may be used in order to evaluate the saving in a specific location.

Domestic hot water preparation is done using the same fossil fuel boiler than for space heating but assuming a storage tank containing 75% of the average daily hot water consumption in the monitored system. Average heat losses of such a tank were then added to the measured heat consumed for domestic hot water.

Cooling is done by a conventional compression chiller. The primary energy ratio for cooling is calculated as follows:

$$PER_{ref,cooling} = \frac{Q_{cold}}{\frac{Q_{cold}}{SPF_{ref} \cdot {}_{elec}}} = \frac{1}{\frac{1}{2.8 \cdot 0.4}} = 112\,\%$$

With

SPF_{ref}	Reference seasonal performance factor:	2.8

This leads to a reference primary energy ratio for cooling of 112%.

This reference primary energy consumption is then compared with the measured primary energy consumption of the system according to the equations below:

$$f_{sav,shc} = 1 - \frac{\frac{Q_{boiler}}{{}_{fossil} \cdot {}_{boiler}} + \frac{Q_{RES}}{{}_{RES} \cdot {}_{RES}} + \frac{E_{el}}{{}_{elec}} + \frac{Q_{cooling,missed}}{SPF \cdot {}_{elec}}}{\frac{Q_{boiler,ref}}{{}_{fossil} \cdot {}_{boiler,ref}} + \frac{E_{el,ref}}{{}_{elec}} + \frac{Q_{cooling,ref}}{SPF_{ref} \cdot {}_{elec}}}$$

$$f_{sav,shc} = 1 - \frac{PER_{ref}}{PER}$$

Some monitored systems use fossil backup systems. In that case, the same primary energy factors and efficiencies as for the reference system were used. Also for the electricity consumption the same electricity conversion efficiency was used.

For other backup systems (in the equation called renewable energy sources Q_{RES}), individual primary energy factors and efficiencies were used. The primary energy factor ${}_{RES}$ here is defined as the “non renewable primary energy factor”:

a) Wood pellets (only Gröbming): ${}_{RES}$=0.9, ${}_{RES}$=10 kWh_{fuel}/kWh_{PE}

b) District heating system city of Graz: ${}_{RES}$=1, ${}_{RES}$=0.96 kWh_{heat}/kWh_{PE}

c) Combined heat and power plants (CHP):

The primary energy for heat taken from a CHP plant is calculated according to the following equation:

$$_{RES} = \frac{Q_{CHP}}{H_{fuel} \Big/ _{fuel} - W_{el} \Big/ _{el}}$$

Where Q_{CHP} is the heat produced by the CHP plant, H_{fuel} is the fuel used (in kWh) and W_{el} is the electricity produced. The electricity produced by the CHP replaces electricity in the power grid, thus $_{el}$ is the primary energy factor for the electricity in the grid. Expressed in CHP efficiencies this gives:

$$_{RES} = \frac{_{th}}{1 \Big/ _{fuel} - _{el} \Big/ _{el}}$$

Where η_{th} is the thermal efficiency of the CHP and η_{el} is the electric efficiency of the CHP.

With this formalism, the efficiency of the heat source η_{RES} has to be considered as equal to 1.

In order to compare different systems in different locations the same average primary factors for electricity ($_{elect}$=0.4 kWh_{el}/kWh_{PE}) and fuel ($_{fuel}$= $_{fossil}$=0.9 kWh_{fuel}/kWh_{PE}) should be used. Local values should be used in order to evaluate the performance of the system in a specific surrounding.

For Freiburg (η_{th}=49.2%; η_{el}=26.2%), using average values for $_{elec}$=0.4 kWh_{el}/kWh_{PE} and $_{fuel}$=0.9 kWh_{fuel}/kWh_{PE} (natural gas) we obtain an average value for the CHP of $_{RES}$=1.08 kWh_{heat}/kWh_{PE}, with local values ($_{fossil}$=0.909 kWh_{fuel}/kWh_{PE}; $_{elec}$=0.36 kWh_{el}/kWh_{PE}) we obtain an average value for the CHP of $_{RES}$=1.32 kWh_{heat}/kWh_{PE}.

For Gleisdorf thermal and electrical efficiency of the two bio-fuel (rapeseed oil) CHP's needs to be estimated based on data sheets, because fuel consumption was not measured. The following values were assumed: η_{th}=45%; η_{el}=24%. Using average values for $_{elec}$=0.4 and $_{fuel}$= 2.35 kWh_{fuel}/kWh_{PE} (rapeseed oil) we obtain an average value for the CHP of: $_{RES}$= -2.53 kWh_{heat}/kWh_{PE}. This negative value shows that the produced electricity is replacing more non renewable energy from conventional (fossil) electricity production (η_{el} x $_{fuel}$ = 0.24 x 2.35 = 0.564 kWh_{el}/kWh_{PE} instead of 0.4 kWh_{el}/kWh_{PE}) than non renewable energy remains in the produced heat by the rapeseed oil fired CHP.

To present the results more clearly, the graphs of the fractional primary energy savings are divided into summer and winter. This way, systems that were only operated or monitored in summer can be taken out from the winter graphs.

3.5.1 Summer

All 11 monitored systems could be analyzed for summer operation. In Figure 5 all systems that have no backup system in summer or a cold backup system are shown.

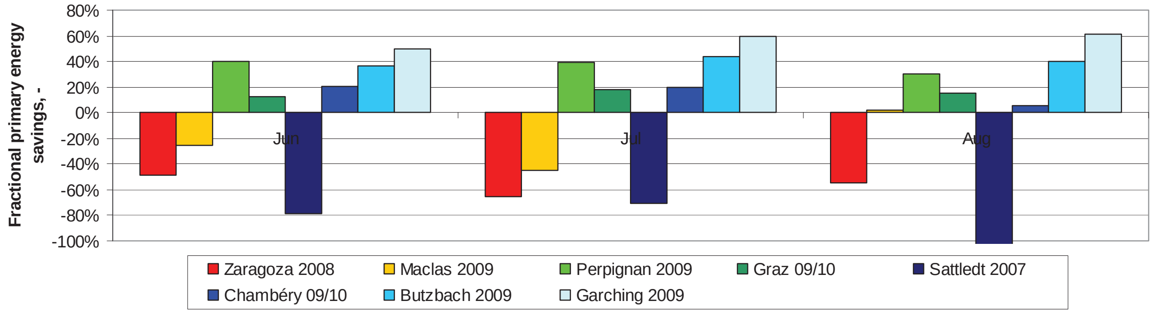


Figure 5: Fractional primary energy savings of the monitored systems without heat backup for the summer months

Five systems show positive fractional primary energy savings, in 3 cases these savings are negative.

Let's first look at the positive cases: The best is the system in Garching which already had the best total electrical COP. The system reaches approximately 60% in July and August. Four other systems (Perpignan, Butzbach, Graz and Chambéry) had reached total electrical COPs between 3 and 5 which translate into fractional primary energy savings between approximately 10 and 40%. While it is good that the primary energy savings of these systems are positive, as has been said before all of the systems can still be optimized (especially in terms of electricity consumption) which would further increase the fractional primary energy savings.

The systems with negative fractional primary energy savings are the ones in Zaragoza, Maclas and Sattledt which had already shown very low total electrical COPs. The fractional primary energy savings of these systems can very likely become positive by means of system optimization.

The three systems shown in Figure 6 use not only solar energy to operate the sorption chiller but a pellets boiler (Gröbming), a gas driven CHP unit (Freiburg) and a combination of a conventional gas boiler and different rapeseed oil fired CHP units (Gleisdorf) respectively.

For a differentiated analysis primary energy savings were calculated based on three different boundary conditions and assumptions respectively:

1) Primary energy factor for the fuel is the "non renewable primary energy factor" for the renewable heat sources (pellets in Gröbming: $_{RES}$=0.9, $_{RES}$=10 kWh_{fuel}/kWh_{PE}; natural gas CHP in Freiburg: $_{RES}$=1, $_{RES}$=1.08 $kWh_{heat}/kWh_{PE;}$ rapeseed oil CHP in Gleisdorf: $_{RES}$=1, $_{RES}$=-2.53 kWh_{heat}/kWh_{PE} (in Figure 6: "res PE factor" or "CHP PE factor")

2) In case of CHP as auxiliary heater the heat is counted for free in terms of non renewable primary energy consumption based on the assumption, that the CHP is operated "electricity production controlled" and heat therefore is waste heat. (in Figure 6: "heat free")

3) For comparison reasons the auxiliary heater is assumed to be the reference natural gas boiler with boiler efficiency $_{boiler}$ = 0.95 and primary energy factor of natural gas $_{fossil}$ = 0.9 kWh_{fuel}/kWh_{PE} (in Figure 6: "fossil PE factor")

The system in Gröbming was only measured in August. At first sight it seems to be the best system in this monitoring campaign (80% of fractional primary energy savings based on "res PE factor" are reached). However, there are two reasons for that: First, the backup system uses a wood pellets boiler with a primary energy factor of 10 kWh_{fuel}/kWh_{PE}. In addition, unlike the other 10 systems, the system was also used for domestic hot water preparation. Both factors are very positive for the performance of a solar heating and cooling system. Mainly due to the different fuel type it makes it difficult to compare the results with the other systems. Therefore, a comparison assuming the same (fossil) primary energy factor (0.9 kWh_{fuel}/kWh_{PE}) was performed. In Figure 6 this is shown in the three hatched columns on the right hand side. For Gröbming, the fractional primary energy savings are thus reduced to roughly 30%.

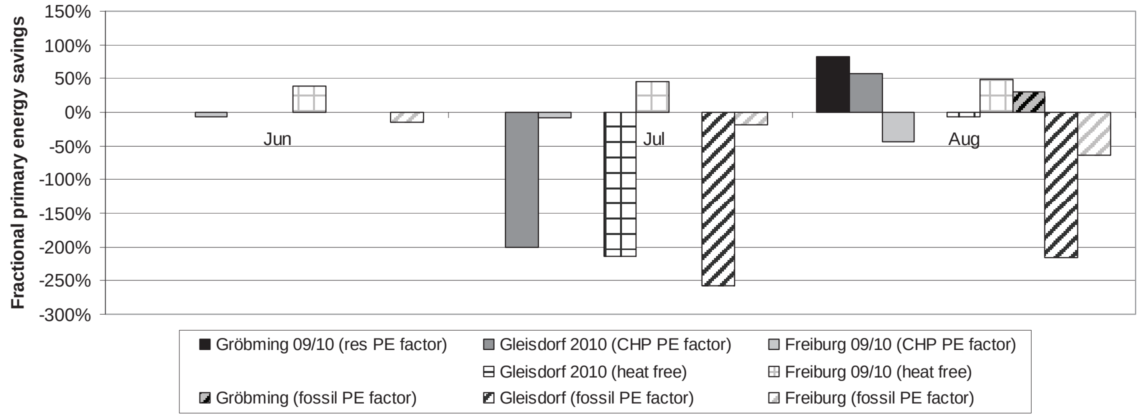


Figure 6: Fractional primary energy savings of the monitored systems with heat backup for the summer months, primary energy factors as RES (solid columns), primary energy factors with heat from CHP is for free (crossed columns) and fossil primary energy factor (hatched columns). In June only Freiburg, in August only Freiburg and Gleisdorf, in August all 3 systems.

Die beiden “heat free” Varianten sehen fast gleich aus, vielleicht sollte man es doch besser bunt machen!

Finally there are two systems with partly high negative fractional primary energy “savings” which had good or very good total electrical COPs: Freiburg and Gleisdorf. The reason for negative savings in this case is the use of a heat backup system rather than a cold backup system. In both cases, these are combined heat and power plants with primary energy factors $_{RES}$ better than the reference fossil fuel $_{fossil}$. But the solar fractions of both plants are too low to compensate for the lower seasonal performance factor of the thermally driven chiller compared to the reference compression chiller.

In the case of Freiburg, the solar fraction is still relatively high, leading to only minus 5-45% (Jun-Aug) savings in the case “CHP PE factor” but real savings of plus 39-49% (Jun-Aug) in the case “heat free”.

In Gleisdorf, the system is mainly operated with non-solar but partly fossil and partly renewable heat sources, which translates in case of “CHP PE factor” into minus 200% “savings” in July (mainly natural gas boiler in operation) and 57% real savings in August (mainly the rapeseed fired CHP in operation). In case of “heat free” in July again due to natural gas boiler minus 213% “savings” are achieved, while in August the “savings” are only minus 7% due to mainly CHP operation.

In the case “fossil PE factor” the last three columns each month are comparable in terms of heat performance quality of the systems. They show clearly that both systems in Gleisdorf and Freiburg with only cold demand and high ratio of heat backup do not reach high enough solar fraction for “virtual” primary energy savings. The system in Gröbming is able to save primary energy mainly thanks to comparably high domestic hot water demand (893 kWh) and space heating demand (139 kWh) in comparison to only little cooling demand (175 kWh).

Concluding it can be stated that in solar cooling systems with heat backup based on CHP the primary energy savings are strongly depending on the type of fuel the CHP is fired (fossil or renewable) and how the boundary conditions for the CHP operation are defined (credit thanks to electricity generation or waste heat for free).

3.5.2 Winter

For the winter months only for 5 systems there are sufficient data to analyze the fractional primary energy savings. One important factor is again the used backup heat source. Therefore, just like for the summer case first a comparison using the real existing primary energy factors is shown and then another comparison assuming the same fossil fuel primary energy factor for all 5 systems.

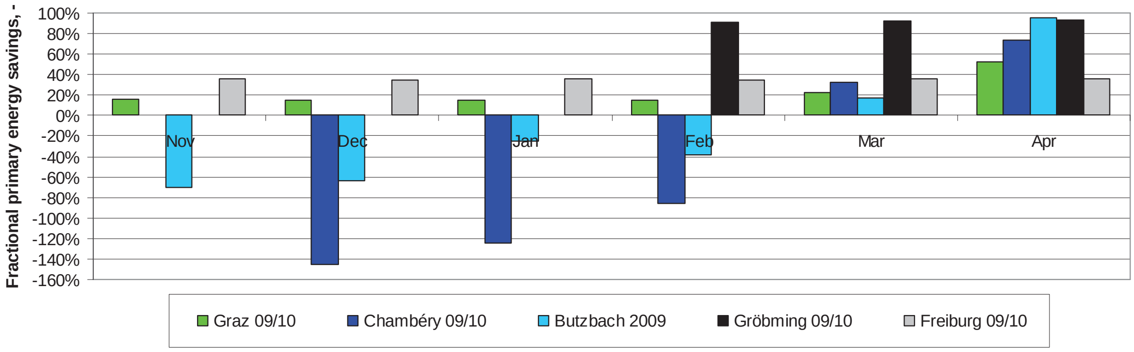


Figure 7: Fractional primary energy savings for the winter months for the 5 systems where sufficient monitoring data are available (real primary energy conversion factors)

The system in Graz uses the municipal district heating network as backup (Primary energy factor 0.96). The fractional primary energy savings are relatively small in November to February which is clear because of the weather conditions and the very small tilt angle of the collectors (11°, optimized for summer operation). However, the primary energy savings reached are still higher (because of the use of district heat instead of a purely fossil backup source). The primary energy savings assuming the primary energy factor of fossil fuels is shown in Figure 8. In March and April already much higher savings could be reached.

The system in Chambéry uses electricity as backup source. This is why the fractional primary energy savings are very negative in months where a lot of backup energy is needed. Figure 8 shows that the situation improves significantly if a gas boiler as backup is used. But in December and January, the savings are still negative because of high heat losses from the storage tank. Obviously, an electric heating element as backup source would not make sense in a real application, but has been used in this experimental installation for simplicity reasons.

In Butzbach, a natural gas boiler is used as backup heat source. But due to storage heat losses the primary fractional energy savings are negative in months without much solar gains.

The system in Gröbming was only monitored in February, March and April. Using 10 as primary energy factor for biomass, fractional primary energy savings of roughly 90% were reached. Assuming a fossil primary energy factor makes it more comparable to the other systems still. Even with that assumption siginifcant savings could be reached. Obviously, the values increase with increasing solar gains.

Finally, the system in Freiburg uses the adsorption chiller as heat pump in the winter months. This system concept reaches approximately 35% fractional primary energy savings for all winter months. These savings have two sources: the primary energy factor for the heat from the CHP unit and the heat pump effect of the chiller. Using a natural gas boiler as backup savings are still achieved, but in a range of only 10%. Savings could be increased if the solar energy is directly used for heating, which was not implemented in the installed system.

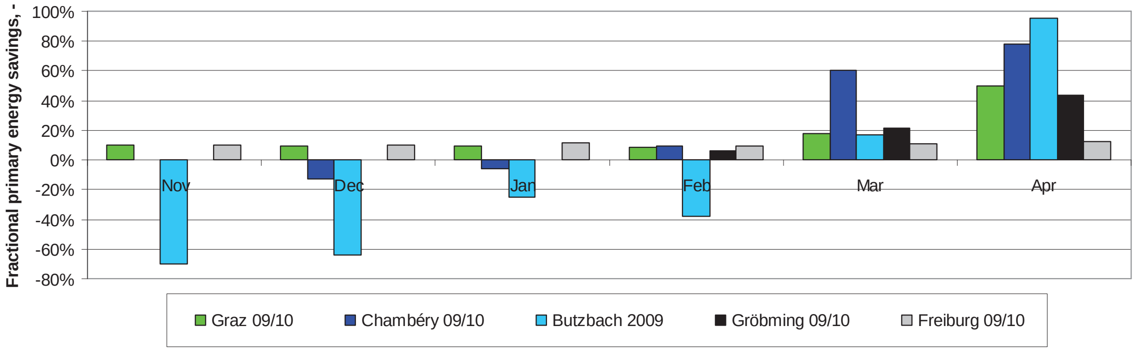


Figure 8: Fractional primary energy savings for the winter months for the 5 systems where sufficient monitoring data is available (assuming backup with gas boiler for all systems)

3.6 Collector Yield

Another interesting figure is the reached collector yield. Only 6 systems were monitored over a whole year of operation. Figure 9 shows the annual collector yield of these 6 systems. The reached values range from 250 kWh / (m² a) to slightly over 400 kWh / (m² a).

Of course the collector yield depends very much on the system concept and the energy management strategy. Monitoring shows that high values around 400 kWh / (m² a) are possible. The systems with significantly lower values very likely still have optimization potential. For example, the system in Freiburg uses solar energy only if it reaches the temperature required to drive the heat pump and not for preheating.

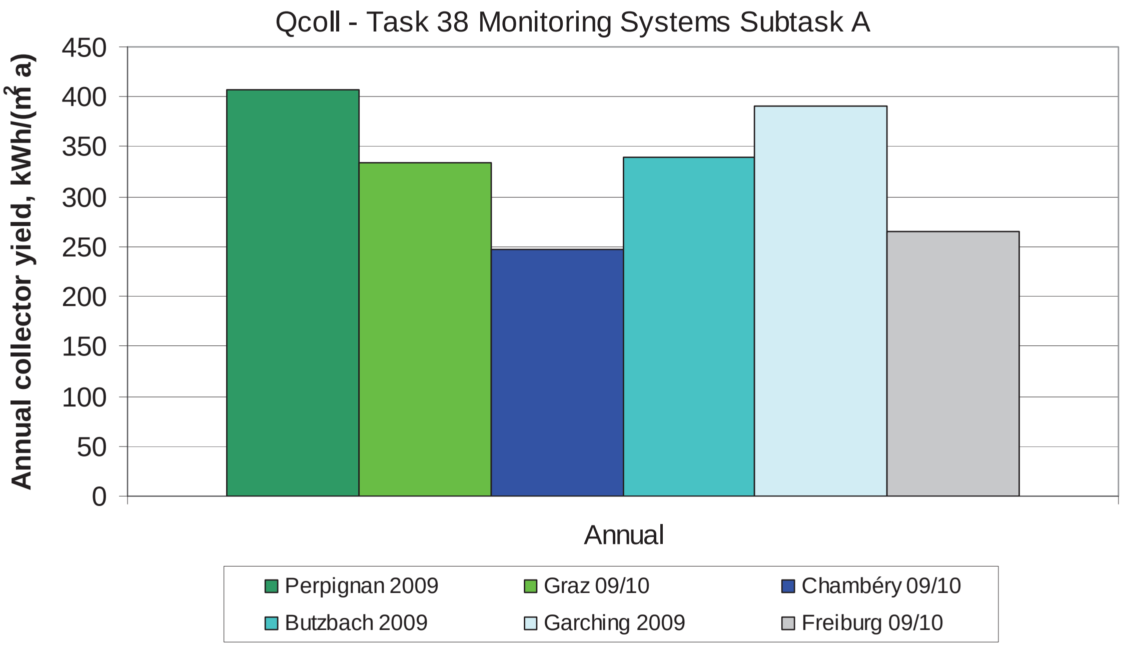


Figure 9: Annual collector yield of the 6 systems with a complete year of monitoring data

3.7 Water Consumption Cooling Towers

For 5 systems the water consumption of the wet or hybrid cooling towers was measured. A sixth system (Freiburg) had no water consumption at all because boreholes were used for rejecting heat.

The Figure 10 shows that the data scatter a lot for the different months and systems. There are some values of 0; this means that the value was not measured for this particular system and this particular month.

The reasons for scattering data are on one hand different weather conditions in the different locations and months as well as the different technologies. The systems in Perpignan and Graz use dry heat rejection systems with external spraying. The other 3 systems use wet cooling towers.

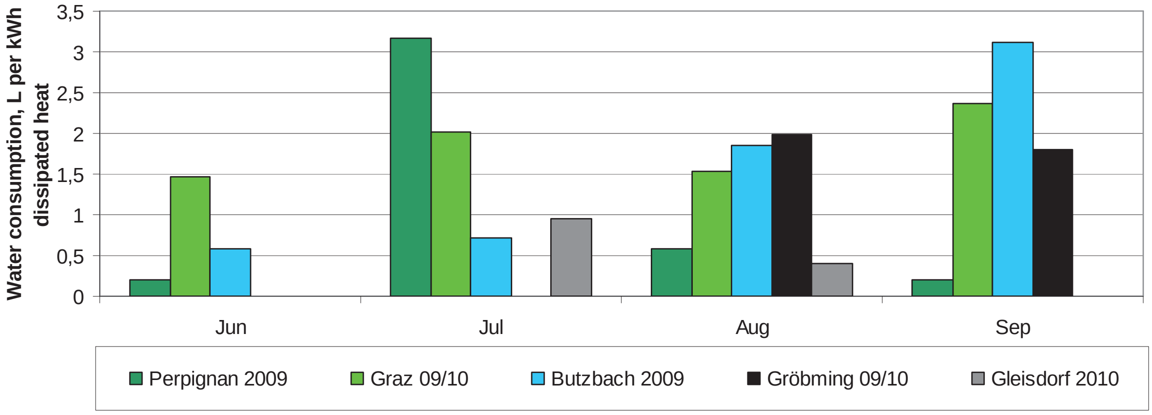


Figure 10: Water consumption of cooling towers per unit of rejected heat

On the other hand, the large differences between the systems also show that also in terms of water consumption, many systems can still be optimized.

4 Conclusions

Within this task, 11 small-scale solar heating and cooling systems have been monitored in great detail. All of them have operated and produced heat and cold reliably during most of the monitoring period.

However, the performance figures vary significantly. Some systems show very good results in terms of total electrical COPs as well as fractional primary energy savings. In some cases, this is due to the fact that during the monitoring campaign the system has been improved in a certain way.

Comparison of the performance figures revealed that all or most systems still have optimization potential. Even the systems that have bad performance figure could very likely be improved to reach good values.

The main optimization potential of most systems lies in the electricity consumption of certain components such as pumps or cooling tower fans. On one hand, the selection of these components is important. Energy efficient units should be chosen. On the other hand, the control strategy especially if operated in part load can reduce the electricity consumption significantly.

This monitoring campaign showed that

- A good system design is important in order to reach good performance figures.
- Monitoring of this kind of system is necessary to ensure proper operation because it still is a relatively new technology. It enables the system supplier to further improve the system during operation and to maximize primary energy savings of the system.

5 Bibliography

To this report related IEA SHC Task38 reports:

B3-B: Monitoring Procedure for Solar Cooling Systems - A joint technical report of subtask A and B

B3-A: Monitored Installations and Results - A technical report of subtask B (large scale applications)

Publications:

- W. Sparber, (2008). 240 - Unified Monitoring Procedure and Performance Assessment for Solar Assisted Heating and Cooling Systems, Eurosun 2008, Lisbon
- A. Thür, D. Jähnig, T. Núñez, E. Wiemken, M. Helm, D. Mugnier, P. Finocchiaro, B. Nocke; Monitoring Program of Small-Scale Solar Heating and Cooling Systems within IEA-SHC Task 38 – Procedure and First Results, Proceedings of EuroSun 2010, Graz, 2010.
- M. Helm, C. Keil, S. Hiebler, H. Mehling, C. Schweigler, Solar heating and cooling system with absorption chiller and low temperature latent heat storage: Energetic performance and operational experience, International Journal of Refrigeration, 32, 2009, 596–606.
- S. Hiebler, H. Mehling, M. Helm, C. Schweigler, Latent heat storage developed for solar heating and cooling with melting temperature 29 °C, Proceedings of Effstock 2009 – 11th International Conference on Thermal Energy Storage, Stockholm, 2009.
- Michael Himpel, Stefan Hiebler, Martin Helm, Christian Schweigler, Long term test results from a Latent Heat Storage developed for a Solar Heating and Cooling System, Proceedings of EuroSun 2010, Graz, 2010.
- Monné, C.; Alonso, S.; Palacín, F.; Serra, L., (2011). ***Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain).*** Applied Thermal Engineering, 31, 28-35. doi:10.1016/j.applthermaleng.2010.08.002.
- Monné, C.; Alonso, S.; Palacín, F., (2011). ***Experiencia en una instalación de refrigeración por absorción con energía solar***. Información tecnológica, 22 (3).
- Monné, C.; Palacín, F.; Alonso, S.; Muñoz, M.; Moreno, F., (2011). ***Analyses of a new dissipation system for a solar cooling Installation in Zaragoza (Spain***). Journal of thermodynamics. In Press Correction.
- Monné, C.; Alonso, S.; Guallar, J.; Palacín, F., (2011). ***Stationary analysis of a solar*** ***$LiBr$-H_2O absorption refrigeration system.*** International Journal of Refrigeration. Article in Press. doi: 10.1016/j.ijrefrig.2010.11.009.
- Tomas Núñez, Florian Mehling: "Heating and Cooling with a Small Scale Solar Driven Adsorption Chiller Combined with a Borehole System – Recent Results-" 3rd Int. Conference Solar Air-Conditioning, Sept. 30th to Oct. 2nd, 2009 Palermo, Italy.
- Tomas Núñez: "Small Capacity Solar Cooling: Two Systems Installed in the Frame of the Project SOLERA", Proceedings of EuroSun 2010, Graz, 2010.
- Palacín, F.; Monné, C.; Alonso, S., (2010***). Evaluation, diagnosis and improvement of a solar cooling plant by means of experimental analysis and dynamic simulation***. 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).

- Palacín, F.; Monné, C.; Alonso, S.; Izquierdo, J. M.; Heredero, J., (2010). ***Design and operational experiences of an alternative heat rejection sink for an existing absorption solar cooling system.*** 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Experiences on heat rejection sinks of a solar absorption cooling system.*** 3rd International Conference on Passive and Low Energy Cooling for the Built Environment (PALENC 2010), 5th European Conference on Energy Performance & Indoor Climate in Buildings (EPIC 2010) and 1st Cool Roofs Conference. 30th September to 2nd October 2009. Rhodas Island (Greece).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Comparing different sinks of heat rejection of an existing solar powered absorption cooling system.*** International conference on renewable energies and power quality (ICREPQ'10). 25th to 26th Marz 2010. Granada (España).
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Instalación experimental de refrigeración solar - primeros resultados.*** XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th to 21th June 2008. Vigo (España). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte I)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th junio 2008. Vigo (España). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte II)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th junio 2008. Vigo (España). SPE.

6 Appendix: Detailed Report for Each System

Appendix 1	Austria: Bachler, Gröbming
Appendix 2	Austria: SOLID Office Building, Graz
Appendix 3	Austria: Municipal Administration MA34, Vienna
Appendix 4	France: Résidence du Lac, Maclas
Appendix 5	France: CNRS PROMES Research Center Office, Perpignan
Appendix 6	France: INES Research Center Offices, Chambéry
Appendix 7	Spain: Gymnasium of the University of Zaragoza, Zaragoza
Appendix 8:	Germany: Technical College Butzbach
Appendix 9	Germany: ZAE Bayern Office Building, Garching
Appendix 10	Germany: Fraunhofer ISE Canteen, Freiburg

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 1
Monitoring Results of Bachler/Gröbming

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: November 2010

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Contents

1 Background

In the beginning the system was designed as a solar assisted heating system for two office rooms, two apartments and a spa area. Furthermore the swimming pool is heated with the solar energy. As a backup the hot water from a near biomass plant is used. The solar cooling concept was added to the system in a later planning stage. Therefore the absorption chiller as well as the wet cooling tower was placed outside the building in the garden.

The solar cooling plant at the Bachler GmbH training and office building was installed in spring 2007. It is delivering its cooling load to the office trough concrete core activation. This plant was built as an attachment to an existing solar combi system.

Figure 1: Exterior view of the office building in Gröbming, Styria [Bachler]

The system is installed without a conventional backup including a 12 kW Pink absorption chiller working with ammonia/water. The 46 m^2 solar panels are flat plate collectors (type Goliath from Neuma-Solar) integrated in the façade and also placed in front of the building. In the utility room three 1500 liter hot storages as well as all auxiliary hydraulics are placed. The absorption chiller together with the wet cooling tower is positioned outside the house. The hot water of the solar thermal collectors is used for domestic hot water production, for warming the water of the swimming pool and in the heating season it is also used to provide space heating combined with local district heating.

2 System Design

In Figure 2 the hydraulic scheme of the installed system in Gröbming is shown. There are two solar collector fields, one on the ground with 30.2 m^2 and the second is integrated in the façade with 15.8 m^2. From the solar collectors the heat flows through a heat exchanger to the hot water storage. Three 1500 liter water storages connected in series are used to store the solar energy. Depending on the return temperature from the heat exchanger the medium is stratified to the hot medium or cold storage. In winter the return flow from the distribution systems is delivered into storage 1 and in summer, when the cooling mode is running, into storage 3. Out of storage 3 the hot water can be distributed to different recipients, the domestic hot water station, the absorption chiller, the swimming pool as well as the space heating for the building. This can also be done alternatively from storage 1 or 2 if the temperature is sufficient.

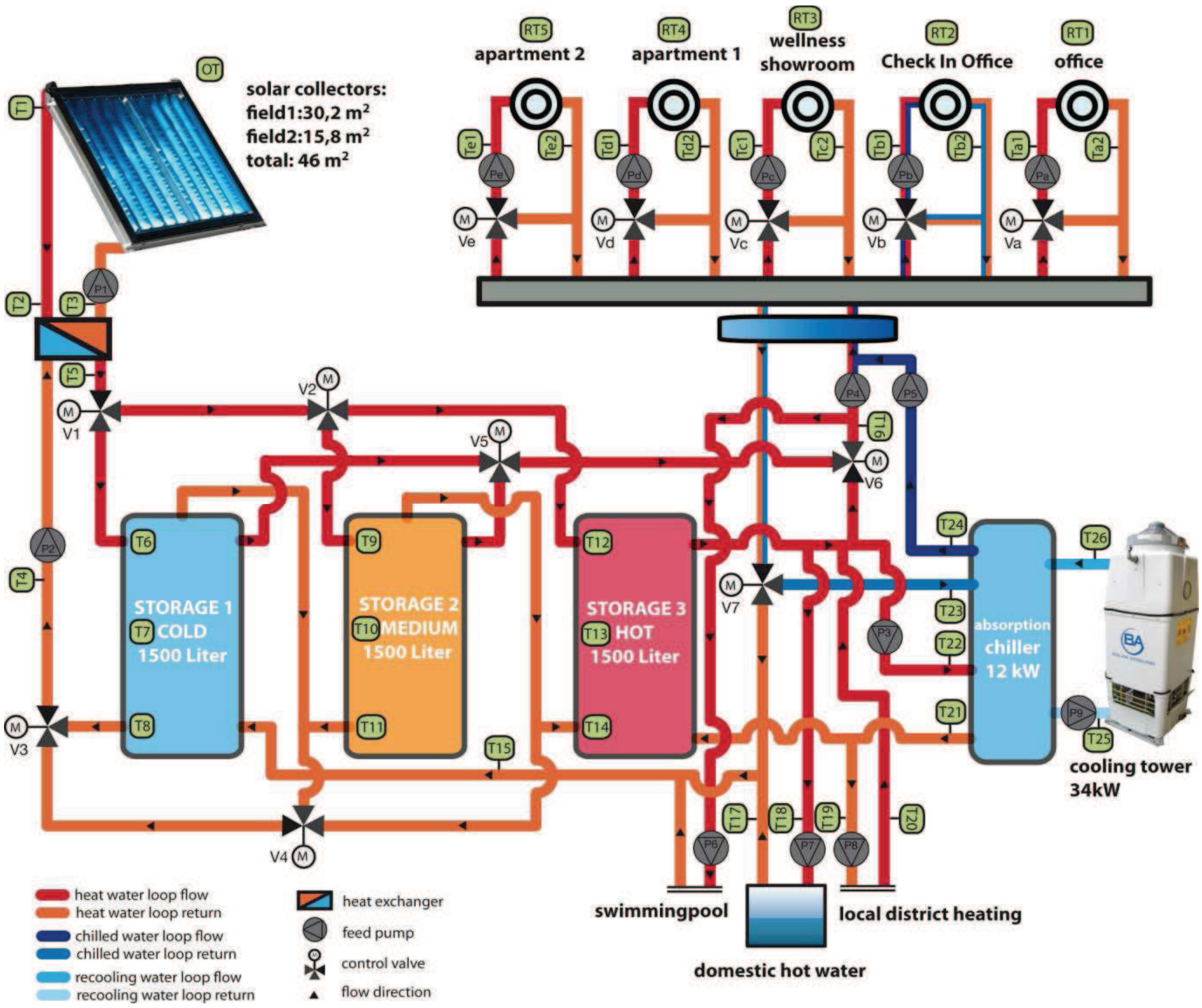

Figure 2: Hydraulic scheme of the solar cooling system in Gröbming

From the central distribution station the hot water is spread to the different recipients. In summer time, when the cooling mode is enabled, the chilled water is also distributed through the same central distributor station.

The recooling water loop connects the absorption chiller directly to the wet cooling tower. The absorption chiller PSC-12 is located outside the building. As a consequence of this location, twice a year a maintenance worker has to discharge and charge the hydraulic cycles of the absorption chiller as well as of the cooling tower before and after the winter period to avoid freezing damages.

All in all nine auxiliary pumps as well as five distribution pumps are installed in the system. To control the plant two Technische Alternative (TA) UVR1611 controllers are included. The monitoring task is done by separate devices installed by the company RKG.

This system design was chosen because of the easy and cheap integration feasibility in the solar combi system. The solar system and even the cold distribution system were there anyway, only the cold production had to be added.

The main problems encountered were caused by the solar system. It was designed originally as a low-flow-system. To drive the absorption chiller higher volume flows are necessary to reach the nominal power. Therefore the pumps of the solar pumping station had to be changed and the loading strategy for the buffer tanks had to be optimized.

Figure 3: View of the utility room [upper left, Bachler], the outside situation with the wet cooling tower and the absorption machine [upper right, Bachler] and the inside of the cooling machine [lower left / right, Pink]

3 Control Strategy

In Figure 4 the control strategy of the solar cooling plant in Gröbming is illustrated. All the temperatures and nomenclatures are linked to the hydraulic scheme (Figure 2). Only the cooling cycle and the management of the solar cycle are described here.

Certainly the combination of the whole system, including the space heating, demands special attention. Through the complex hydraulic layout of the system the control strategy of the solar cooling plant had to be harmonized with all other components such as the domestic hot water station and the pool heating. The main goal was to avoid a clocking behavior of the chiller.

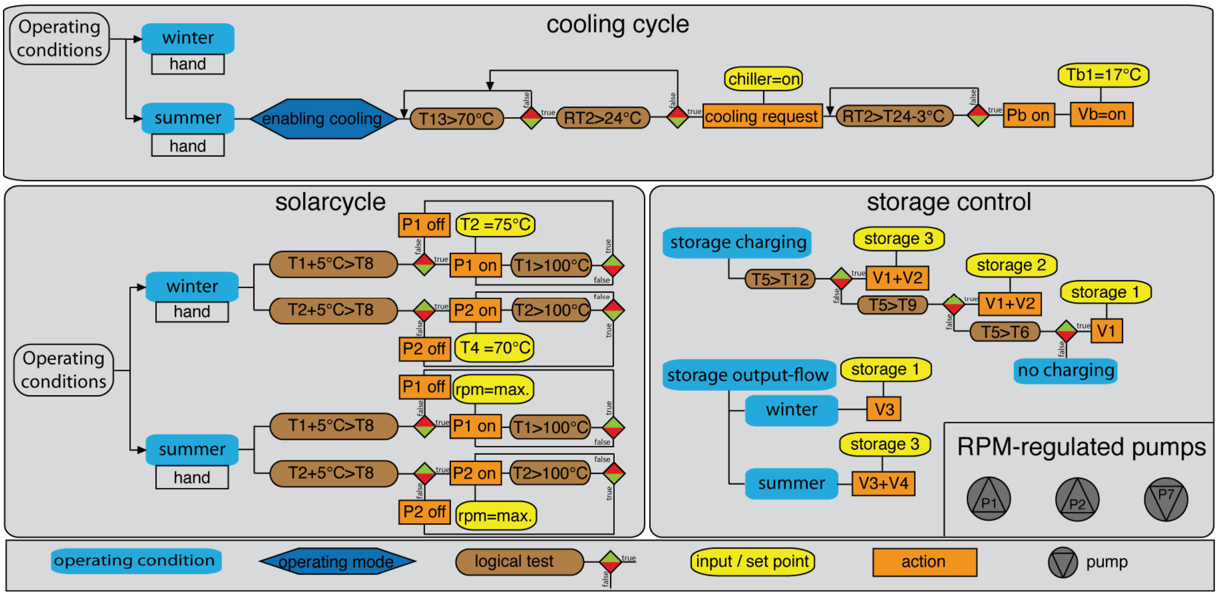

Figure 4: Control strategy of the solar cooling plant in Gröbming

Cooling cycle

To eliminate any control failure regarding the specific winter and summer behaviors the change between winter and summer operation is done by the maintenance technician. If summer mode is switched on cooling is basically enabled. The first continuous logical test is checking if temperature T12 is higher than 70℃. A hysteresis of 7 K is defined to secure that the chiller is not switched on and off too often. If the test turns out true another logical test for the office room is carried out. A cooling request is stated if the room temperature (RT2) exceeds 24℃ and the chiller is switched on. In the next stage the chilled water supply temperature has to be more than 3℃ below the room temperature, then the distributor pump is started and the control valve (Vb) is enabled to regulate the flow temperature at 17℃.

Solar cycle

Switching between summer and winter is done manually. Temperature T1 plus a margin of 5℃ has to be above the lowest temperature of stora ge 1 (T8) to start the primary solar pump. In summer this rpm-regulated pump (P1) is not controlled to any set temperature; it is always running at maximum speed. In winter the primary solar pump is regulated to a set temperature of 75℃. Because there are two solar co llector fields, T1 is averaged out of two collector temperatures measured in those two fields. Analogically the secondary solar cycle is turned on with T2. It is not regulated in summer but regulated to a set temperature of 70℃ in winter. To avoid too high pressure in the hot storage, the solar cycle is switched off if either T1 or T2 is above 100℃.

Storage management

As shown in Figure 2 the entire system has three 1500 liter hot water storages. To use this volume for storing the solar energy effectively, it is important to have a straight forward storage control strategy as it can be seen in Figure 4.

Depending on the temperature T5 the storages 1 to 3 can be charged from the solar collectors. This is especially useful in winter time and transition periods where solar energy should be stored as much as possible. In summer time, when the solar cooling plant is in operation, only the hottest storage (storage 3) is in use. The main reason for not using the entire storage size is the driving temperature of the absorption chiller. Following the manufacturing information of the chiller, a constant driving temperature of 75℃ to 80℃ is recommended. Nevertheless the machine is running down to 65℃ driving temperature, but with poor thermal COPs. To reach those heating temperatures for the chiller it would take too long for the solar plant to charge all three storages. A time offset of cooling demand and cooling distribution would occur. Using only storage 3 brings down the required temperature difference done by the solar panels and raises the volume flow through the collectors. This reduces the time in the morning until the chiller can be started.

4 Monitoring Equipment

4.1 Installed Equipment

Figure 5 shows the symbolic scheme of the system indicating all monitored energy fluxes.

The heat flow from the district heating Q2 is partly used direct and partly heating the hot water storage. This fraction cannot be calculated and therefore Q2 is handled as input to the storage. The heat back up is used for space heating, domestic hot water as well as an unmeant heat back up for the chiller. The heat source from the local district heat is additionally monitored from the district heat supplier. The small hydraulic switch linked to the central distribution station is not taken into account. Cold losses due to the switch are expected quite low.

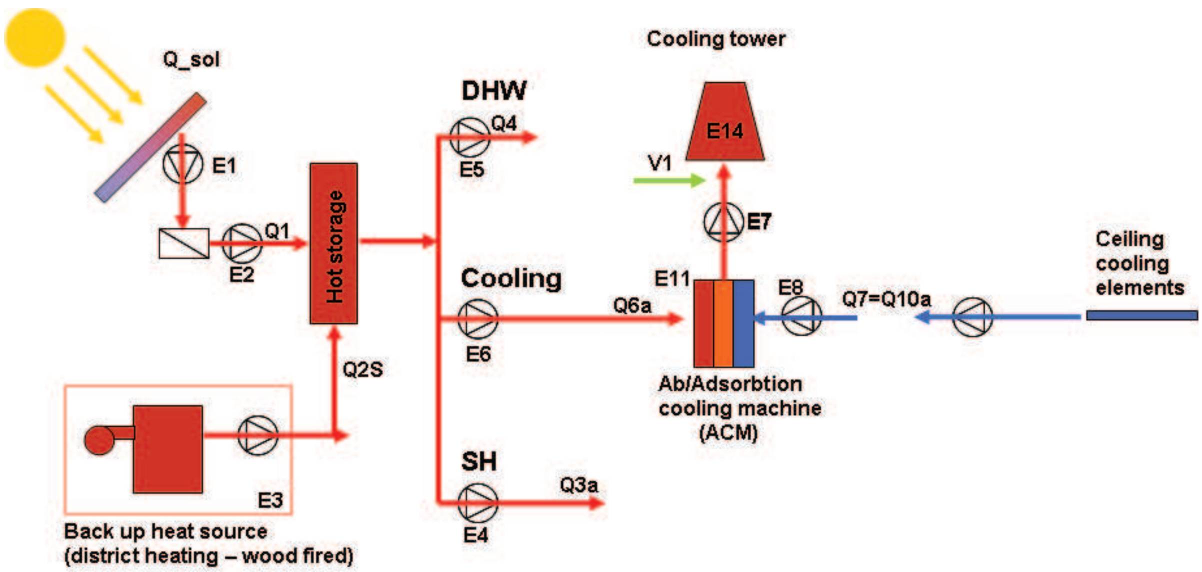

Figure 5: Monitoring scheme of the plant including electricity and heat measurement points

All in all seven heat flow meters and seven electricity meters are installed. The Kipp & Zonen SPlite pyranometer is used to measure solar radiation. The data points for monitoring only are logged separately. The two control units (Technische Alternative TA UVR1611) provide further information (temperatures, pump status,…), which is necessary for a detailed analysis of the plant.

The data is collected by two IQ3 controllers. These controllers are Building Management System controllers, which use Ethernet and TCP/IP networking technologies. Each controller incorporates a web server, which can deliver user-specific web pages to a PC or mobile device running internet browser software. If a system is set up with the correct connections, a user with the appropriate security codes can monitor or adjust the controller from any Internet access point. In the first cooling season (2009) the operator station was located externally at the office of the chiller manufacturer.

Due to a combination of monitoring problems during summer 2009 not enough valuable and comparable monthly data could be collected, but still some useful system operation experiences were made.

An overview of all monitored data points is given in Table 1.

Table 1: Monitoring data points in Gröbming

Quantity	No.	Type	Accuracy
monitoring (IQ3) / control inputs (TA UVR1611)			
temperature (UVR1611)	18	KTY 81-210	± 1% (25°C)
radiation	1	SP-Lite / Kipp&Zonen	temp. Dependence +0.15%/°C directional error ±5% at 80°
temp./relative humidity	1		±0.5% °C / ±3%RH
heat meter energy/volume/power/flow rate/return-supply-temp./ΔT	7 (x7)	Kamstrup Multical 601	E_C= ±(0,15 +2/ΔΘ)% E_T= ±(0,4 +4/ΔΘ)%
electricity meter energy/power	7 (x2)	Kamstrup 162B	Class A
logged control outputs (TA UVR1611)			
Speed control	2		
on/off	7		
valve position automatic	10		
valve position hand	2		
…. **sum**	**148**		

4.2 Period of Measurement

The data logging of the control units (TA UVR1611) was started in January 2010.

The first monitoring equipment was installed and commissioned within the construction of the cooling plant in 2007. These measurements were mainly completed to observe the chiller performance. A detailed energy balance of the absorption chiller could be calculated.

The monitoring equipment for level III measurements was completed in July 2009. With this equipment only energy balances can be calculated. For a detailed analysis additional data logging of the control units was implemented in January 2010.

Due to problems in data logging (connection to operator, constant time intervals in logging,...), the operator station was moved to the plant. Now it’s located in the utility room and it’s logging the data from the monitoring and control devices.

5 Monitoring Results

5.1 Annually / Monthly Data

Figure 6 and Figure 7 illustrate the monitoring results for August and September 2009. The black line indicates the space cooling power for the office. The bars show the solar-, the district heat-, the domestic hot water- and the space heating power.

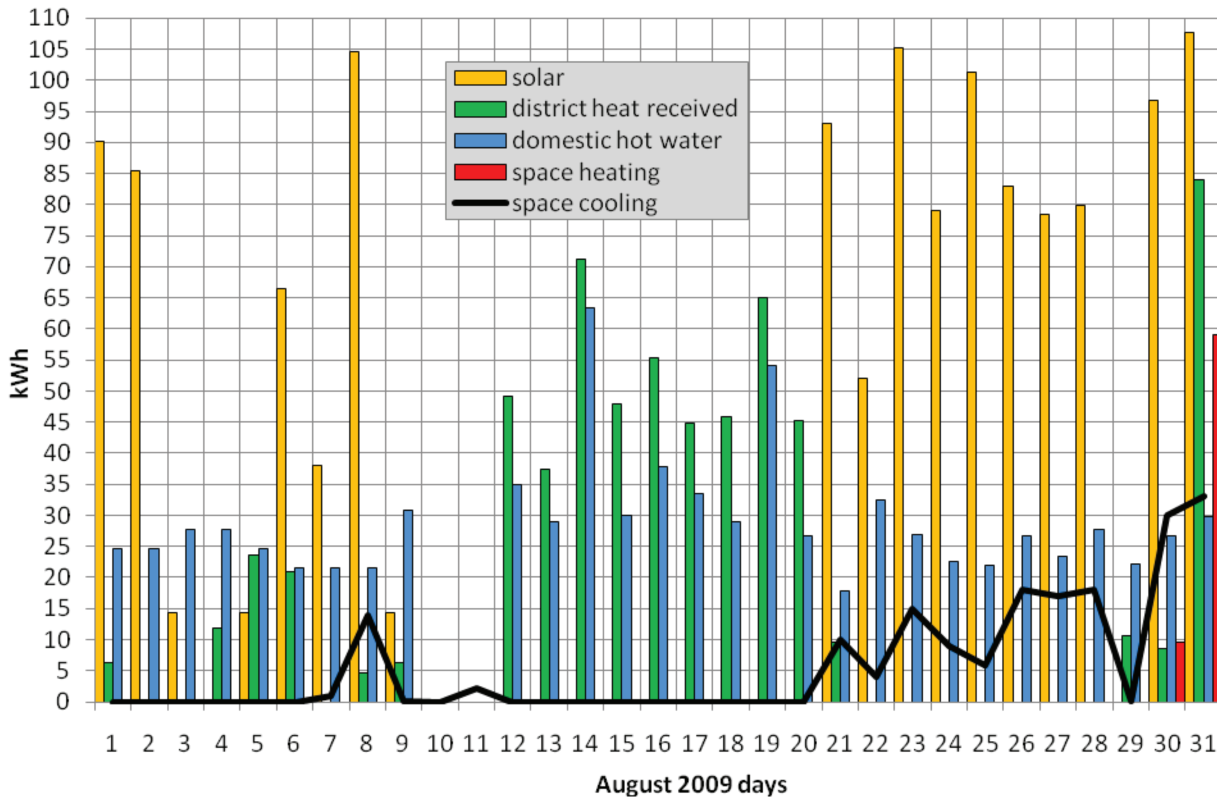

Figure 6: Daily energy fluxes for the plant in Gröbming in August 2009

Before August 2009 there were no cooling activities of the solar cooling plant. Reasons were the cold outside temperatures in May and June especially during night times and therefore no clearance through the control unit (Figure 4). A three day average value for outside temperature was calculated and used. Due to the low night temperatures in Gröbming this value never cut across the limiting temperature to enable the cooling mode.

In the beginning of August this logical test was skipped and the summer mode was turned on manually. On the 8th of August the chiller started for the first time in this summer. Unfortunately one day later one of the two controllers broke. The broken unit was controlling the solar- and the cooling cycle. After an exchange the chiller started every sunny summer day until the end of September. On the last day in August space heating was switched on for the first time. Gröbming is a small township in northern Styria located 776 m above sea level, where night temperatures fall easily below 10-15℃ even in summer. The outside temperature regulated heating system heated regularly during the nights in September.

To heat in the night and to cool during the day raises the costs of providing a pleasant room climate to the office and increases the primary energy consumption of the system as well.

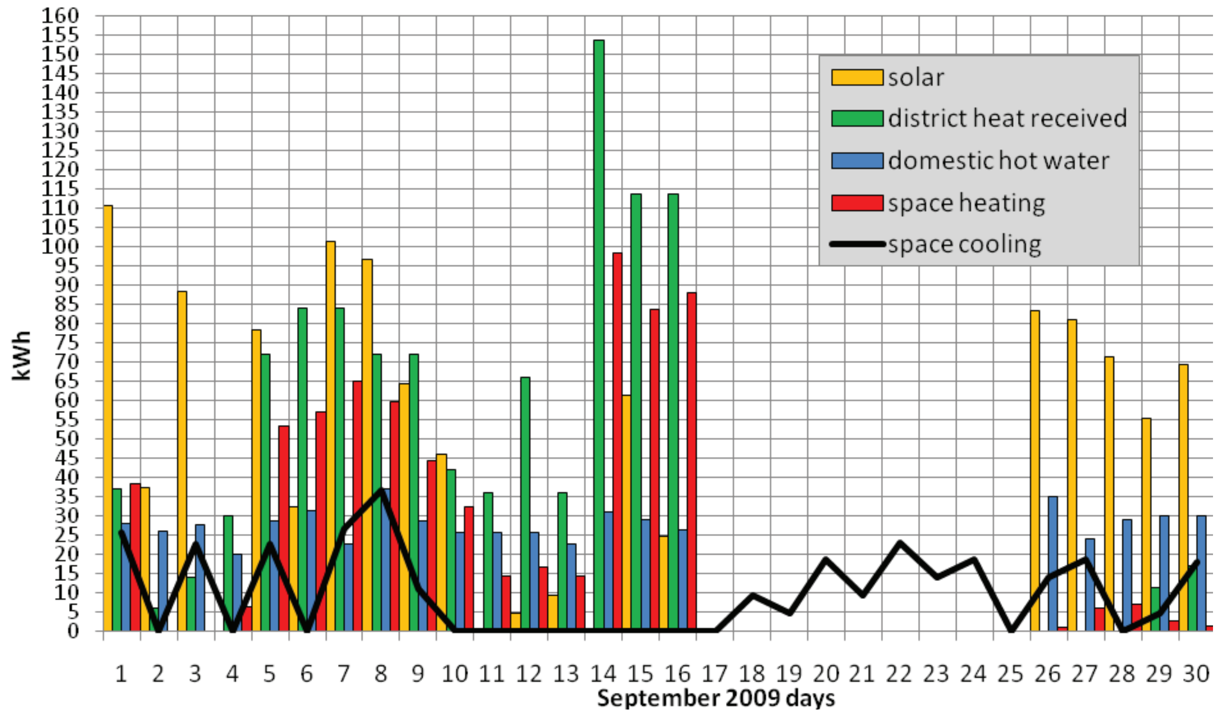

Figure 7: Daily results for the Bachler plant in Gröbming for September

The domestic hot water demand (blue bars in Figure 7) has a stable value of around 25 kWh per day. A nearby biomass heating plant provides district heat to the house shown with the green bars in the diagram. This heat source is planned as a backup system for the domestic hot water production in summer and also for the space heating in winter.

The energy balance of all incoming and outgoing heat flows outlines high overall system losses as shown in Figure 8. An average loss of 28.6 kWh per day is calculated for the period between August 21, 2009 and September 16, 2009. These losses are even higher than the average hot water consumption (25kWh).

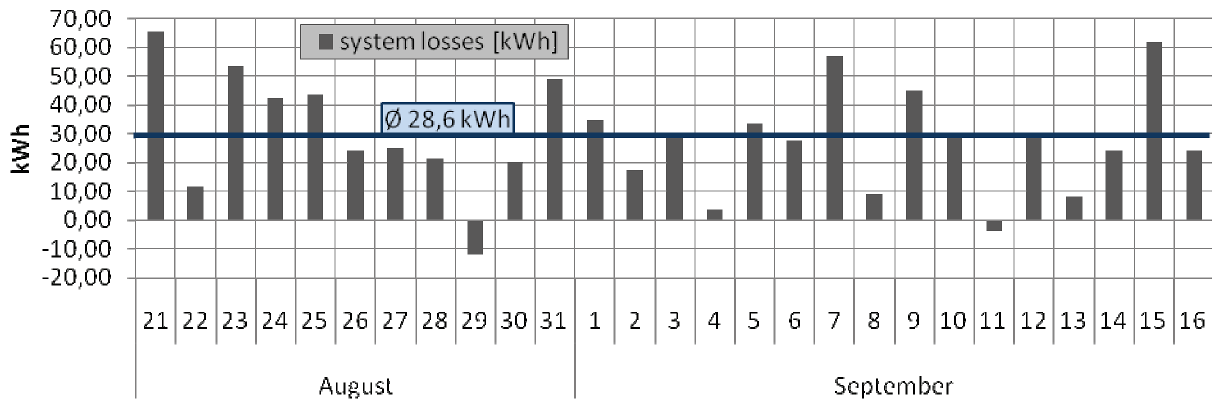

Figure 8: System losses of the solar heating and cooling plant in Gröbming

5.2 Analysis of Typical Days

Figure 9 shows data regarding the ammonia-water chiller on an average sunny day in September 2009.

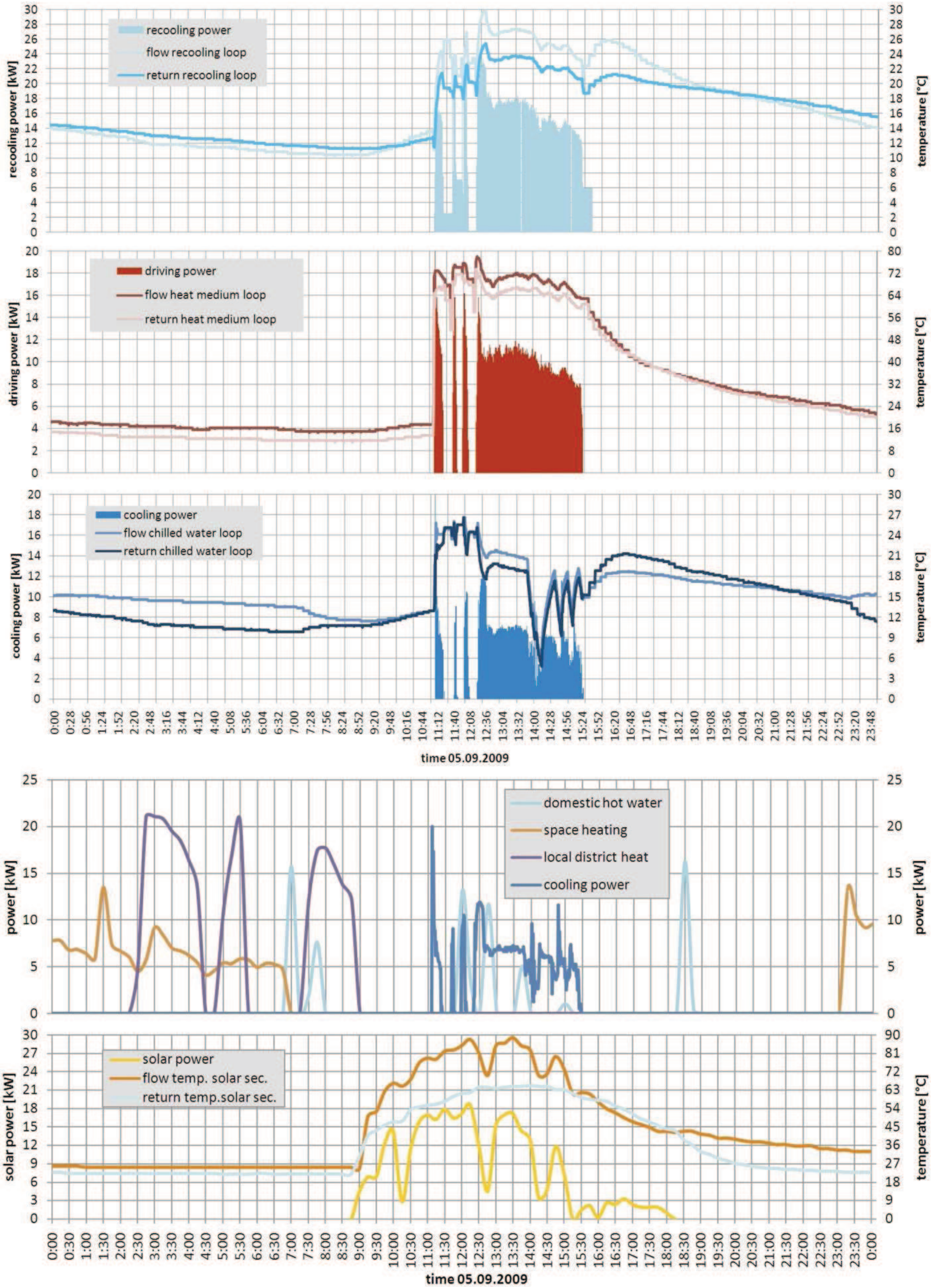

Figure 9: Absorption chiller and system analysis of the SHC plant on September 5, 2009

In the first three charts the cooling-, driving- and recooling power is drafted together with the linked supply- and return temperatures. In the beginning the chiller shows a clocking behavior, starting three times before running in a stable condition. The reason for switching on and off is the insufficient useable heat quantity (with a temperature base over 63℃) in the hot storage. At approximately 12:30 p.m. the incoming solar yield reaches sufficient values to keep the machine running. The solar plant is designed for a low flow configuration in order to assist the space heating of the house in winter times. This has certain disadvantages concerning the cooling mode. On this day the chiller has an average power share of around 6/10/16 kW (cooling/driving/recooling). For recooling the stable chiller inlet temperature values of approximately 22℃ were reached. The heat rejection system is working well throughout the day. Furthermore the driving heat inlet temperature of the absorption chiller reaches an average value of 69℃ for the day.

In the beginning the cooling outlet temperature towards the core activation shows stable values of approximately 19℃. At around 14:00 p.m. the flow and return temperature drops rapidly down to a value of 5℃. After a fast temperature rising up to 18℃ the same characteristic temperature drop repeats two times. A reason for the temperature drop could be the closing of the regulation valve V_b correlating with a domestic hot water priority function. If the valve is closed the chiller cools down in its short closed cycle as far as possible until the valve opens again.

These low cooling temperatures affect thermal performance of the chiller. In Figure 10 the thermal COP during September 5, 2009 is shown. The bad performance can be identified exactly at the same time, when the cooling temperature drops down. Due to an internal ammonia storage inside the chiller there are some peaks in those periods, where the COP exceeds the value of 0.8. Generally the Pink absorption chiller handles the rough operating conditions quite well and balances some of the outside influences.

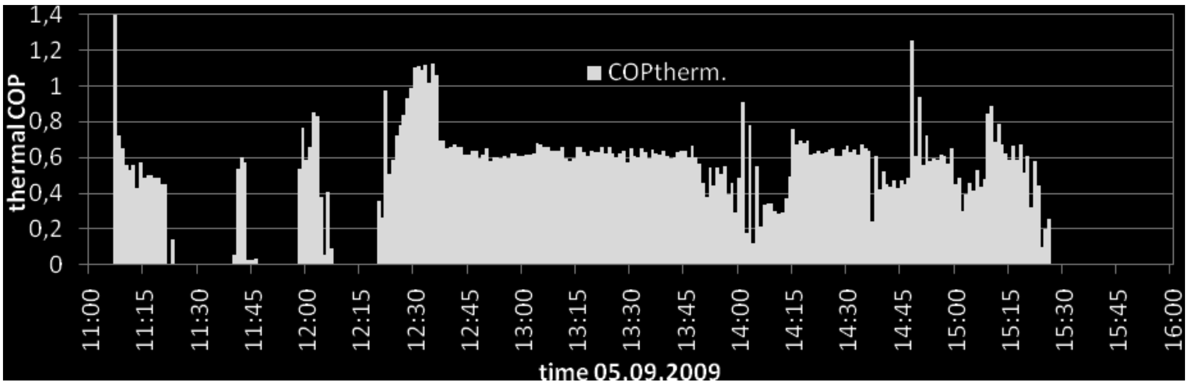

Figure 10: Thermal COP of the Pink chiller on September 5, 2009

Thermal and electrical COPs for the period between August 21, 2009 and the end of September are shown in Figure 11. An average thermal COP for that period of 0.57 could be reached. Electrical COPs range in daily values from 0.5 to 5 and achieve an average value of 3.1. In the period between the September 11 and 18, 2009 no results were monitored due to a data processing problem of the external operator station.

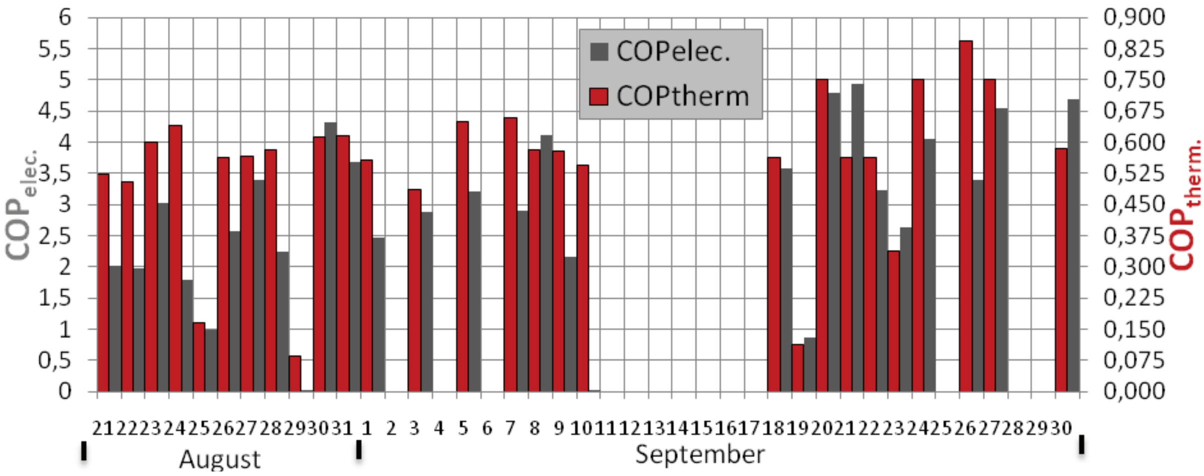

Figure 11: Daily thermal and electrical COPs for August and September

6 Outlook

Most of the occurred problems during summer 2009 happened in the field of the monitoring system. For that reason a redundant system including protection against energy blackouts is to prefer. Several changes of the system have already been succeeded.

Reengineering the monitoring system includes following five points:

- Room temperature and several other data points in the monitoring system
- switching the monitoring period of all data points towards minute intervals
- changing the logging function to an UPS protected local computer
- repositioning of the outside temperature- and humidity measurement sensor
- linking the data of the TA controller with the monitoring infrastructure of the IQ3

These suggestions of improvement regarding the monitoring system are expected to raise the reliability and accuracy of the monitored data. Furthermore a calibration of the temperature sensor is recommended in order to minimize the measurement errors.

Summarizing the five main practical suggestions for improvements:

- adapting the control system in order to make clear division between heating and cooling periods
- including the cooling cycle of the DHW priority rule
- excluding the logical test RT2>24℃ of the control strategy in order to use the concrete core activation as a storage and make the control system easier
- changing the switching operations between summer and winter to automatically by including the valves V3 and V4 in the controller
- rising the starting driving temperature in the control strategy, if there are still problems with a clocking behavior of the chiller

Expected improvements of these suggestions are reliable cooling and monitoring operations as well as higher thermal- and electrical COPs. In order to simulate other changes of the system in the course of the Solar Cool Monitoring project further computer simulations will be carried out.

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 2
Monitoring Results of SOLID CoolCabin

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: October 2010

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Contents

1 Background

The office building was renovated in 2004 and the solar cooling device was installed in 2008. The office façade has a south and west orientation. To reduce the solar gains external shading devices are installed at each glazing. Because of internal gains and ventilation via windows, active cooling is indispensable. The solar cooling equipment is installed in a so called “CoolCabin” placed in front of the office. The solar collectors are installed on the roof. The hybrid cooling tower is placed on the flat roof of the office building. The cooling load of the office rooms is taken out via ceiling cooling elements.

Figure 1: View of the CoolCabin in front of the office building [SOLID]

A closed absorption cycle for generating cooling energy is employed. Autonomously solar thermal generated heat by high temperature flat plate collectors is used to regenerate the process. Within the absorption cooling machine water is used as refrigerant and lithium bromide is used as solvent. The cold water generated by the absorption cooling machine is used to cover internal and external heat gains. In winter the solar collectors are in assistance to the conventional space heating. In summer and winter the solar generated heat is stored in one buffer storage and all energy demand is taken out of it. The additional heat from the district heating is not stored in the tank but directly carried to the space heating system. A special application is the direct usage of the hybrid cooling tower for free cooling via chilled ceilings.

2 System Design

Figure 2 shows the functional scheme of the solar system. Hot water is produced in the 57.6 m^2 flat plate collectors, which are used as roof for the CoolCabin. The collector field is aligned south with a slope of 11°. The collector pump P1 drives the hot water to the heat exchanger. Pump P2 delivers the medium in the secondary solar cycle to the hot water storage tank, which has a volume of 2000 liter. Valve V1 controls the position (top or medium) where the hot medium is stratified into the storage.

In summer, when there is a cooling demand, the hot water flows to valve V4 through pump P3 to the absorption cooling machine (ACM). In winter, when space heating is needed, the hot water flows through valve V3 pumped by P6 directly to the distributor.

The ACM has a maximum cooling capacity of 17.6 kW, a rated COP of 0.65 and was manufactured by the Japanese company Yazaki. The hot water loop is piped back to the hot water storage or mixed in V4 with the flow to control the hot water inlet temperature.

The cooling tower can work in two different operation modes. Either the recooling water is going directly to the ACM through P5 or in free cooling mode to the heat exchanger by P7. Switching between chiller and free cooling is done with the valves V5 and V7. The chilled water produced by the ACM or from the heat exchanger (in case of free cooling) flows into a 200 liter hydraulic switch and further on to the distributor. Through the radiant ceiling system of the office building the chilled water is distributed.

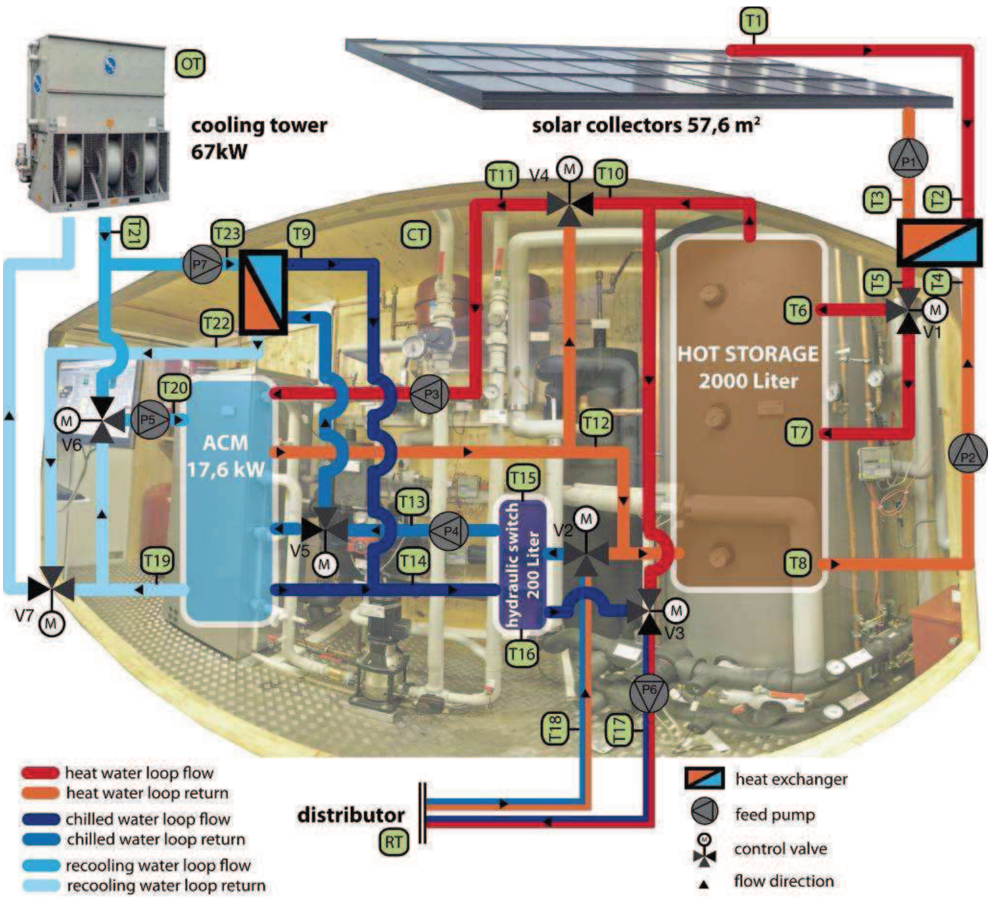


Figure 2: System scheme of the CoolCabin

Reasons for designing the system the way it has been designed [SOLID]

- The system is designed for 40-50 W/m² cooling load
- Restrictions in tilt angle for collectors (see Figure 1)
- No cold back-up for saving electricity and to see how this works in Austria
- Storage size 35 l/m² for heating support in shoulder season. Could be smaller for cooling
- Free cooling for research purposes
- Hybrid cooling tower mainly for free cooling. Over dimensioning of cooling tower for higher cooling output of cooling machine than nominal power.

Problems encountered [SOLID]

- Works well without backup. Some south-facing offices become too hot.
- Free cooling like it's done here is not efficient and useful for building cooling in summer. Could be more effective using free cooling in other ways and times.
- Sometimes it is too humid inside the building in summer time as only chilled ceilings are used and no air treatment is done.

Figure 3: View of the inside of the CoolCabin [SOLID]

3 Monitoring Equipment

3.1 Installed Equipment

The equipment is completed for IEA Task SHC Task 38 monitoring level III. To succeed simulations the relative humidity is measured additionally. The amount of heat and electricity meters is reduced to a minimum. To differentiate between the two modes (free and thermal cooling) the valve positions are logged and used for this allocation.

The conventional heating system is not included in the monitoring but is carried out through the clearing of the district heating. The district heating is directly carried to the distribution system.

The data is recorded each minute except V1 and E20. The water consumption of cooling tower (V1) is monitored by hand each week. E20 is logged each 10 minutes by a separate device - an electrical socket counter.

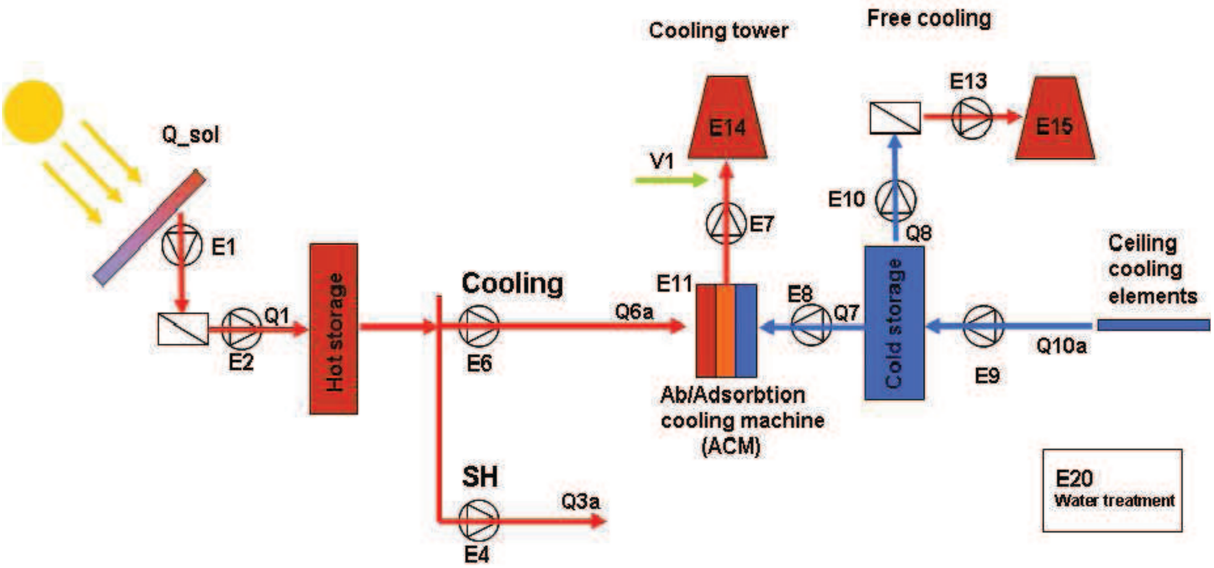


Figure 4: Monitoring scheme of the CoolCabin

Following data points are from the same device or measured together. The allocation between thermal and free cooling is done by the valve positions mentioned above. Nevertheless the important results for Task 38 calculations can be delivered.

E1 + E2	measured together
E6 + E11	measured together
E4 + E9	same pump, allocation by valve position V2, V4 (heating, cooling)
E7 + E13	same pump, allocation by valve position V5, V7 (free, thermal cooling)
E8 + E10	same pump, allocation by valve position V5, V7 (free, thermal cooling)
E14 + E15	same device, allocation by valve position V5, V7 (free, thermal cooling)
Q7 + Q8	same heat meter, allocation by valve position V5, V7 (free, thermal cooling)

An overview of all monitored data points is given in Table 1. The controlling and monitoring issues are done by three programmable logic controllers. Overall 111 points are logged on a local PC. The access to this data store is provided by internet and a remote control system.

Table 1: Data points of the CoolCabin

Quantity	No.	Type	Accuracy
monitoring / control inputs			
temperature	35	PT1000	class B + calibration
pressure	4	huba control, OEM relativ-Drucktransmitter	endvalue ±1.5% FS temp. dependence ±1.5%/10℃
radiation	1	SP-Lite / Kipp&Zonen	temp. Dependence +0.15%/℃ directional error ±5% at 80°
temp. / relative humidity	1		±0.5% ℃ / ±3%RH
heat meter (energy/volume/power/flow rate/return-supply-temp./ΔT)	4 (x7)	Landis + Gyr, UH50, ultraschall	class II; 2+0.02 q_p/q% (max 5%)
electricity meter (energy/power)	6 (x2)	Görlitz DCi 230V/400V, 65A	class I; IEC 62053-21 (±1%)
logged control outputs			
speed control	2		
on/off	3		
valve position	5		
failure signal / set temp. / …			
sum	**111**		

3.2 Period of Measurement

The monitoring system of the CoolCabin is in operation since the commissioning in September 2008. Additional monitoring equipment for level III was completed in June 2009. Monitoring is ongoing.

The patency of data logging is very high (>96%). Nameable gaps crop up only when the CoolCabin is out of service. One problem occurred was a roof avalanche that got down on the radiation sensor. Consequently no solar radiation data was monitored from Jan10 to April10. The missing data was calculated out of Q1 using a monthly efficiency of the solar collectors including the heat exchanger.

4 Monitoring Results

4.1 Annual / Monthly Data

As an overview the solar energy source management is shown in Figure 5. Start of monitoring is July 2009 at the left side of the graph. The first cooling season lasts until October 09 including free cooling. Q3 represents the sum of solar heating and the heat from the conventional system. The solar fraction in summer is 100% (no heat backup) in winter the solar fraction for heating drops down from approximately 20% in Nov09 to 7% in Dec09. The highest fractions occur in spring (30% in Feb10; 60% in March10). The overall solar efficiency is approximately 30%, including efficiencies up to 40% in summer and 10% during winter.

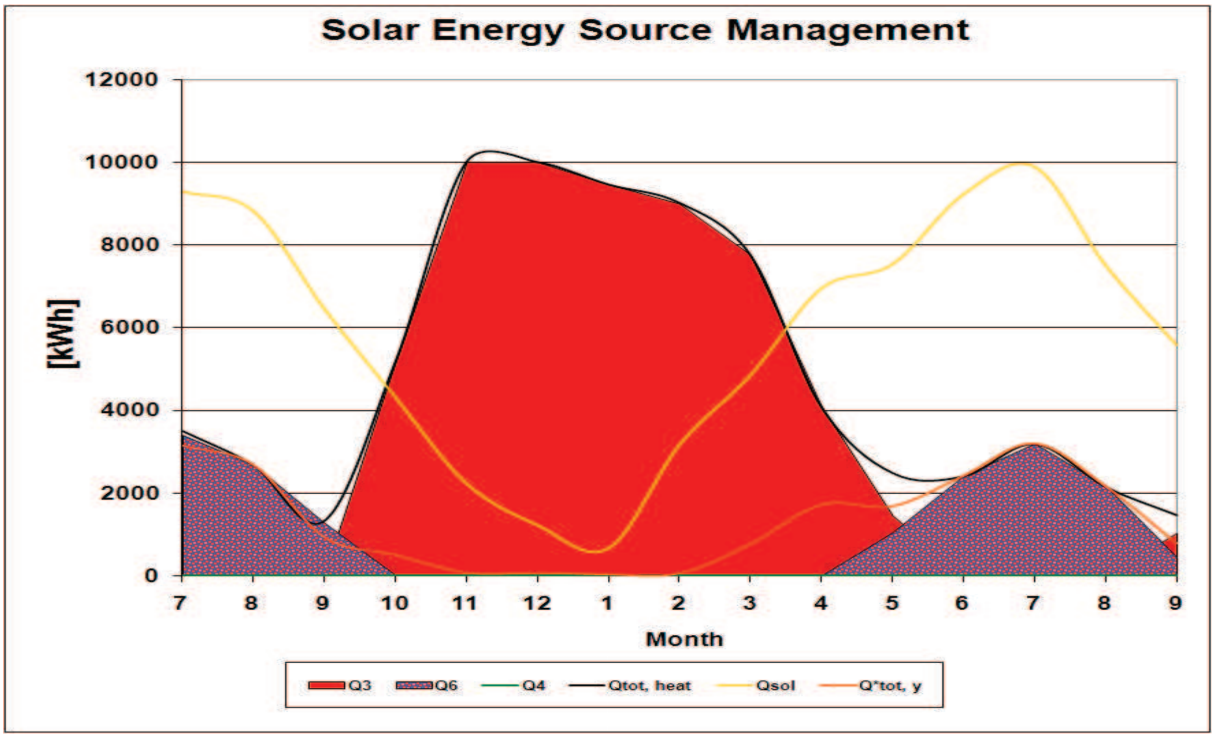


Figure 5: Solar energy management over the whole measuring period July 2009 to September 2010

In the first cooling season the primary energy ratios (PER) of the SHC plant are lower than the defined reference system (Figure 6). As a result of these PER negative savings in terms of primary energy occurred!

After improvements and changes in May 2010 the fraction of primary energy savings got slightly positive. Higher values are expected for the cooling season 2011 with further improvements. Since operation started in July 2009 savings of 14% could be achieved. If the first (negative) cooling period is excluded the fraction of primary energy savings is greater 20%. Taking into account the 100% solar cooling plant with no conventional backup this value has to be improved.

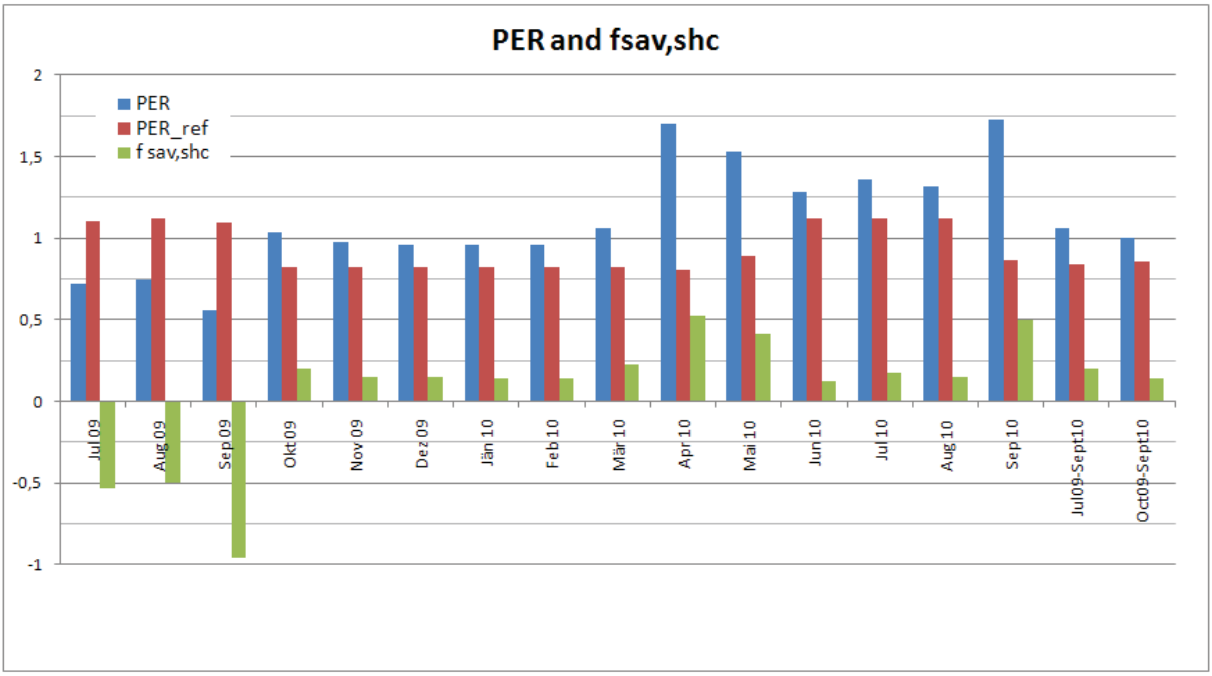


Figure 6: Primary energy ratios and fraction of primary energy saving over the whole measuring period July 2009 to September 2010

Having a look on the general energy consumption of all devices (Figure 7) the average fraction is about

36% for cooling tower including internal pump and ventilator

28% for recooling pump

18 % for hot water pump and ACM

9% for solar pumps

5% for chilled water pump

4% for distribution pump

The standby consumption of E11 and E14/15 leads to a lower average electrical COPs. If the solar system is working but without any cooling demand the electric consumption of E11/E14 is even higher than the solar pump consumption. In case of August 2010 the difference between average with standby and without is 0.1. This difference gets blatant in months with low cooling demand like September 2010. The electrical COP without standby would be 3.2 including the total electrical consumption but only a COP_{el} of 2.4 was achieved!

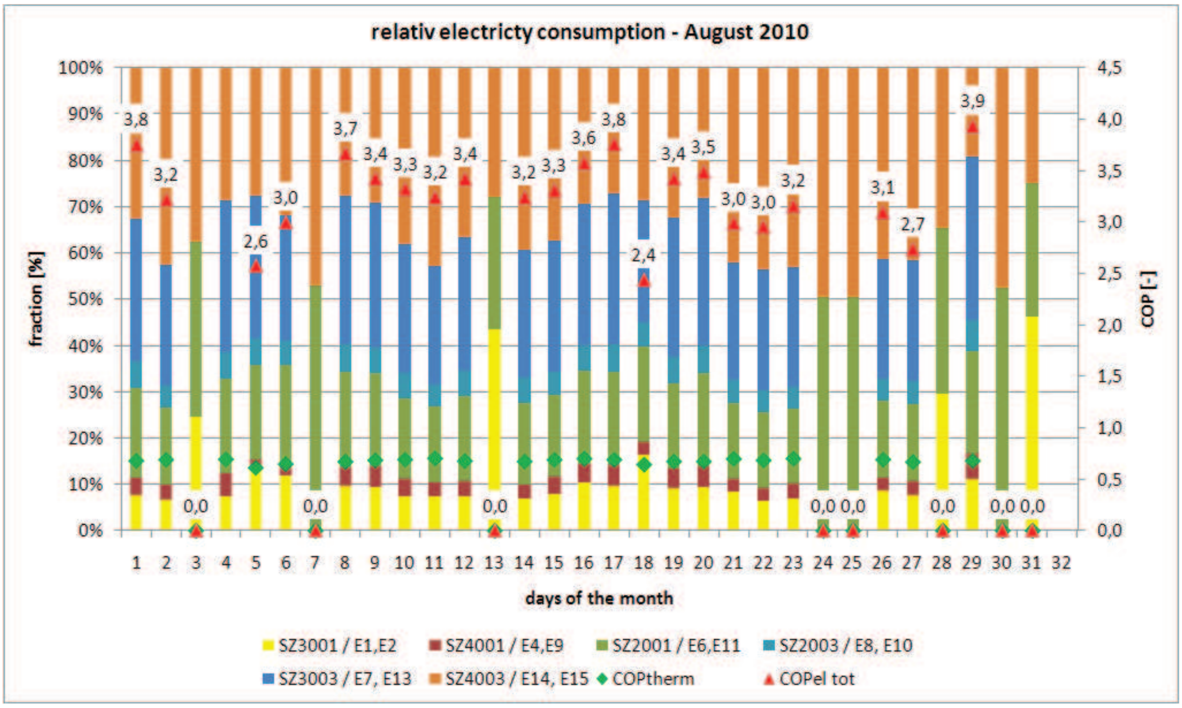


Figure 7: Relative electricity consumption of all devices in August 2010 after first improvements in May 2010

Figure 8 shows the annual load duration curve for cooling in summer 2010. Overall 500 hours of operation can be observed. The absorption chiller is running approximately 100 hours at full load conditions. The remaining period the chiller is working at part load conditions.

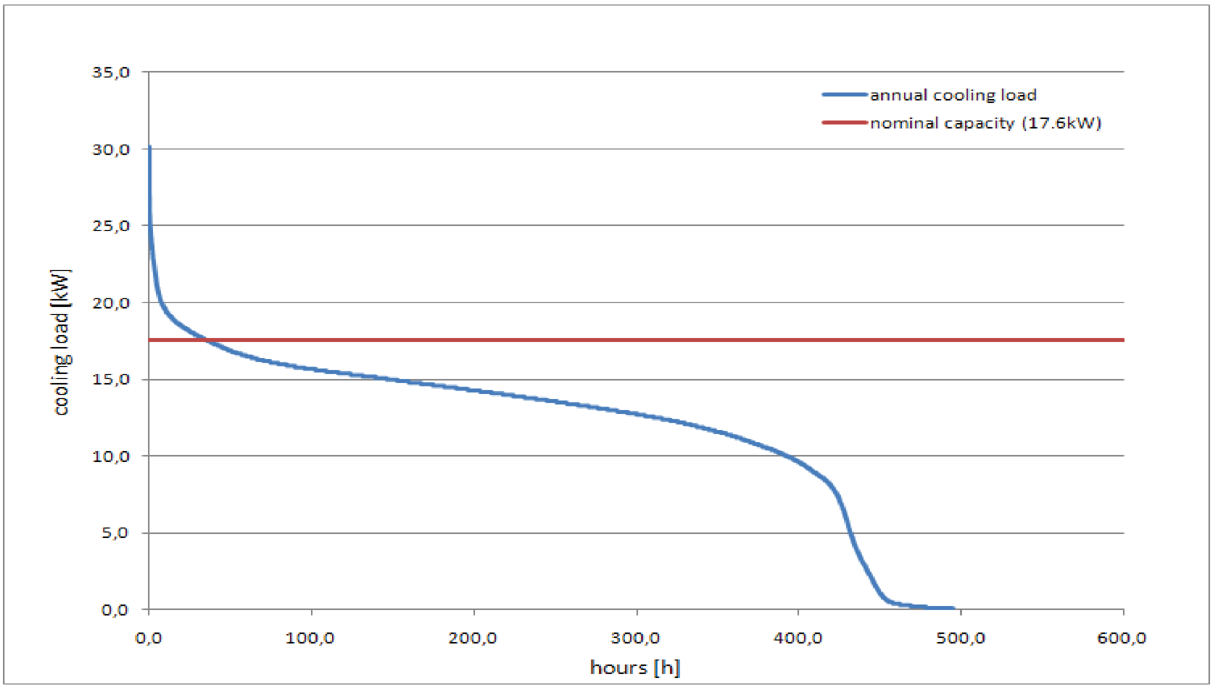


Figure 8: Cooling load curve for 2010

4.2 Analysis of Typical Days

4.2.1 Cooling period 2009

The 25th of August 2009 is an average sunny summer day in 2009. The cooling power, driving heat and solar yield are drafted together with the linked respective temperatures.

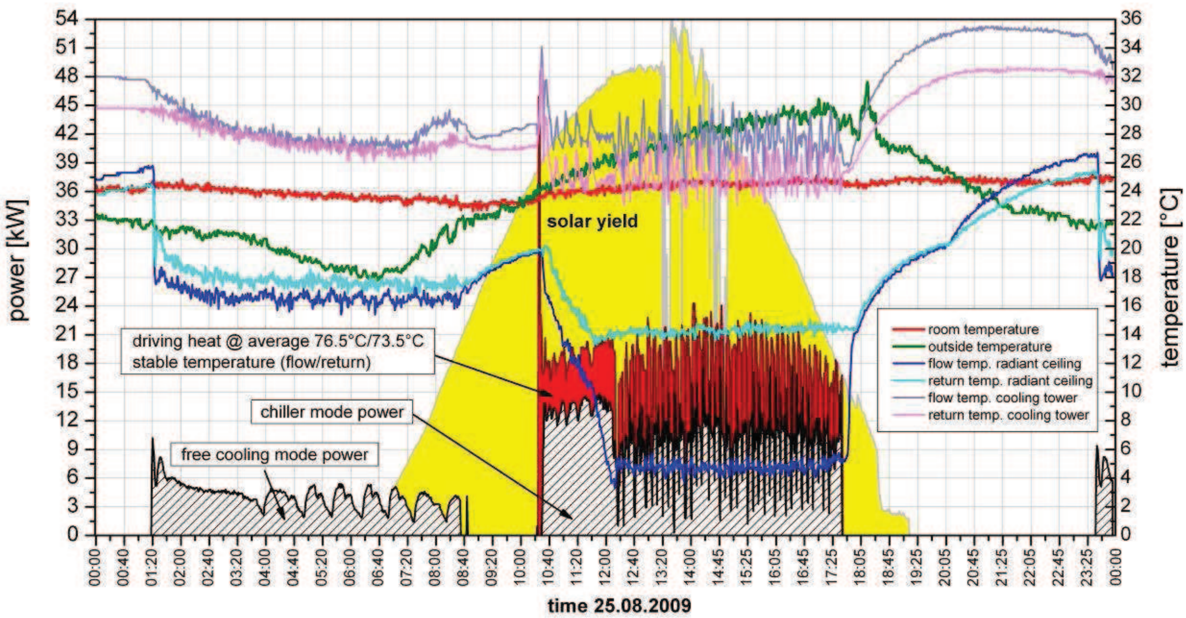


Figure 9: Power characteristics of the CoolCabin on Aug. 25, 2009

At 10:16 a.m. the WFC-SC5 chiller is switched on and is running nearly seven and a half hours without break. The average cooling power alternates between 15 and 10 kW with a matching driving heat power between 20 and 15 kW. Some strong swinging conditions can be observed in the last two-thirds of the cooling period. Referring average temperatures for the driving heat are showed in Figure 9. The heat flow temperature remains in steady conditions throughout the cooling period. It can be observed that the solar yield fits quite well with the delivered cooling load. The recooling water loop coming from the cooling tower is regulated with the valve V6 to control the output cooling power. After two hours of declining temperatures at the radiant ceiling - steady supply and return temperatures of 5°C/14°C are reached. Bad thermal COPs and fluctuating power conditions in the machine are the consequence of the low temperature. These fluctuations can be seen in the recooling loop temperatures, the heat medium power and the cooling power. Following this daily analysis it appears that the plant works how it was designed, but some design decisions as well as some control adjustments are irreproducible.

Out of the manufacturing information of the Yasaki chiller WFC-SC5 the thermal COP reaches a value of 0.65 under standard conditions. In the beginning of this day the thermal COP reached stable values around 0.7. Afterwards the machine shows alternating COPs. In average a thermal COP of 0.616 was reached in the whole period. Considering measurement uncertainties and the lower heat medium inlet temperature mentioned before the thermal COP is feasible.

The chiller was running very reliable throughout the whole summer 2009. It also brought the expected power level and reached a feasible thermal COP. In the following section a closer look will be set to the free cooling mode in order to check the practical operation behavior and performance.

4.2.2 Free cooling

The CoolCabin is equipped with a special hydraulic design in order to run on a free cooling mode. This feature is included to ensure the cooling of the office rooms also on days without sunshine. Therefore the chiller is bypassed so that the cooling tower is able to run on wet mode and can serve the cooling load directly. The mode is activated if the outside temperature is falling below 21.5℃ and the office still has a cooling demand.

Similar to the absorption chiller above a day in August was picked out in order to evaluate the operation behavior of the free cooling mode. In Figure 10 the free cooling power, the solar yield and four temperatures are drafted in a diagram. To be able to compare the solar yield to a standard sunny summer day the solar radiation of the August 25, 2009 is added.

It can be seen that the free cooling mode started three times that day, always switching on when the outside temperature falls under 21.5℃ and stopping when the outside temperature exceeds the 22℃ level. The room temperature stays quite stable throughout the day showing a slight downtrend in a bandwidth between 25℃-23℃. Furthermore the flow and return temperature to the radiant ceiling of the office shows values between 18-16℃/19-17℃ (flow/return). The realized temperature difference between flow and return temperature is very small. In total 71 kWh cold were delivered to the office rooms on that day.

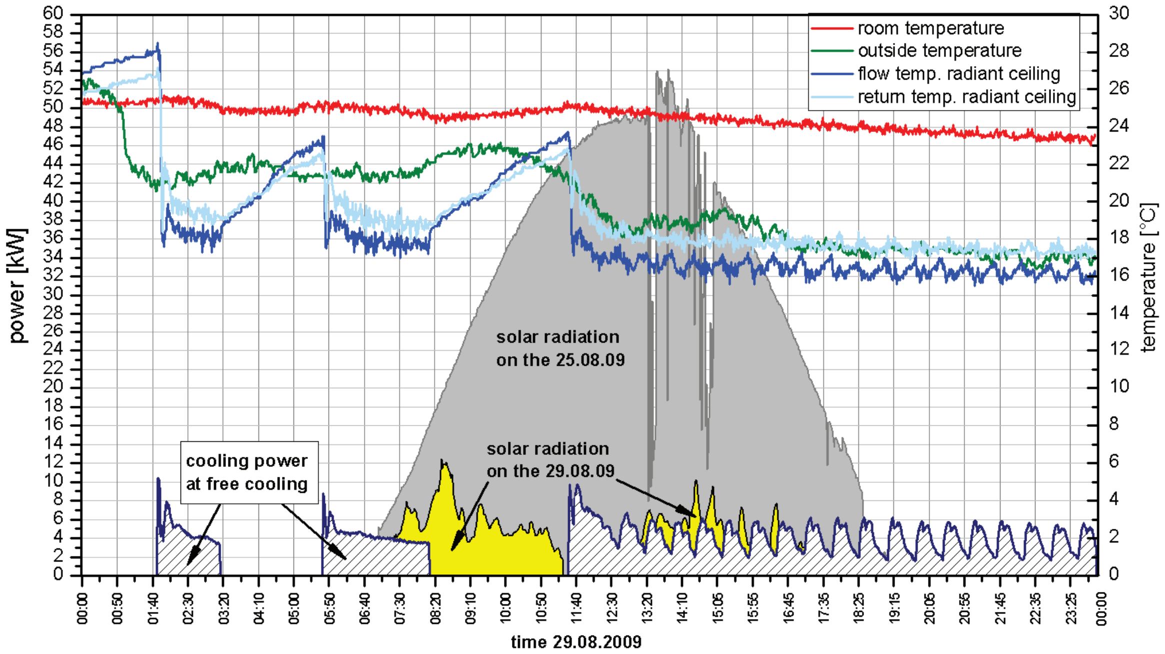


Figure 10: Free cooling operation performance of the CoolCabin on the Aug. 29, 2009

The cooling power revealed a saw tooth tread design which can also be seen in the developing of the radiant ceiling flow temperature. In Figure 11 the practical operation experience of the August 23, 2009 is marked in a diagram. The room temperature stays stable between 25℃ and 23℃. In night times the free cooling mode is activated and running on a poor power level. On the day the chiller is operating at an average power level of 10 kW. The room temperature in this time declines only slightly.

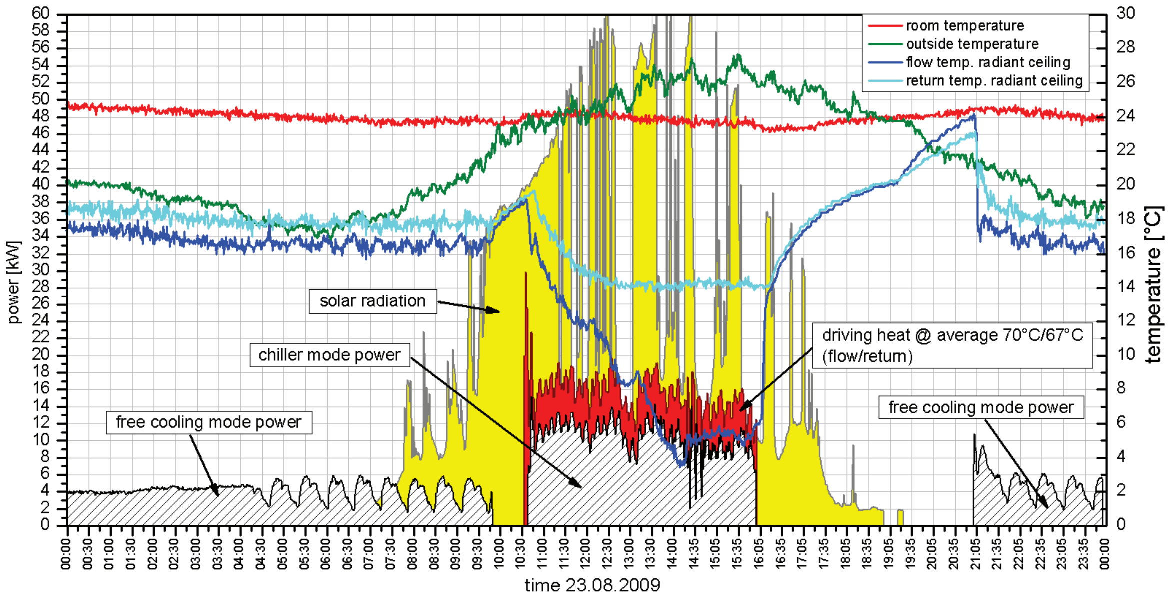
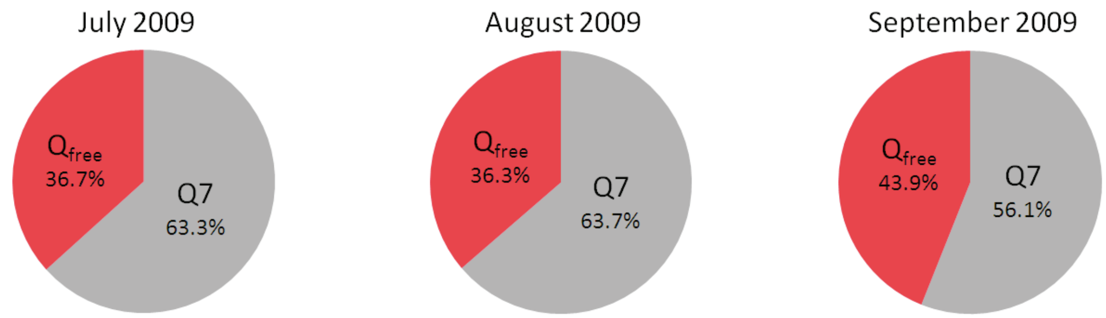


Figure 11: Operation performance of the CoolCabin on the Aug. 23, 2009

The fraction of free cooling in the months July, August and September (2009) is shown in Figure 12 including detailed numbers in the tables below.



Q10a	Q7	Q_{free}
2998.0 kWh	1898,5 kWh	1099,5 kWh
100%	63.33%	36.67%

Q10a	Q7	Q_{free}
2673.0 kWh	1702.0 kWh	971.0 kWh
100%	63.67%	36.33%

Q10a	Q7	Q_{free}
1195.0 kWh	670.0 kWh	525.0 kWh
100%	56.07%	43.93%

Figure 12: Free cooling share of the total delivered cooling load for July, August and September 2009

Approximately one third of the total cooling load was done by free cooling. The free cooling mode is especially running in night times and on rainy days.

4.2.3 Improvements / changes 2009-2010

After the monitoring period 2009 some changes and improvements have been performed.

The main changes were:

- beginning of May10: Maintenance of the cooling tower
- end of May10:
    - supply temperature of chilled ceiling from 17°C to 15°C
    - clearance temperature (upper storage temp.) for starting the ACM from 75°C to 88°C

- end of May10:
    - o clearance temperature (upper storage temp.) for starting the ACM from 88℃ to 80℃
    - o Change of ventilation control strategy on/off to speed control

The effects of these changes are shown in Figure 13.

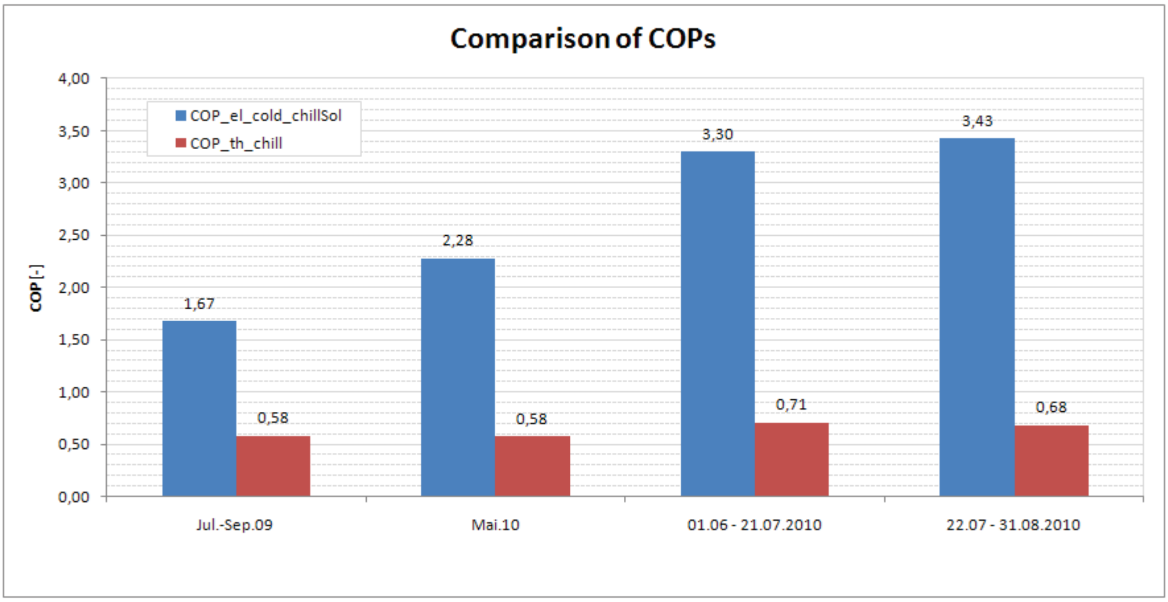


Figure 13: Comparison of thermal and electrical COPs 2009 and 2010

Compared to the cooling period 2009 (including free cooling and thermal driven chiller) the electrical COP was rising 36% up to value of 2.3 while the thermal COP was on the same level. After changes in the control parameters another increase of 44% could be achieved. Finally, including the change of the control strategy of the cooling tower fan, the electrical COP is at about 3.4. Compared to 2009 a plus of 105% in the electrical COP was obtained. The thermal COP mainly affected by the reccoling, regeneration and chilled water temperature level was enhanced by 17%.

Main reasons for the better performance are higher loads due to weather conditions and changes in the set temperatures (supply temp.) and as a result less part load condition!

4.2.4 Cooling period 2010

The daily performance of a typical sunny day is shown in Figure 14. The chiller starts its operation at 11:30 a.m. after the storage temperature reaches 80℃. The driving temperature is slightly increasing during operation and the average is at about 77℃. The chilled water temperature decreases from 20℃ to 8° C while the recooling temperature is controlled to 29℃. Within these conditions a decre asing power for driving heat and cold production is visible.

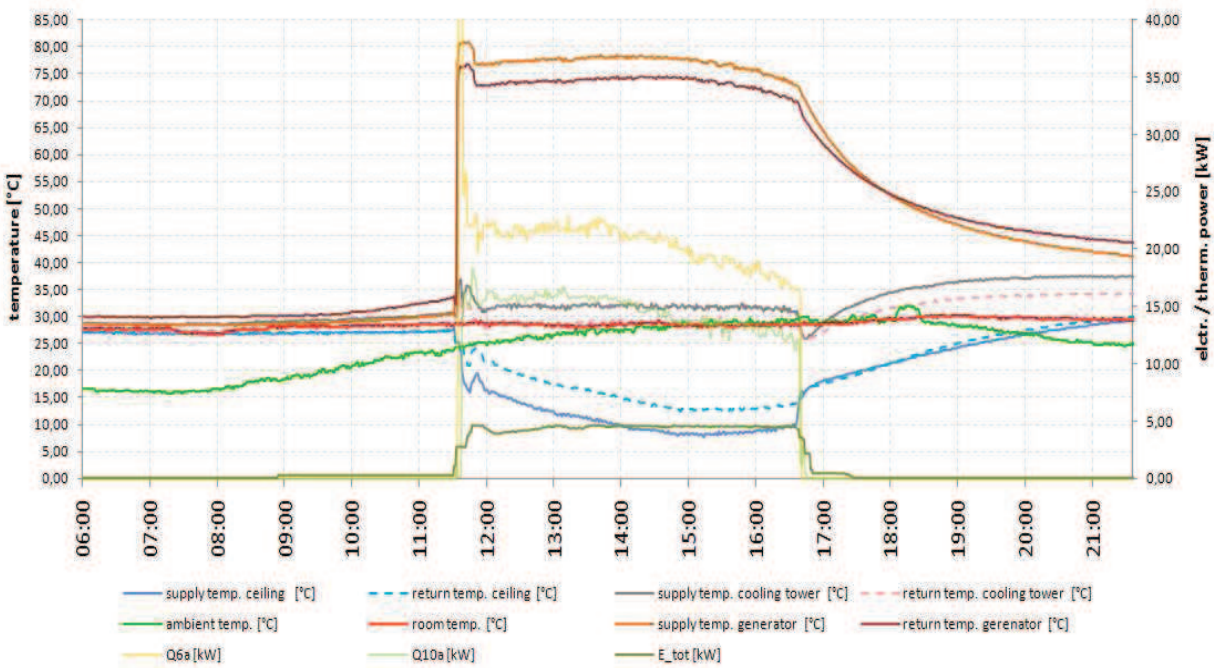


Figure 14: Typical sunny day after improvements in cooling season 2010

An average generation power of 20.7 kW and an average cooling power of 14 kW leads to an average thermal COP of 0.68. The electricity consumption is almost constant during operation at a level of 4.5 kW. Due to the part load conditions the average electrical COP amounts to 3.1. The average COP_{el} of August 2010 is 3.4; mainly due to the same reason.

The under dimensioning of the chilled ceilings can be seen by having a look on the room temperature. During operation of the cooling plant there is hardly any effect on the room temperature. To put all in a nutshell this plant shows a good performance but points out the importance of correct system integration and long term monitoring. Nevertheless the overall primary energy savings of this plant during the period Oct 09 to Sept 10 are 51% compared to the reference system.

After improvements the alternating behavior of the chiller could be stopped (Figure 9). Consequently the thermal COP was increased. The electricity consumption was slightly decreased but at the same time a increasing of the cooling load was achieved.

5 Experiences / Lessons Learned

The average electrical COP 2010 of about 3.4 is still too low and should be improved. Main focus should be the part load conditions.

- Main components focus: recooling pump, cooling tower
- Control strategy
  - o Lower flow rate (constant or variable) for driving pump
  - o Including the actual load to speed control of main components
- Reactivation of free cooling mode with better control strategy

6 Conclusions

Analysis of the operation stability of the system shows reliable running conditions over the whole summer with only a few days where the absorption chiller did not work due to electrical problems.

The daily operation performance indicates that the radiant ceiling is dimensioned to small for the system or vice versa. If the radiant ceiling is under-dimensioned for the cooling plant, as it is the case at the Solid plant, the whole cooling system is running in part load.

Matching the temperatures between two systems is important. The solar cooling system was designed for running on a 9℃/17℃ nominal flow/ret urn temperature, but the cold distribution (radiant ceiling) requires only 16℃/1 9℃. Consequently this leads to high mixing losses in the radiant ceiling inlet temperature regulating valve. This affects mainly the absorption chiller, because it leads to lower thermal COPs but also to higher system losses and therefore to lower electrical COPs.

The free cooling mode (2009) delivers one third of the total cooling load to the building. The electrical COPs of the free cooling mode are not satisfying. Due to the low heat storage capability of the building the cooling effect is very limited. Maybe there are other reasons that make a further running on the free cooling mode feasible. If the free cooling mode is continued a new time control has to be implemented in the control strategy in order to run the mode only in times when the office is occupied. Free cooling of an empty office with low heat storage capability just raises electricity costs.

Measurement results show that the heat rejection system including the cooling tower and the recooling pump plays a key role in reducing the electrical consumption of the plant. Replacing the existing cooling tower to a smaller and more efficient one could be one solution. With a more efficient cooling tower and more efficient pumps the electrical COP could easily be doubled. Therefore a pure wet cooling tower is preferable.

Expected improvements of these suggestions are higher thermal and electrical COPs as well as improved conditions in the office. In order to simulate other changes of the system in the course of the Solar Cool Monitoring Project simulations will be carried out.

SOLID: "…Sophisticated solar cooling systems need 1 to 3 years of surveillance and optimization after start-up. This ensures optimal operation according to the customer's needs high efficiency and long life time of the system…"

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 3

Monitoring Results of solar cooling plant at Municipality Department 34, Vienna

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 23.05.2011

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Contents

1 Technical description of the plant

The thermally driven adsorption SOL ACS 08 - manufactured by Sortech AG in Halle a.d.S./ Germany - generates cold water with a nominal capacity of 7,5 kW (see Figure 1). The adsorption chiller is a new development and applies innovative coating, compact construction, optimized subsystem including dry heat rejection unit, which has EC-fan technology and an additional fresh water spray mode implemented. In order to provide solar energy to the adsorption chiller 12 flat-plate collectors, with an overall gross area of 32,4 m², are mounted on a garage roof, facing south with an inclination of 40 degree. The hydraulic scheme of the overall system design shown in Figure 2 contains two thermal storages, hot water storage (2000 l) as well as cold water storage (800 l). In summer fan coils unit extract heat from several office rooms in order to control the room temperature. The three pumps around the adsorption chiller (cold water pump, heat rejection pump, hot water pump) are not designed for automatic speed level control; they have three levels of switching, which can be changed only by hand. AIT had direct access to the plant management system for changing set points and analyzed their impact on the system performance.

Figure 1: Technical room MA34 solar cooling plant in Vienna (Source: SOLution)

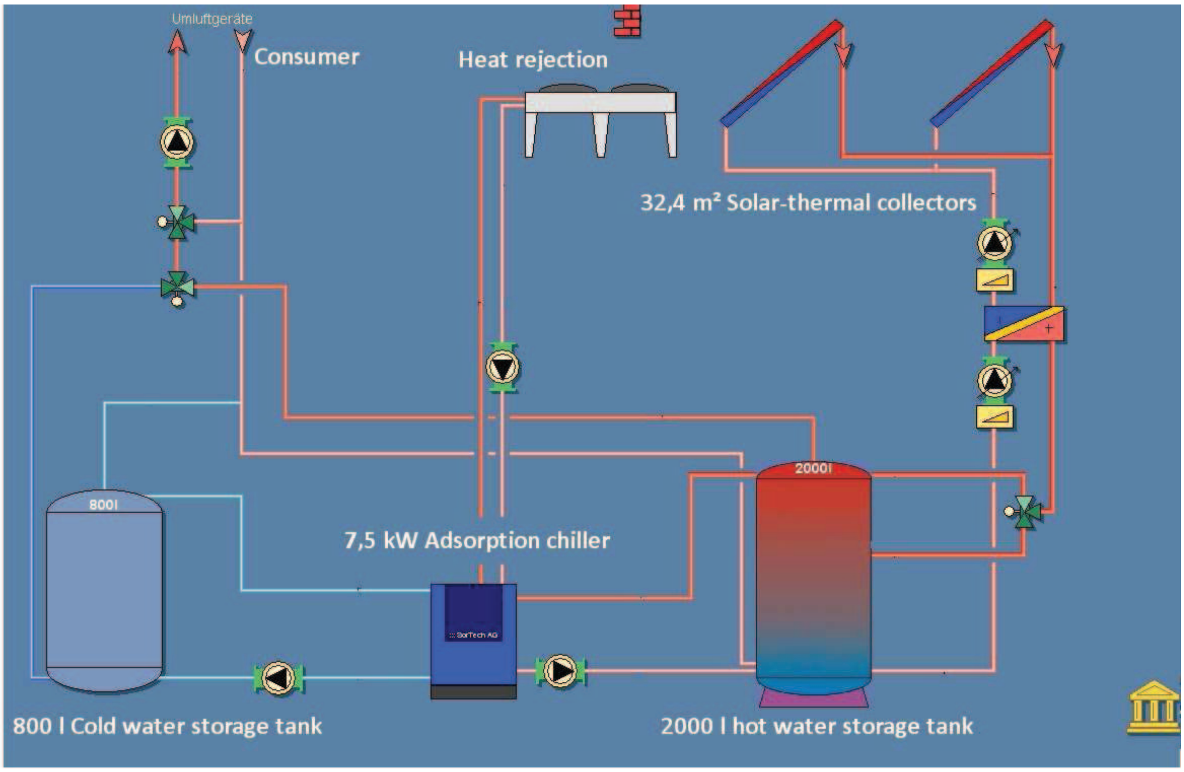

Figure 2: Scheme of MA34 solar cooling plant in Vienna (Source: R&S)

2 Recording of monitoring data

The solar cooling plant was installed in spring 2009 and the system operation was monitored in summer 2009 and 2010. In summer 2009 the monitoring data transfer contained gaps for the duration of some hours to several days, caused by the used monitoring recording software. Therefore, a new monitoring server was installed at AIT for summer 2010 which led to a continuous recording of monitoring data in the second summer.

Approximately 75% of the monitoring data from the summer 2009 were recorded. This data represents all relevant weather conditions for Vienna. As a result, it was possible to make a first evaluation of the energy performance of the solar cooling plant.

Due to the new monitoring server at AIT there weren´t any losses of monitoring data in summer 2010.

3 Experience report summer 2009

The monitoring of the first cooling season showed that the adsorption chiller worked reliable and the desired air-conditioning of the office rooms was given. The room temperature in the offices was kept under 24℃ over the whole summer. There weren`t any error operations in the overall system or in the individual components. The average cooling capacity of the system lay between 1,7 kW and 2,4 kW, which is way below the nominal capacity of 7,5 kW of the adsorption chiller. This part load operation caused by the low cooling capacity on the building side has a negative impact on the thermal Coefficient of Performance (COP_{th}) as well as on the electrical Coefficient of Performance (COP_{el}) of the solar adsorption chiller system. The documentation of the control was incomplete, which made the check of the implemented control strategies in the plant through monitoring evaluation quite difficult.

In July 2009, the cold water pump showed a varying electricity consumption. The comparison between the electricity consumption of the pump and the mass flow in the cold water circle showed that there was an electricity consumption of the pump, even when the mass flow was zero. This led to the conclusion that air was in the cold water pipes. Therefore, the water pipes were de-aired which affected the desired continuous mass flow.

The energy performance of the system is evaluated by two indicators. The Thermal Coefficient of Performance (COP_{th}) gives the ratio between produced cooling energy and used thermal driving energy for a certain period of time (see Equation 1).

Thermal Coefficient of Performance:

$$COP_{th} = \frac{Gains}{Thermal\_Demand} = \frac{Q_{cooling}[kWh]}{Q_{solar\_thermal}[kWh]} \qquad \text{Equation 1}$$

The Electrical Coefficient of Performance gives the ratio between the produced cooling energy and the used electrical consumption for a certain period of time (see Equation 2).

Electrical Coefficient of Performance:

$$COP_{el} = \frac{Gains}{Electricity\_Demand} = \frac{Q_{cooling}[kWh]}{E_{driving,el}[kWh]} \qquad \text{Equation 2}$$

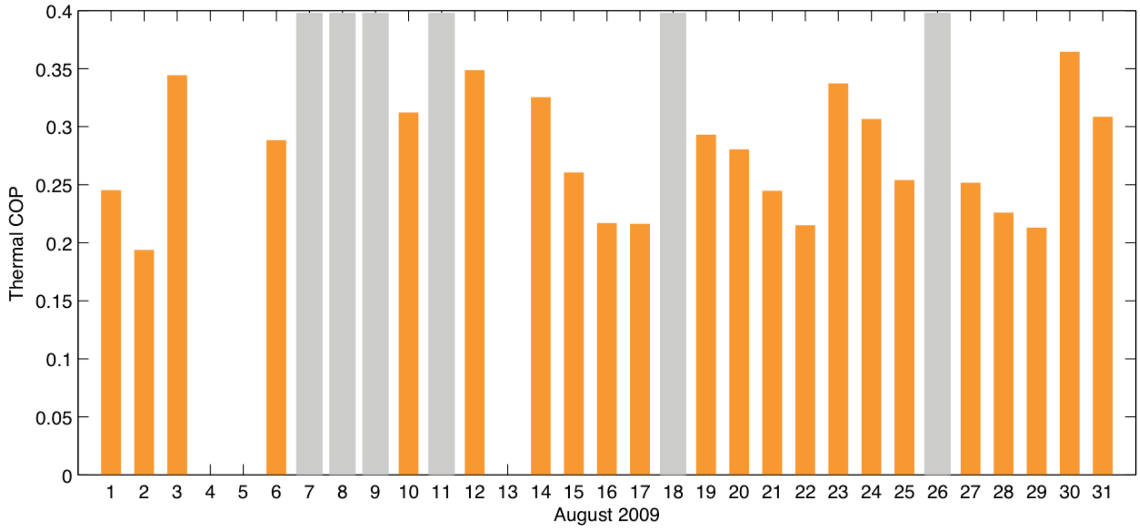

Figure 3: COP_{th} of MA34 solar cooling system in August 2009

Figure 3 shows that the daily COP_{th} for August 2009 are in the range between 0,19 to 0,35. On the days with grey bares the recording of monitoring data was insufficient.

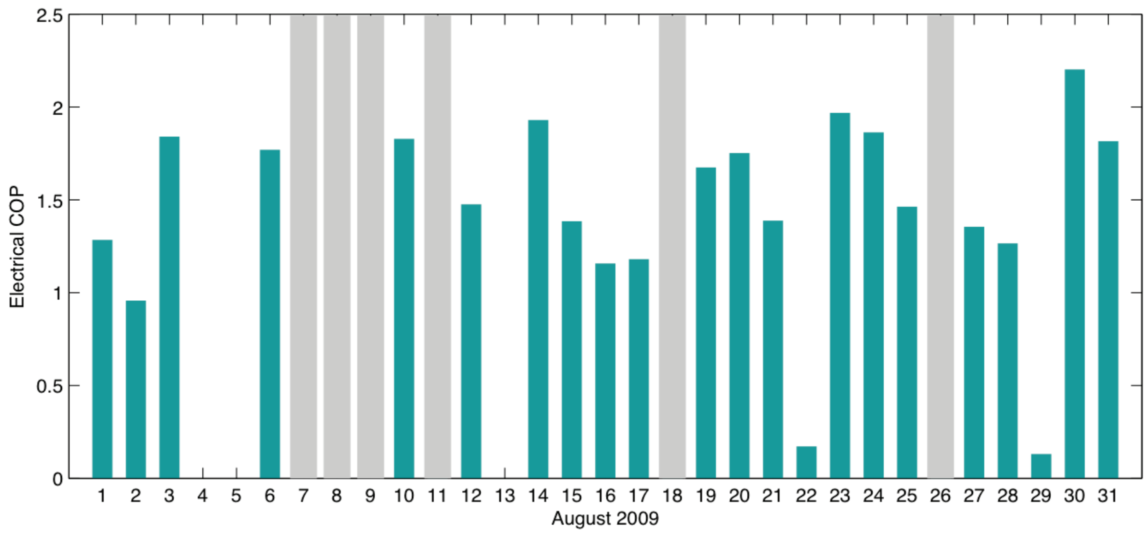

Figure 4: COP_{el} of MA34 solar cooling system in August 2009

Figure 4 shows that the daily COP_{el} are in a range between 0,2 and 2,3 in August 2009. The low COP_{el} are mainly caused by the low cooling capacity needed in the offices. The adsorption chiller operated nearly all the time in a very low part load, but the three pumps around the adsorption chiller and the heat rejection were operating in full load.

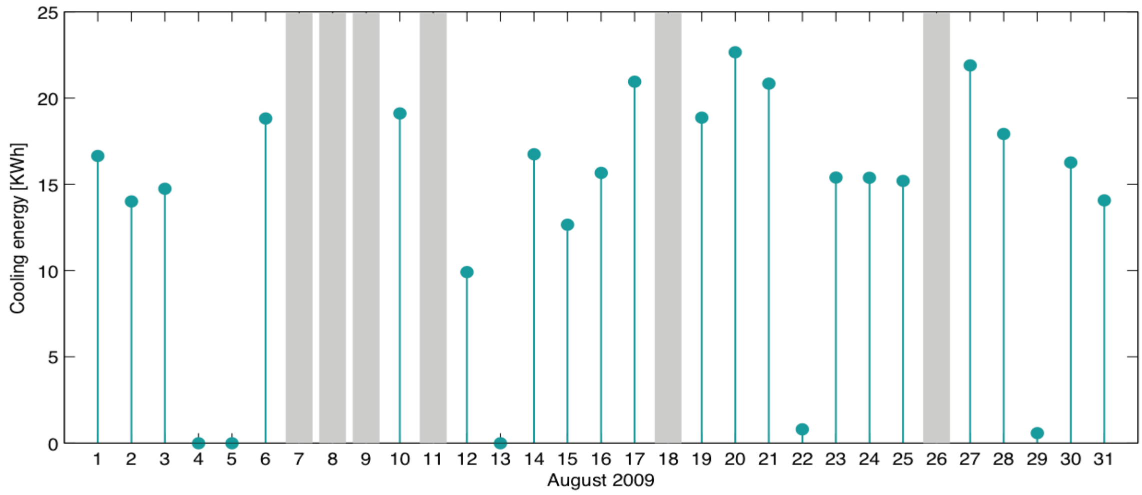

Figure 5: Delivered cooling energy by MA34 solar cooling system in August 2009

Starting from the nominal cooling capacity of the SorTech chiller, the maximum daily cooling energy can be determined. A 7.5 kW chiller can yield a maximum of 180 kWh of cooling energy per day, assuming incessant operation. The number of operation hours in a solar cooling system, however, can be limited by the heat source. An six-hour long operation would ideally still yield 45 kWh of cooling energy. Figure 5 shows daily cooling energies between 0 and 23 kWh.

The values vary according to the control strategy, which indirectly reacts to the outside temperature and solar radiation. In addition, the consumer behaviour must be taken into account. Constant low values of the cooling capacity imply that the chiller is over-sized for the system.

Apart from the energy output of the system, it is also very important to look at temperature levels going in and out the adsorption chiller, as the hot side and the re-cooling temperature levels can drastically influence the performance. Figure 6 shows the inlet and outlet temperatures on all three temperature levels at the chiller for a typical August day. The monitored outside temperature and solar irradiation can be found in the lower graph. The outside temperature peaked at 32℃ and the solar ir radiation reached 917 W/m2. Overall, it was a bright sunny day with a cloudy passage in the afternoon.

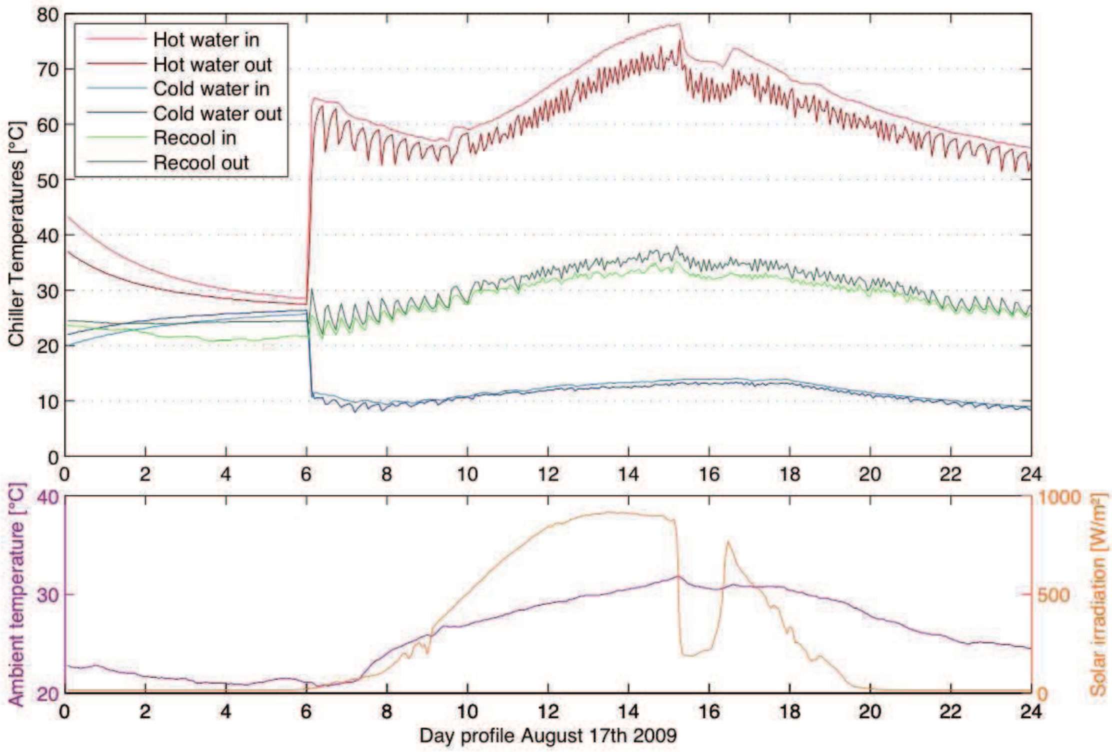

Figure 6: Temperature profiles adsorption chiller

The chiller turned on at 6 a.m. and was in operation through the rest of the day. Hot water, heated by the solar collectors, was driving the chiller with 57-78℃, depending on the time of the day (top curve). It is noticeable, that the loss in radiation by a cloud decreased the hot tank temperature with almost no time delay. The temperature difference between the two red curves stands for the energy intake of the adsorption chiller. Oscillations in the dark red curve reflect the timing of the chiller, which occurs when it switches between the adsorption chambers. The timing can still be seen in the re-cooling temperatures (green curves). The top green line is the temperature of the water going from the chiller to the cooling tower. After being cooled down, the water returns to the chiller (light green curve). The temperature spread generated by the cooling tower did not seem to be affected by the high outside temperature, as the ΔT was not decreasing. Chilled water, represented by the blue curves, only showed a small temperature spread of 1-2 K. However, the overall temperature level of the cold water went down towards the end of the day. This was due to a stop in the cooling demand from the offices.

This characteristic diagram of temperature levels at the MA34 adsorption chiller shows, that it was not a problem for the chiller to meet the cooling demand of the offices. Temperature levels, at which the chiller is driven, are within the recommended range of the manufacturer. Nevertheless, the ΔT for the cold water remains below expectations, which is expressed in the actual cooling capacity and also affects both the thermal and electrical COP.

ΔT_cold is dependent on both, the hot side temperature and the heat rejection temperature entering the chiller according to thermodynamic principles and the efficiency of the chiller. In order to maximize ΔT_cold, it is necessary to drive the chiller with the highest desorption temperature possible. This can be done with changing volume flows and adapting the control strategy.

It is also essential to improving ΔT_recool to create a higher ΔT_cold. The more energy the cooling tower can dissipate, the higher ΔT_cold the chiller is able to reach. Modifying volume flows and the controls could be a way to make some improvements.

Despite all attempts to increase ΔT_cold, it is also necessary to look at the actual temperature level of the cold water. The SorTech adsorption chiller can cool down to about 5℃, however, the lower the temperature level, the less efficient the chiller works. It can be seen in the above figure that the cold side temperature level drops below 10℃ at night. On other monitored days, going down of temperatures to 6℃ as not uncommon, and only then the chiller turned off. Such low temperatures are certainly not necessary for cooling offices with fan coils. Flow temperatures from 15-18℃ are sufficient and probably even more comfortable. Also, heat losses in the cold storage tank could be reduced. Therefore it is highly recommended to change the set-point for the chiller, to turn off at a higher temperature.

All previously mentioned adaptations aim at increasing ΔT_cold or the efficiency of the chiller, which results in both, higher thermal and electrical COPs.

It is, however, necessary to decide whether to optimize the solar cooling system to a high electrical COP or to a high thermal COP. Improvements in the COP_{el} can be attained by applying earlier mentioned modifications as well as by cutting down the electrical consumption.

Figure 7 shows typical COP_{th} and characteristic curves of the here analysed adsorption chiller (Type ACS08).

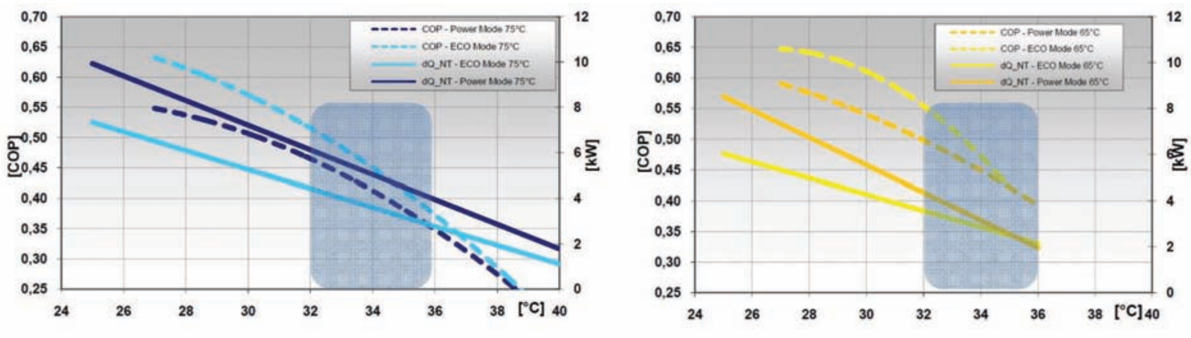

Figure 7: COP_{th} und characteristic curves ACS08: cooled ceiling (T_{in} = 18 °C) (Source: SorTech AG)

Figure 8 shows the share of all devices in electricity consumption. It can be seen that heat rejection and the associated pump make up almost two thirds of the entire electricity consumption. Hot and cold water pumps together with the control device account for almost all the rest. The pumps in the solar cycle I and II are the only energy saving pumps and they are also the only speed controlled pumps in the solar cooling system. A lot of electricity could have been saved by equipping the entire system with energy saving pumps. However, some additional improvements may be made by operating the pumps connected to the chiller at variable speeds instead of steady flow.

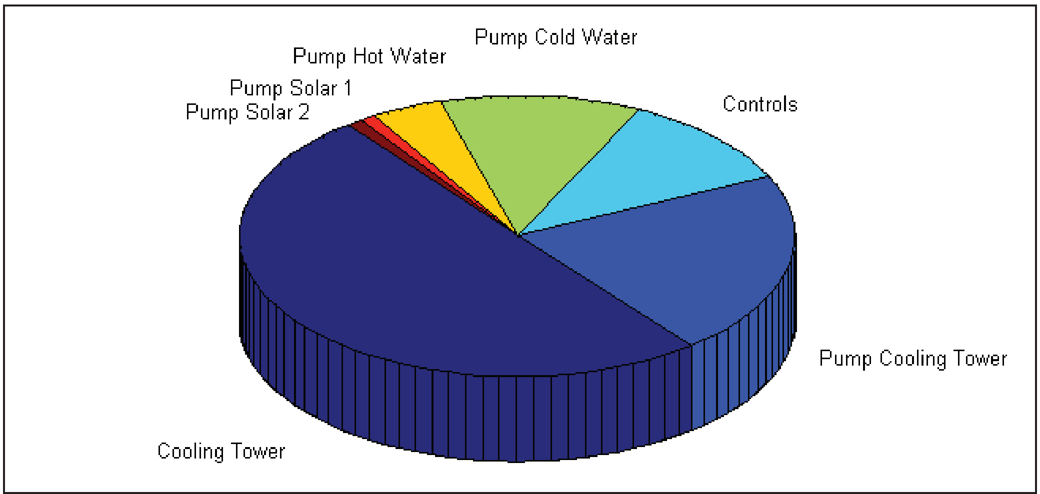

Figure 8: Electricity consumption of MA34 solar cooling system in August 2009

The control system also exhibited a surprisingly high energy use. Monitoring showed a constant consumption of 40 W, which is definitely higher than many other control systems available in the market. The cooling tower is by far the biggest consumer of all electrical components.

Interestingly enough, monitorings uncovered a random consumption of the cooling tower of constant 130 W on top of the regular consumption during operation times. There is no indication of a stand-by mode and speculations about an incorrectly wired anti-freeze element turned out to be wrong. So far, it has not been clarified what really causes the extra electricity consumption.

4 Experience report summer 2010

The following section contains monitoring results of the MA34 solar cooling system for August and September 2010 with a focus on the variable speed control mode of the heat rejection fans.

The monitoring evaluation of August showed that the heat rejection operated permanently, even when the rest of the plant wasn´t in operation (see Figure 9). That was caused by an error set at the adsorption chiller control panel, which was “manual” instead of “automatic”. This error was corrected at the 14th of August 2010; the resulting decrease of electricity demand is shown in Figure 9. Furthermore, the COP_{el} increases significantly from that date on (see Figure 10) although the heat rejection is operating with maximum capacity (ca. 650 W).

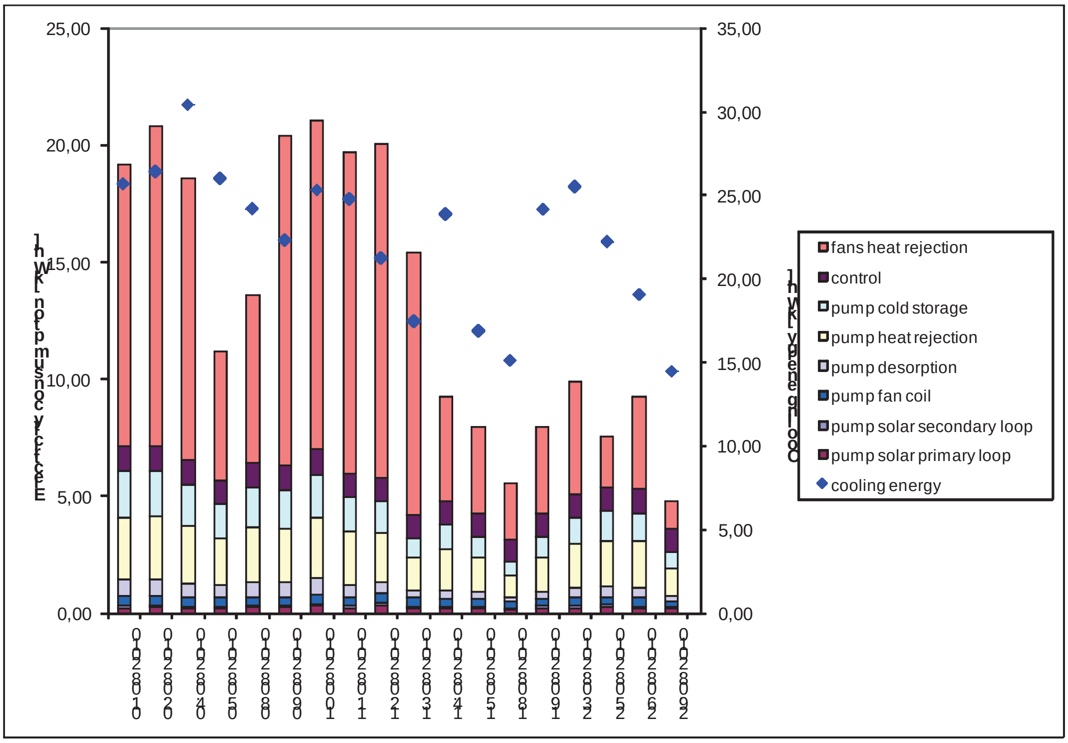

Figure 9: Electricity consumption of MA34 solar cooling system in August 2010

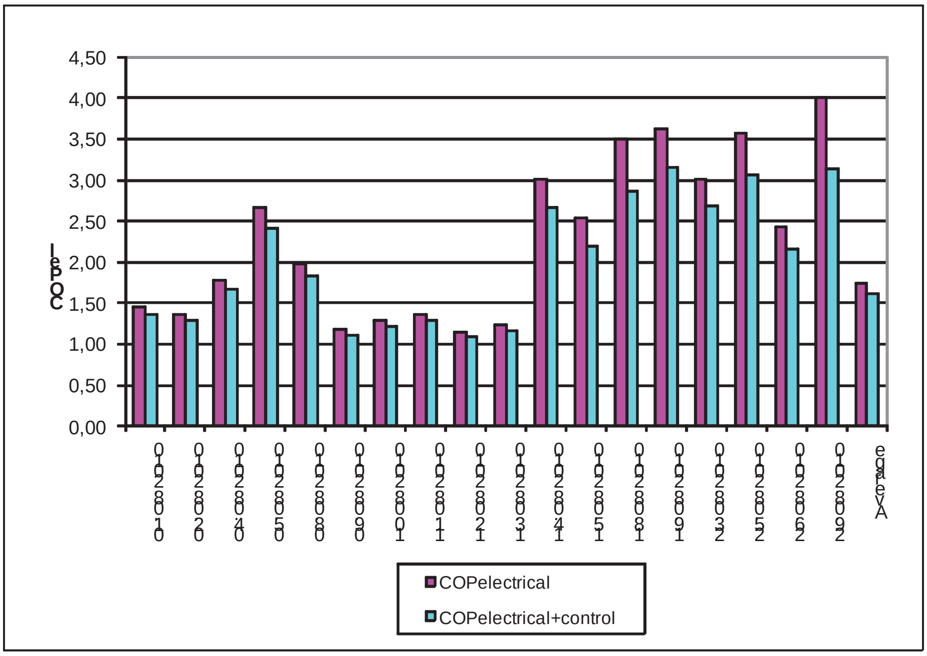

Figure 10: COP_{el} of MA34 solar cooling system in August 2010

On the 25th of August 2010 the variable speed control of the heat rejection fans started to operate the first time for a longer time period. The heat rejection fans - which cause the main part of the electricity consumption – operate between 670 W and 65 W driving capacity (see Figure 11).

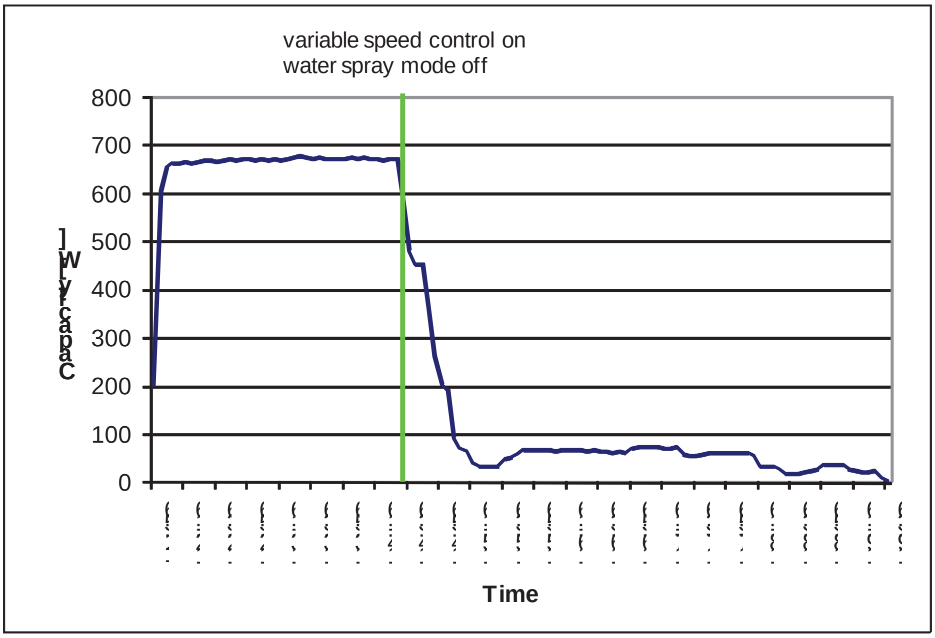

Figure 11: Capacity heat rejection at 25.08.2010

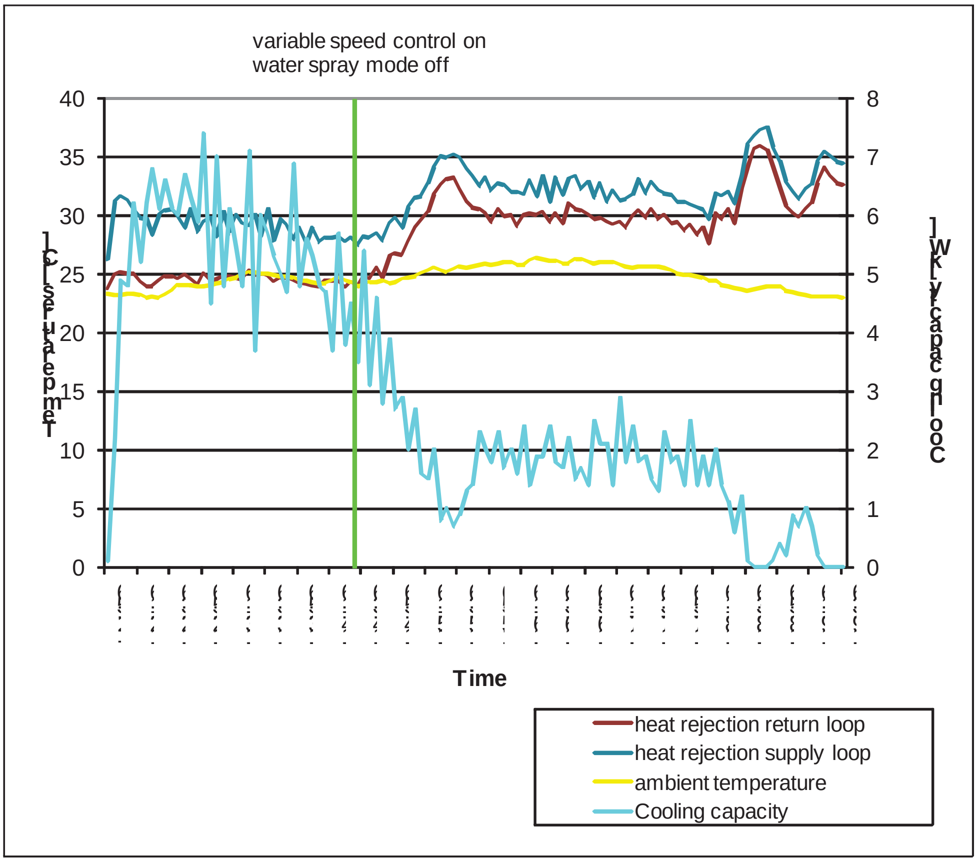

Figure 12: Impact of variable speed control mode of the heat rejection fans on cooling capacity at 25.08.2010

As a result, the heat rejection temperatures increased which affected the delivered cooling capacity (see Figure 12). The cooling capacity decreased, due to the higher heat rejection temperatures, significantly (see Figure 13).

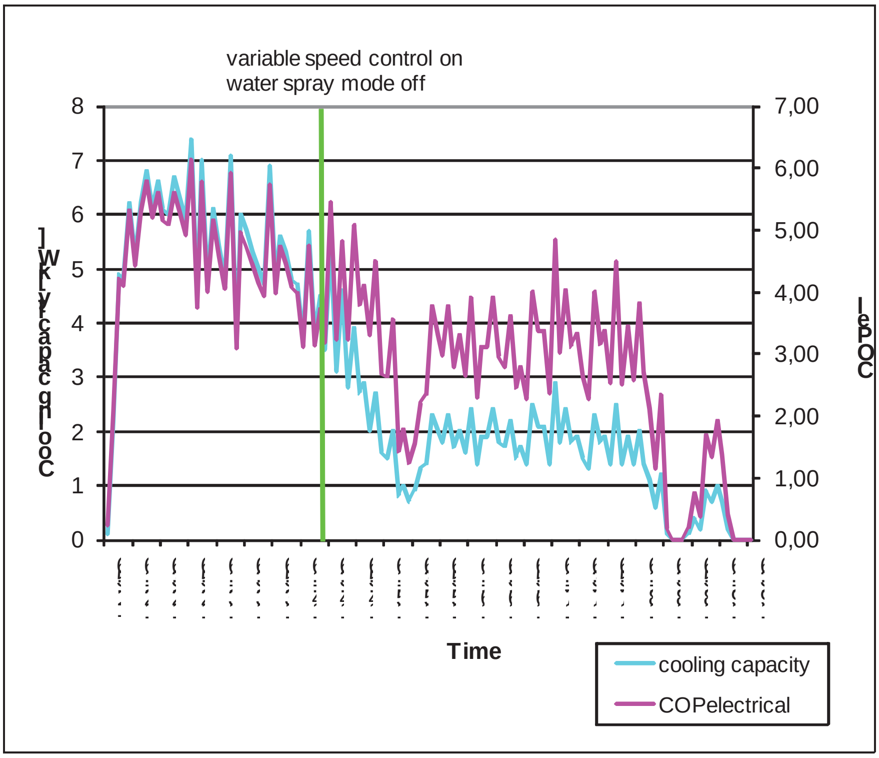

Figure 13: COP_{el} and cooling capacity of MA34 solar cooling system at 25.08.2010

The average COP_{el} without variable speed control of the heat rejection fans is 4,55 at the 25th of August 2010 (see Figure 13). In the time period, when the variable speed control of the heat rejection fans is in operation, the average COP_{el} decreases to 2,78. The average cooling capacity of the adsorption chiller in the time period without variable speed control of the heat rejection fans amounts to 5,46 kW; with variable speed control of the heat rejection fans this value drops to 2,24 kW. Furthermore, the wet operation mode of the heat rejection turns off by starting the variable speed control mode.

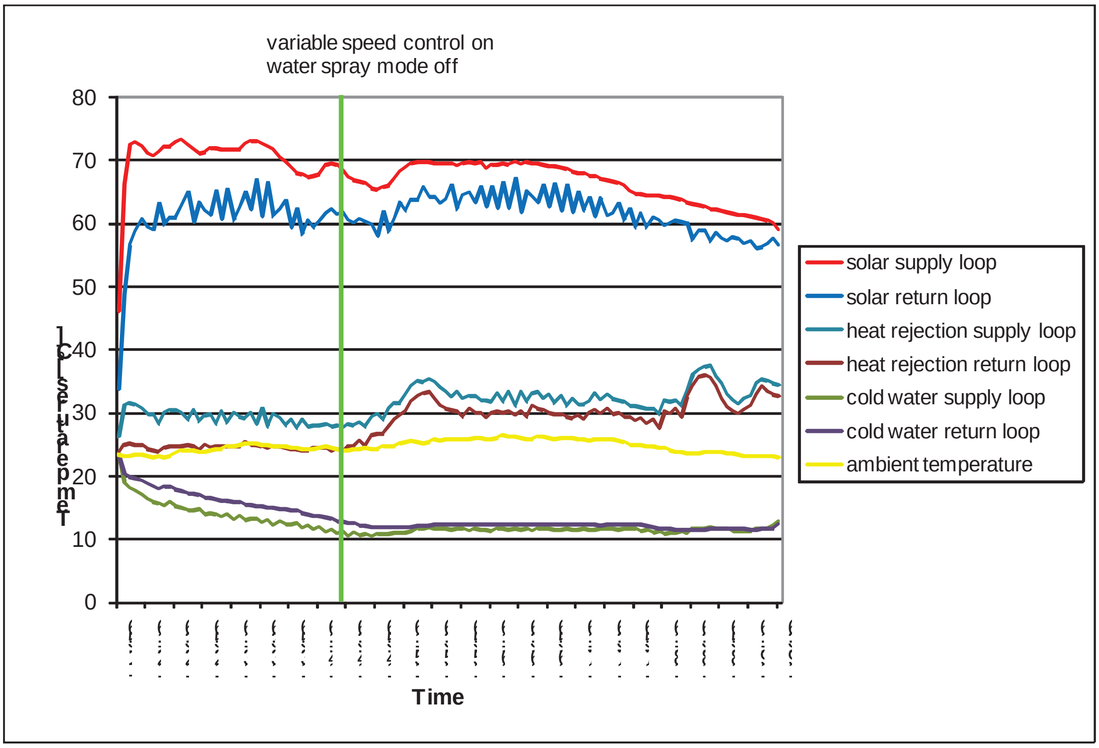

Figure 14: Temperature profiles around adsorption chiller at 25.08.2010

In Figure 14, the temperatures around the adsorption chiller on the 25th of August 2010 are shown. It is clearly identifiable, that while the heat rejection temperatures are rising, nearly any temperature difference between supply and return cold water side happens during the variable speed control mode of the heat rejection fans. Therefore, there wasn´t nearly any cooling demand in the offices and the adsorption chiller shouldn´t have been working at all. The control of the absorption chiller is only taking a certain cooling set point temperature on the cold supply water side into account (here 12 °C). It is therefore the duty of the system controller to turn off the adsorption chiller, when an unreasonable operation like this happens, but that was not adapted by installing the variable speed control mode of the heat rejection fans in July 2010.

To avoid the operation, as it happened on the 25th of August 2010, the cooling set point temperature was changed to 6 °C until the adaptatio n of the system controller happens (see Figure 15).

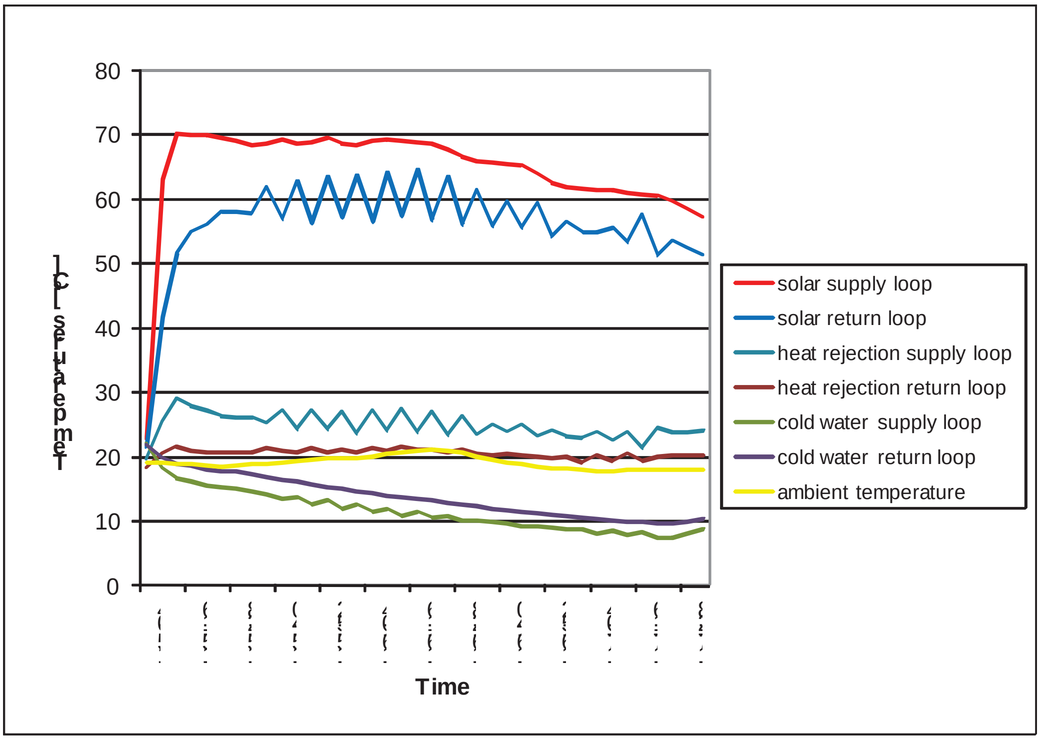

Figure 15: Temperature profiles around adsorption chiller at 04.09.2010

5 Summary and Conclusions

Within the IEA SHC task 38 for the solar adsorption cooling plant of the Viennese Municipality Department 34 (MA34) following investigations were accomplished:

- COP_{el} and COP_{th} of the solar adsorption cooling plant
- Operation of hybrid heat rejection in dry or wet cooling mode
- Effects of variable speed control of the heat rejection fans on the energy performance

Summary of the substantial realizations:

- The heat rejection in this plant causes three quarters of the electricity demand, therefore the selection of a heat rejection device with a high efficiency class (preferably wet cooling tower or at least hybrid heat rejection) is essential to achieve a high COP_{el}.
- Selection of variable speed, energy-efficient pumps are also important to achieve high COP_{el}.
- The monitoring results of the 25th of August 2010 showed that a COP_{el} of 4,55 is already possible with this plant. Therefore, high desorption temperatures are

necessary (> 70 ℃) and a wet cooling mode in the heat rejection. This value is clearly over the maximum daily value of the COP_{el} in August 2009 with 2,3.

- Due to the water temperatures in all three hydraulic circles (cold water, heat rejection and hot water circle) of the adsorption chiller daily COP_{th} were measured in summer 2010 within the range of 0,16 to 0,58. The nominal COP_{th} is about 0,56.

- The plant optimization attempt, by using variable speed control of the heat rejection fans didn´t work due to the lack of coupling between the adsorption chiller control and the system control; on the contrary, the COP_{el} decreased during the variable speed control mode. Generally, the usage of a system controller including the chiller control must be recommended to avoid such errors.

6 Publications

A. Preisler, M. Brychta, F. Dubisch, F. Stift, T. Edlinger, Solar-gestützte DEC-Anlage ENERGYbase, Wien: Evaluierung der Anlage durch Vergleich TRNSYS Simulationen mit Monitoring-Ergebnissen für den Sommer 2009, 20. Symposium Thermische Solarenergie, Bad Staffelstein, 5/2010

A. Preisler, T. Selke, Experience report on two different solar driven air-conditioning systems in Vienna/Austria based on monitoring data of summer 2008/2009, 3rd International Solar Air-Conditioning Conference, Palermo, 10/2009

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 4

Monitoring Results of Résidence du Lac, Maclas (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

By Romain Siré[1] and Daniel Mugnier[1]

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Contents

1 Background

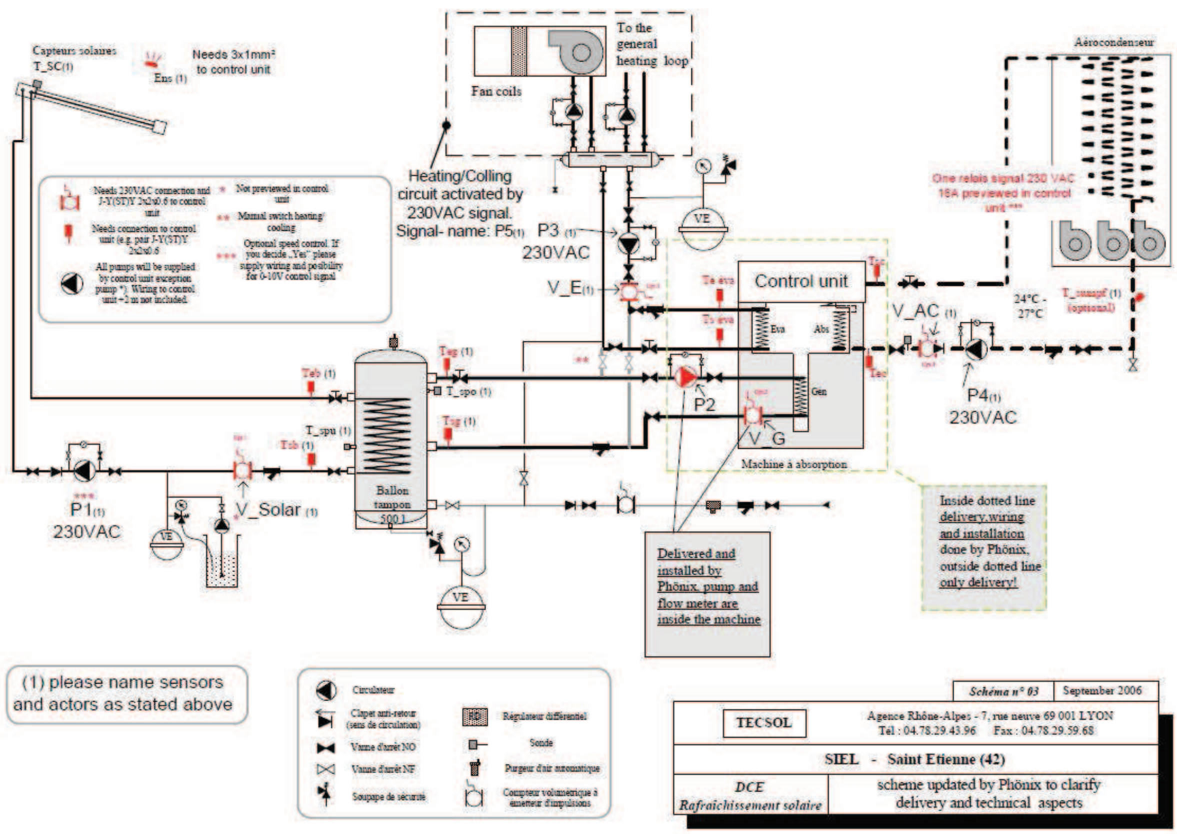
The targeted building welcoming the solar cooling application is the Résidence du Lac, a building dedicated to retired people. This building is located in the small town of Maclas in the Rhône Alpes area, close to Lyon. The town is in altitude, nearly 450 m high.

The building was created in the seventies and is of an average quality level for the energy efficiency. Only one small part of the building is cooled, the leisure space/restaurant which is compulsory since summer 2003 in retired buildings. This area is of 210 m² and includes a veranda oriented in the Southern direction. Efforts were made to increase the solar protection level in the veranda by adding dark thin protection films. Till 2007, the building owner used electric compression chillers (3 monosplits). Two of them were out of order in 2007 and the management took the decision with the help of the SIEL (Syndicat Intercommunal d'Energie de la Loire) to go for a solar cooling system. The owner of the system is the SIEL itself.

2 System Design

The system is based on an absorption chiller of 10 kW coupled with evacuated tube collectors. The system is in configuration of a quasi solar autonomous cooling system because only a small electric compression chiller (split type) is used in case of failure of the solar system. The load of a part of the entire building is based on the following scenario: cooling demand from June to mid September and heating demand from mid October to end of May. The solar system is using fan coils for the cooling and heating modes but thanks to buffer storage, it can be valorized as well in the heating mode through the central heating network of the Résidence du Lac. The heat rejection system is done by a drycooler located in the northern facade of the building.

<u>Installation's scheme:</u>



Hydraulic scheme of the Maclas installation

As it can be seen, the system was designed to limit as much as possible the electricity consumption of the overall installation. Indeed only 4 pumps are used to run the entire system (without distribution). Moreover, a simple system always works better than a complicated one.

Another point which must be enlightened it the choice of a dry cooler for the system's heat rejection. The "Résidence de Lac" is a building for retired peoples. As a consequence legislation is very drastic concerning the legionella bacteria, and a drycooler was the only affordable way to reject efficiently energy without any restrictive regulation.

Building:

- Type of building: Retired people residence
- Location: Maclas, France
- In operation since 2007
- System operated by: SIEL
- Air-conditioned area: 210 m²
- System is used for space cooling and heating (No DHW preparation)

System general properties:

- Technology: closed cycle
- Nominal capacity: 10 kWcold
- Type of closed system: Absorption
- Brand of chiller unit: Sonnenklima
- Chilled water application: Fan coils
- Dehumidification: no
- Heat rejection system: dry

Solar thermal:

- Collector type: Evacuated tube
- Brand of collector: Thermomax Mazdon 20
- Collector area: 24 m² absorber
- Tilt angle, orientation: 30°, 15° west
- Collector fluid: water-glycol
- Typical operation temperature: 75°C driving tempe rature for chiller operation

Configuration:

- Heat storage: 0.5 m3 water
- Cold storage: Buffer water (80 liters)
- The hot and cold backups are not included into the solar project, so they are not monitored. Indeed, the building is an existing building, so a conventional heater and air conditioning system were already used.

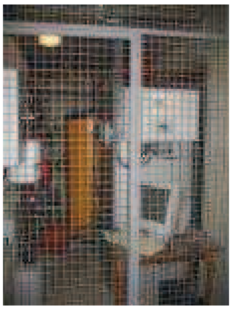
Photos:

Building

Cooled space

Solar collector field

Dry cooler

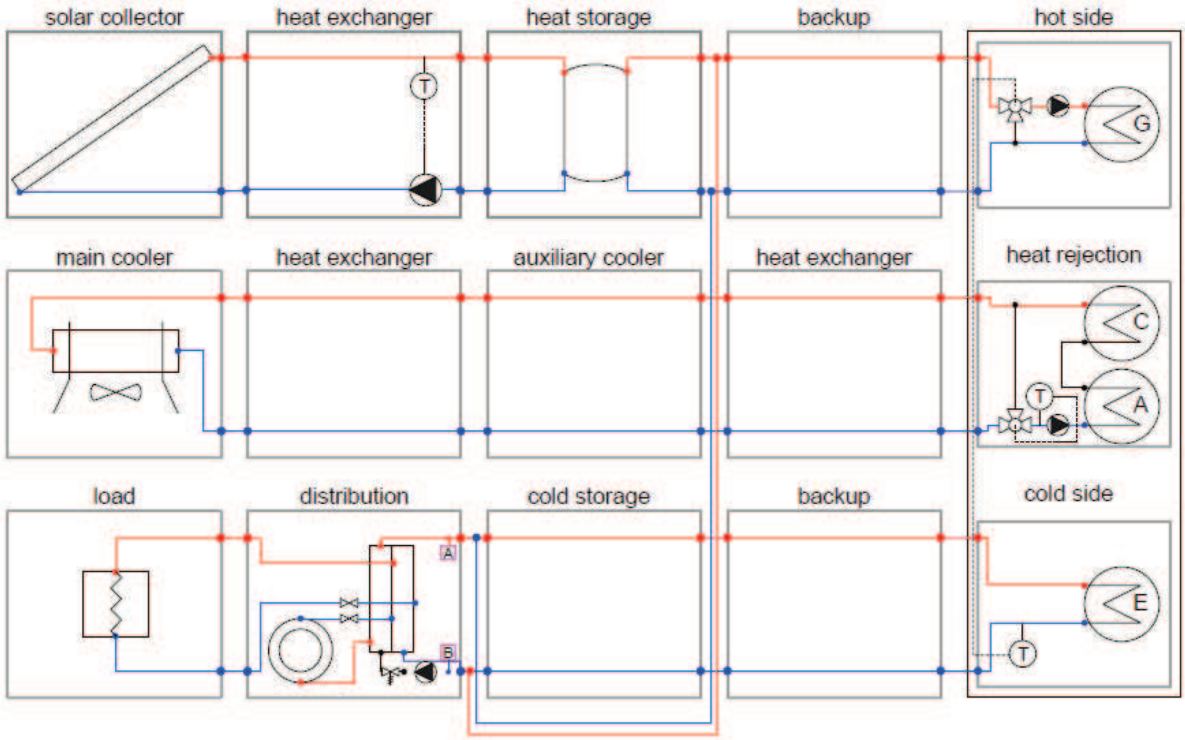


Technical premises

Distribution system

Task 38 standard Scheme for the installation:

If the standard Task 38 scheme is used to describe the installation, the scheme is as follows:



Task 38 standard hydraulic scheme for the Maclas installation

3 Control Strategy

The installation control is realized by the internal controller of the Sonnenkilma chiller. Indeed, this chiller goes with its internal regulation. This controller can be remotely programmed via an internet access and a VPN.

The control strategy is as follows:

The solar loop's pump starts when the solar radiation reaches a certain value. So the hot storage starts warming up.

Then, when the temperature at the top of the hot tank reaches a certain value (75°C), the three pumps of the chiller starts running (generator, evaporator, and condenser). And the cold production begins.

Of course, several setpoint and thresholds were established to stop the installation in case of problems or at the end of the day.

As it was explained in the previous paragraph, the system was designed to be as simple as possible. And it is the same principle for the control strategy.

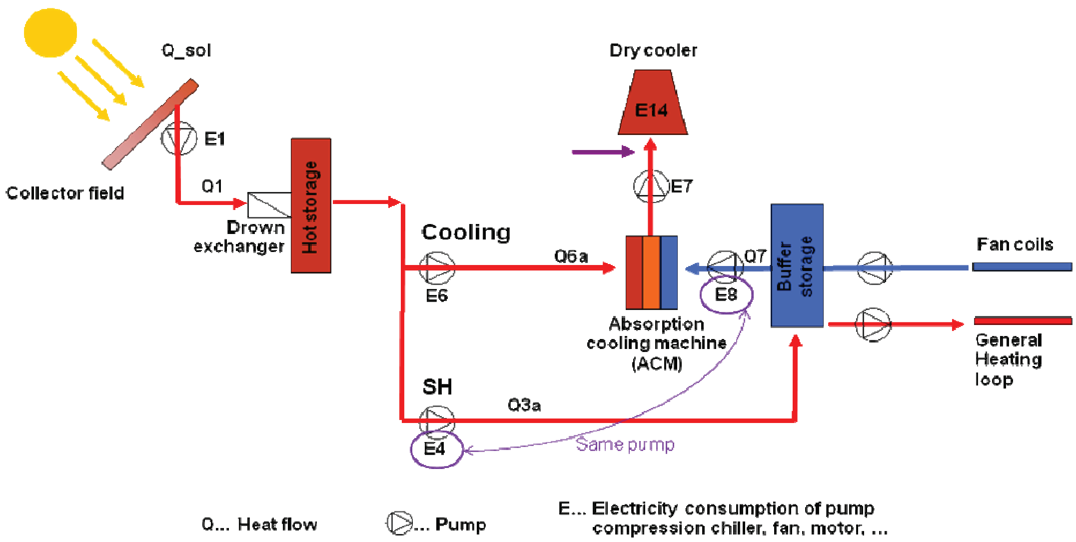
4 Monitoring Equipment

4.1 Installed Equipment

The measured data are basically the energy flow for every loop of the installation. As a consequence, at least two temperature measurement and one flowrate measurement are taken for each loop of the installation.

The solar radiation and the overall electricity consumption of the installation are also measured.

The Task 38 standard monitoring scheme below shows the different energies monitored:



Task 38 standard monitoring scheme for the Maclas installation

Going deeply into the details, the monitored data are:

VARIABLE	EXPLICIT MEANING	UNIT
Zeit	Date and time	
V_G	Flow in the generator piping	m³/h*100
V_E	Flow in the evaporator piping	m³/h*100
V_AC	Flow in the absorber/condenser piping	m³/h*100
E_TS	Solar Irradiation	W/m²
V_K	Flow in the collectors piping	m³/h*100
t_Gh	"temperature generator hot" = temperature at the entry of the generator	℃*10
t_Gc	"temperature generator cold" = temperature at the exit of the generator	℃*10
t_Ec	"temperature evaporator cold" = temperature at the exit of the evaporator	℃*10
t_Eh	"temperature evaporator hot" = temperature at the entry of the evaporator	℃*10
t_ACc	"temperature absorber/condenser cold" = temperature at the entry of the absorber	℃*10
t_ACh	"temperature absorber/condenser hot" = temperature at the exit of the condenser	℃*10
Q_G	Energy used by the generator	kW*100
Q_E	Energy accumulated at the evaporator	kW*100
Q_AC	Energy given at the absorber/condenser	kW*100
COP	Coefficient of performance	none*100
Q_Ballon_out	Heating Power in winter operation	kW*100
t_Koll	"temperature collectors" = where is it taken exactly? You have to check at site, as far as I know directly in the collector	℃*10
t_Kh	primary circuit solar coming from field	℃*10
t_Kc	primary circuit solar going to field	℃*10
t_sp11	Upper storage temperature	℃*10
t_sp12	Lower storage temperature	℃*10
t_stout	Storage outlet temperature (to SAC=Solar absorption chiller)	℃*10
t_stin	Storage input temperature (coming from SAC)	℃* 10
Q_Solar	Power Solar Circuit	kW*100
t_room_AKA	Temperature inside control unit	℃*10
Setpoint_Ec	Cold Water Setpoint	℃*10

4.2 Period of Measurement

Every data are measured and saved in daily "historic" files with a time scale 30 seconds. These files were downloaded and analyzed frequently about every week.

The data available for the Task 38 monitoring is only the summer 2009 (from June to September). Unfortunately, after this date a few issues prevented us to get the monitoring data (problems with the building owner, and it was not possible to get the monitoring data because of the Sonnenklima insolvency).

5 Monitoring Results

5.1 Annual / Monthly Data

The table below shows the energies considered in each loop of the installation. It also shows some of the main performance factors calculated monthly and yearly for the monitoring period:

	Yearly	June	July	August	September
	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Total Electricity Consumption	351,9	115,0	82,2	84,3	70,4
solar irradiation on collector aperture area	8516,9	3519,0	1584,4	1727,5	1686,0
solar thermal output to hot storage	3404,6	1387,9	795,3	607,3	614,1
hot storage input to cooling machine (ACM)	1693,3	511,9	346,5	447,0	387,8
cold output ACM to cold-storage	915,5	271,0	167,3	252,4	224,8
cold storage output to cold-distribution	869,8	257,5	158,9	239,8	213,6
	[-]	[-]	[-]	[-]	[-]
Collector efficiency (-)	0,40	0,39	0,50	0,35	0,36
Thermal COP (-)	0,53	0,48	0,56	0,58	0,53
Electrical COP (-)	2,24	1,93	2,84	3,03	2,24

During the considered monitoring period about 900 kWh of cold was produced and supplied to the building. The collector efficiency were about 0,40 which is good, but expected because the collectors are evacuated tubes collectors. The performance of the chiller is correct, indeed the yearly average of the thermal COP reaches 0,53. On the other hand, the electrical COP is a little bit lower than it could be expected: the average of the electrical COP for this cooling period reaches 2,24 which is not bad because only the cooling period was considered, but it could be better for a solar cooling installation. Actually this value of electrical COP can be explained because of the use of a drycooler to reject the heat. Indeed its two fans are consuming more than a conventional cooling tower, but it was the only affordable way to avoid every legionella risk.

5.2 Analysis of a typical good day: 23th August 2009

The table below summarizes the performances of the installation on 23th August 2009:

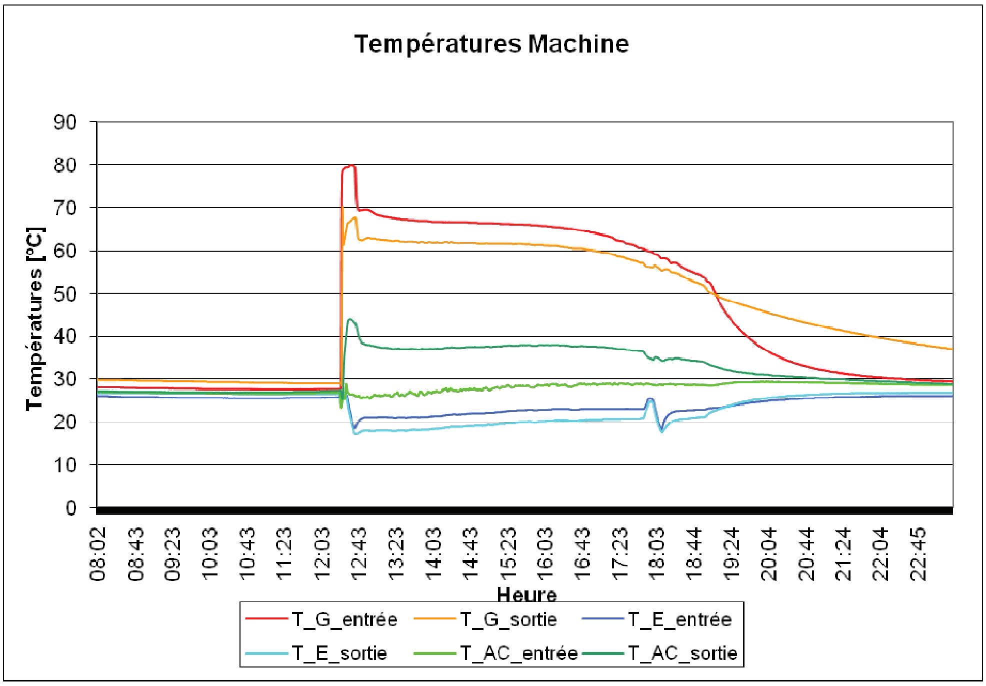
Ensoleillement	177,56	kWh
Energie solaire	60,39	kWh
Energie générateur	50,87	kWh
Energie Evaporateur	30,66	kWh
Total énergie élec utilisée	9,19	kWh
Rendement capteurs	**34,01**	**%**
COP thermique	**0,60**	
COP électrique	**3,34**	

During this day 30,66 kWh of cold was produced and supplied to the building. The collector efficiency were about 0,34 which is good, even if we could have expected a little bit better because the evacuated tubes collectors are used.

The performance of the chiller is good, indeed the daily average of the thermal COP reaches 0,60 which is a value which can be expected from this chiller.

The average of the electrical COP for this day reaches 3,34 which is good. Of course this value could have been better if another heat rejection system have been used, but this value is still good when a drycooler is used.

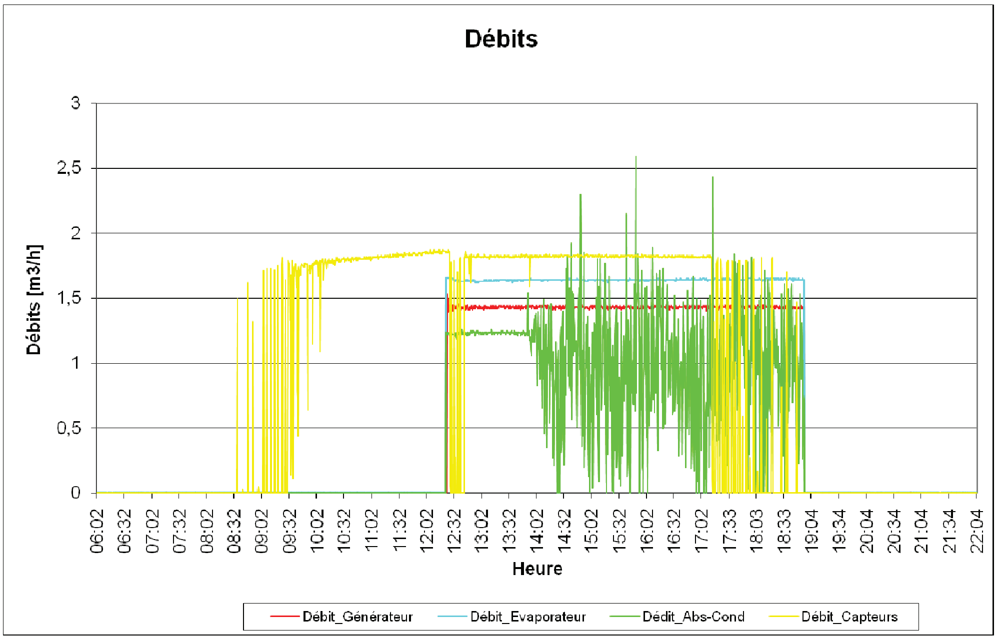
The chart below shows the temperatures at the input and output of the chiller:



Temperatures at the input and output of the chiller

It can be concluded the chiller works properly from about 12:30 to 19:00. The driving temperature at the generator is more or less always between 60℃ and 70℃, while the cold is produced at about 20℃.

The chart below shows the flowrates into the different loops of the installation:



Flowrates into the different loops of the installation

The solar pump is starting at about 9:00 and as said before the pumps for the other loops (generator, evaporator, and condenser) are starting at about 12:30.

The Generator and evaporator flowrates are pretty stable, but it's not the case for the condenser and for the solar pumps. Indeed, the solar pump works in short cycle at the beginning and at the end of the day, and when the chiller is starting running. And the condenser pump works in short cycle a few hours after the chiller started running and until the end of the day.

6 Experiences / Lessons Learned

As said before, no main problems were identified during the monitoring period. On the other hand a few problems happened after this period.

- The monitoring was not possible to do because of the insolvency of the chiller manufacturer, and because the monitoring devices were part of the chiller. But now this problem is solved, and the monitoring still can be done via a VPN.
- Because the installation stopped for a long period, the solar collectors which used a heat exchanger were cycling so now the performance of the collector field are lower than what it should be and what it was when the installation started.

7 Conclusions

The installation was able to run and have good performances for the entire monitoring period. Then the monitoring was not possible for some external reasons, and a few problems started. Now the installation is being renovated, and will start running soon.

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 5

Monitoring Results of CNRS PROMES Research Center Office, Perpignan (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

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Contents

1 Background

The targeted building where the solar cooling and heating system is installed is the CNRS research centre PROMES in Perpignan. This building is an office building, but there is also numerous labs into this building. This building is located at the “TECNOSUD” area in Perpignan (Languedoc Roussillon, France), close to the Mediterranean Sea, with thus a Mediterranean climate.

PROMES laboratory building

The building is a large one: more than 5 000 m² on 3 floors. The collector field is located on the building roof, and the other part of the system is located on the first floor. The installation is cooling and heating only a small part of the building.

Area cooled/heated by the solar system

The solar collector field has been installed in the same axis than the building, that is to say the solar collector field is oriented to the South-East (45° East), so the system works earlier in the morning but stops faster at night. The building was built in the year 2000 and presents a good level of energy efficiency.

2 Solar cooling and heating system description

2.1 Key components of the system

The system is based on an adsorption chiller of 7.5kW capacity, and a double glazed flat plate solar collector field of 25m². The system itself is working without any backup, and is thus an autonomous solar cooling and heating system. However, comfortable conditions into the building are insured by a general cooling and heating system (multi split system) in parallel and working independently to the solar system.

Technical data about the chiller

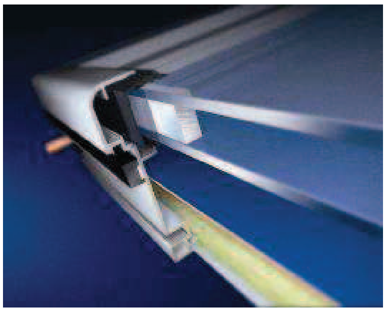
- Technology: closed cycle
- Nominal capacity: 7.5 kW cold
- Type: Adsorption
- Chiller brand: SORTECH
- Dehumidification: no

7,5 kW adsorption chiller used in Perpignan

Technical data about the collectors

- Collector type: High performances double gazed flat plate collectors
- Collectors brand: Schüco
- Collector field area: 25 m² (aperture area)
- Tilt : 30°
- Orientation: South/East (45°)
- Collector fluid: water
- Typical operation temperature: 75°C driving temperature for chiller operation

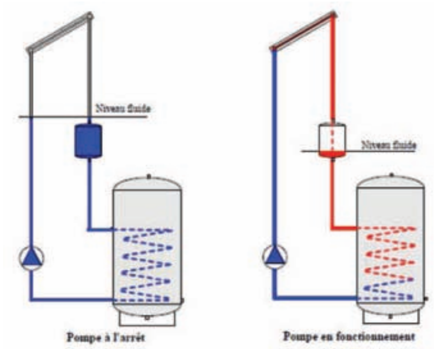
25 m² of double grazed solar collector



Double glazed technology

These solar collectors are auto-drainable. It was thus possible to design the installation with a drainback system. This system permits to have an installation safe (no risks of freezing or overheating with this system), efficient (performances of monitored drainback systems are very good), and simple (no expansion systems nor drain-cock devices).

The principle of this drainback system is as follows: when the solar loop pump is not working, the collector fluid level in slightly above than the drainback tank. The collectors are thus filled in with air at atmospheric pressure. This amount of air in the system permits to absorb the expansion of water when the system is working. When the solar loop pump is starting, the manometric pressure of the pump permits to pump up the level of fluid until a level above the solar collectors, as a consequence, the air is driven out to the drainback tank. The volume of air inside the loop permits to drain the collector field and prevent it to freeze or overheat, and it also permits to absorb the expansion of water when the system is working. In the loop neither water not air are replaced when the system is working, so there is no need to treat water, and there is no risk of corrosion.



Scheme of a drainback system

The energy distribution into the building is managed thanks to a independent loop. This loop is compatible with cold and with hot water. This loop is connected to three fan coils working at a temperature level of 14/18°C.

Fan coils for the distribution system

The heat rejection is managed by a drycooler. To insure a sufficient heat rejection for the entire cooling period, the drycooler is equipped with a water spraying system. This system is used only for the very warm days in summer, when the drycooler alone can't insure a sufficient heat rejection from the system.

Drycooler used at the perpignan installation

A small hot storage of 300 liters is installed between the solar collectors and the chiller. An internal exchanger is present in this tank. This storage is actually a buffer storage, which permits to smooth the solar radiation variations.

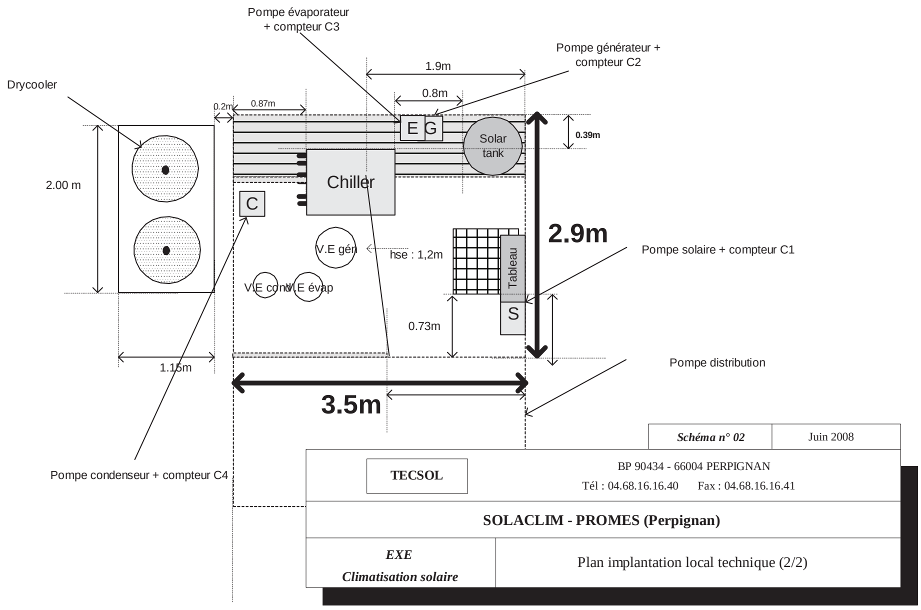
Similarly, a small cold storage is also installed between the chiller and the distribution circuit.

Hot storage tank of 300 liters

2.2 The system in its entirety

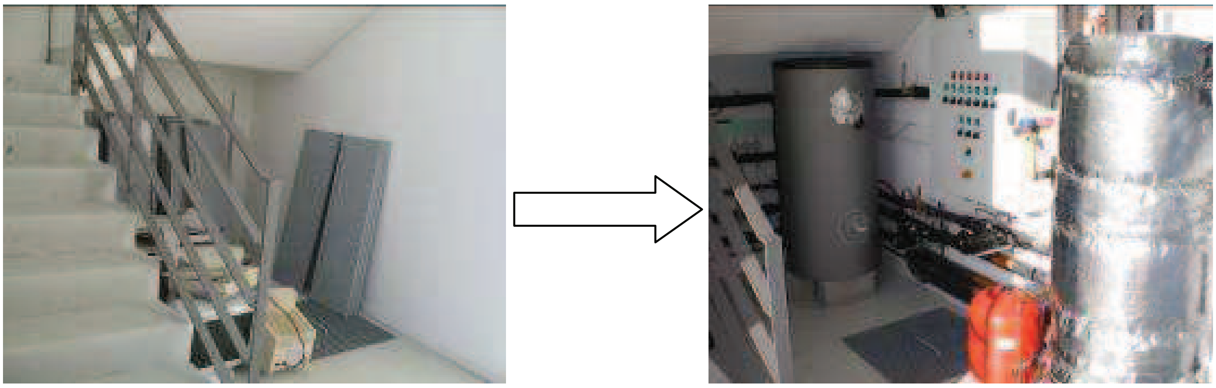
The layout constraints were very large for this installation. Indeed, the only space available for the system was located into one of the building stairwell. The system had to be designed very carefully and as cleverly as possible to avoid any trouble in the traffic of the stair.

The implantation scheme is as follows:



System implantation scheme

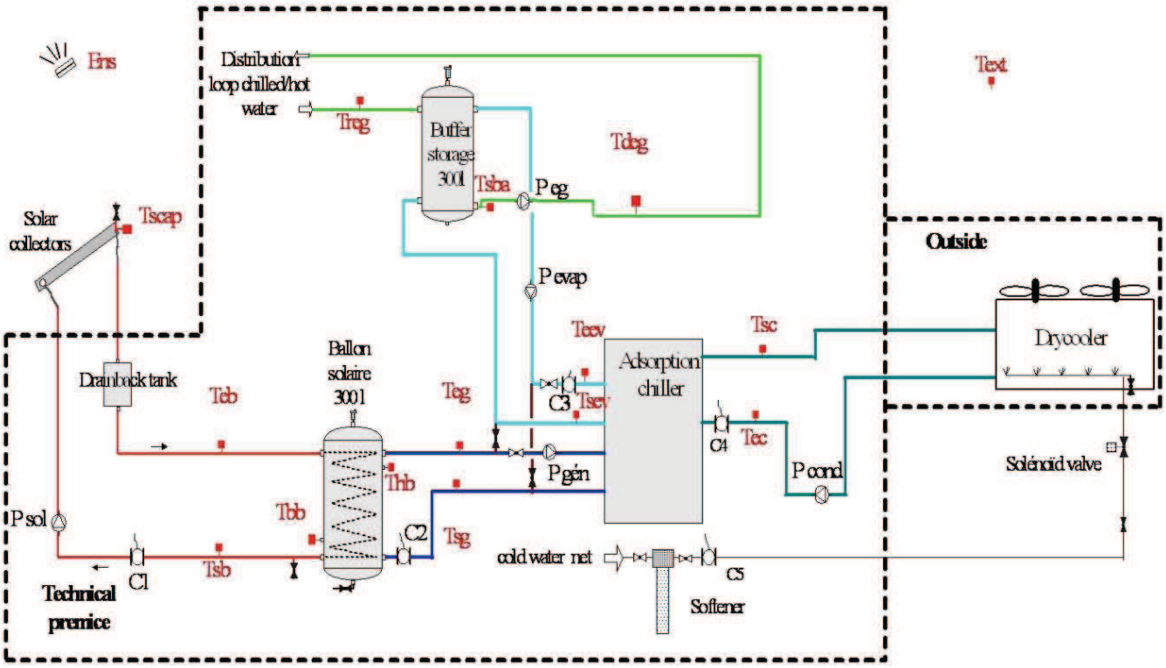
So it was possible to implant the entire system into this small spot of about 10 m², and a large part of it is located under a stair.



Technical premises before and after the works

The installation was designed to be used following to working mode: the first one is the cooling mode which is used in summer from May to October, and the second one is the heating mode which will be used during winter from November to April.

The installation's hydraulic scheme is showed below:



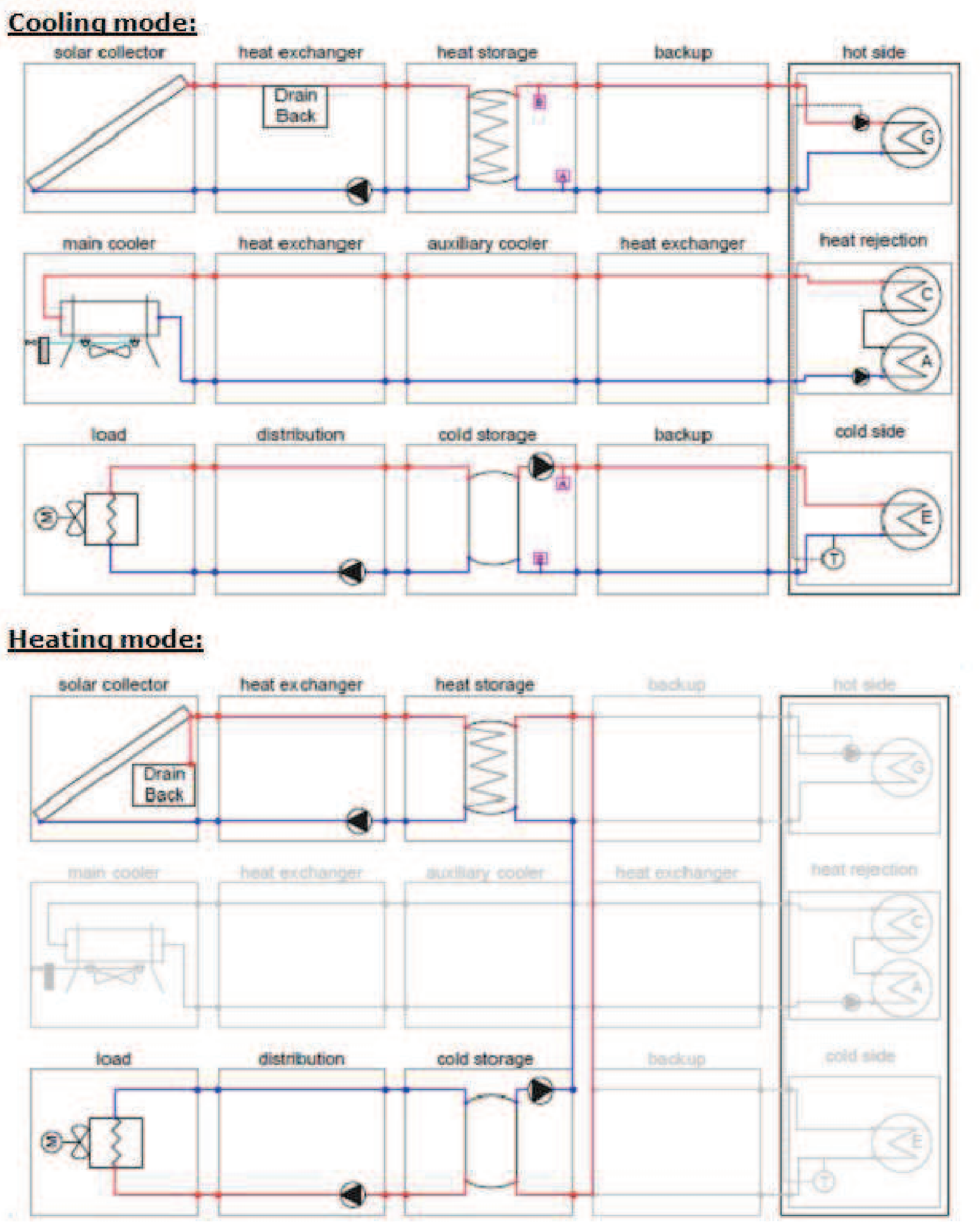
Installation's hydraulic scheme

When the installation is used in cooling mode, the circuit between the generator circuit to the evaporator circuit (in brown on the scheme above) is not used.

When the installation is used in heating mode, this circuit is used, and the chiller is by-passed, and the drycooler is not used neither.

For the monitoring of the installation, the thermal energies of each circuit are calculated by measuring the temperatures and the flow rates. The solar radiation is also measured as well as the total electricity consumption to characterize the energy required to drive the system..

The standard task 38 hydraulic scheme for this installation is shown below:



3 System monitoring and performance calculation

The first start of the system was done in July 2008.

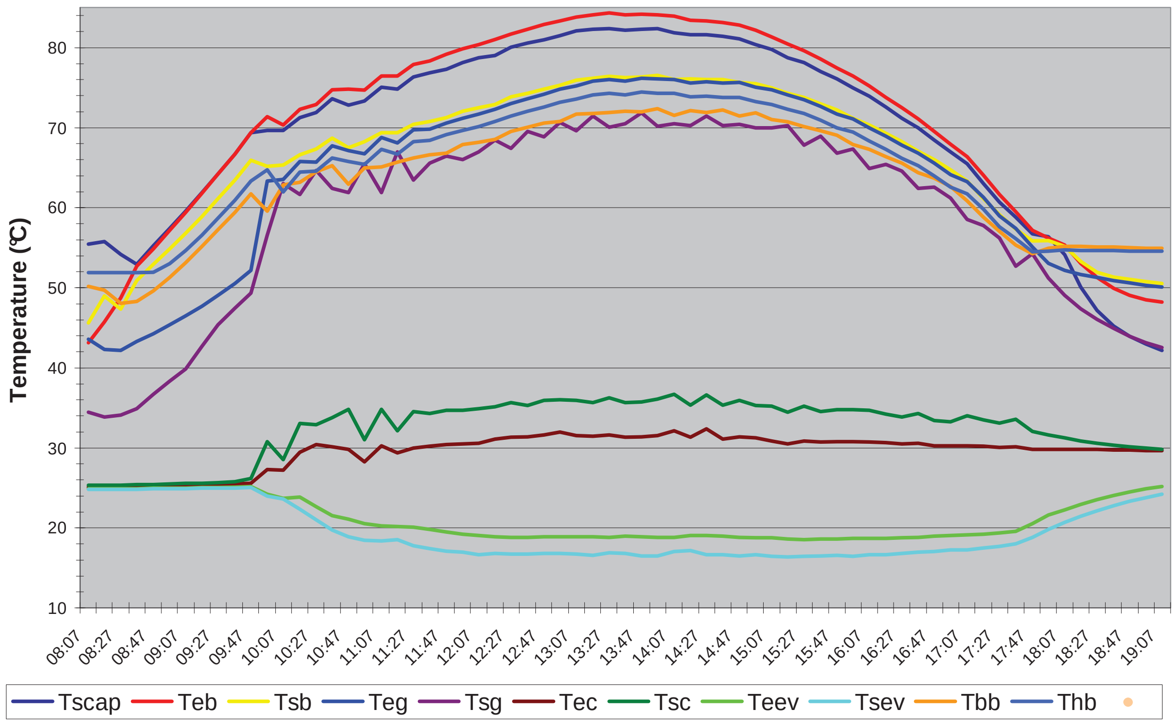
Measurement with a time scale of 1 and 10 minutes were automatically sent daily. It was then possible to calculate balances on a day long, a month long or a year long.

3.1 Daily analysis

3.1.1 <u>Sunny day for the cooling mode: 15 July 2008</u>

The 15th July 2008 was considered to evaluate the performances of the installation in cooling mode.

<u>Temperatures:</u>

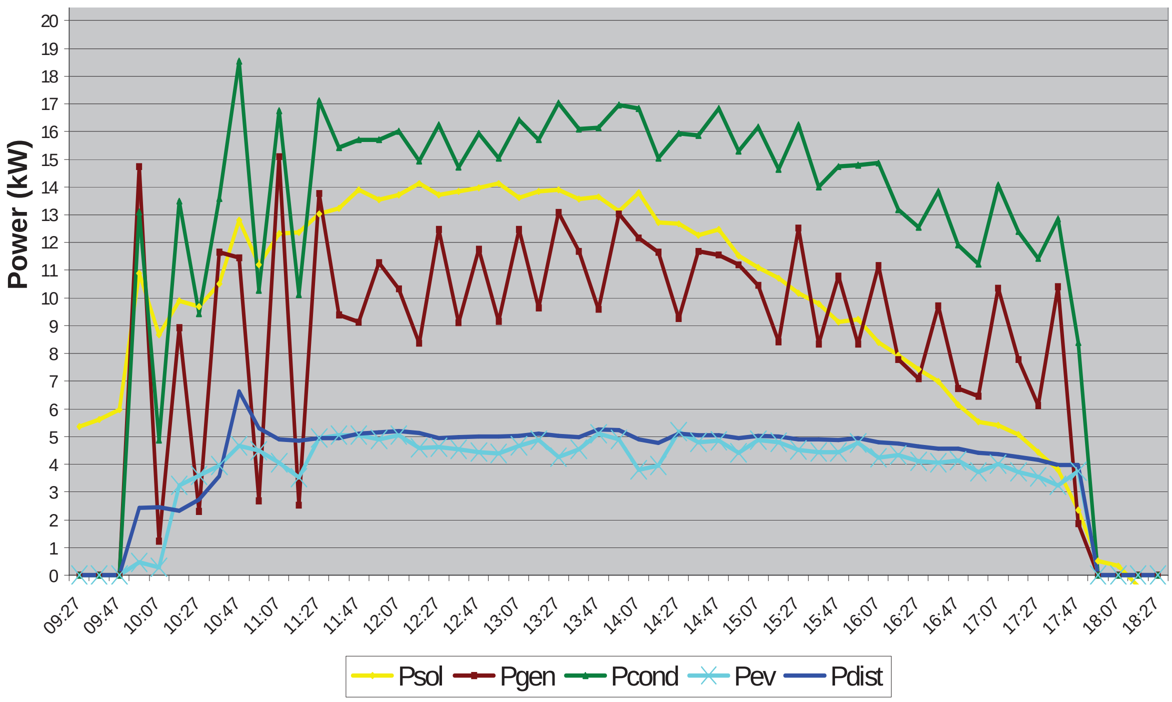


Temperatures for the day 15/07/2008

The behavior of the installation is very normal and correct for an adsorption technology. The temperatures into the solar collectors are about 80/75 °C at the maximum capacity. This level of temperature allows the chiller to produce chilled water at about 15 to 20 °C. And the heat is rejected via the drycooler at about 35 to 30°C.

Power:

The power at each circuit could be calculated ad are shown below:



Powers for the day 15/07/2008

The power in the solar loop depends of the solar radiation coming to the collector field, and reaches about 14 kW. When the installation is working, the power going to the chiller at the generator loop reaches about 11kW. Thanks to this power at the generator, the chiller is able to produce about 5 kW cold, so a value not far from its nominal capacity. The power rejected by the drycooler is thus about 16 kW.

Energy balance:

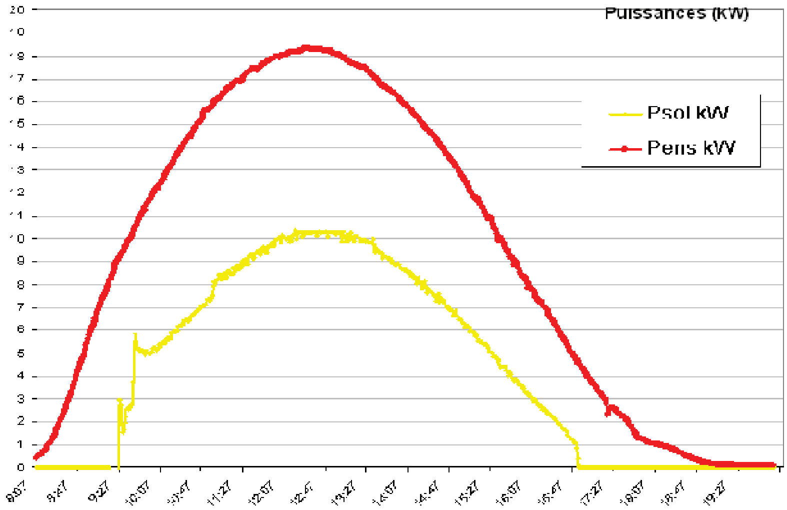
- The energies for the day 15/07/2008 are:
- Solar radiation: 260 kWh
- Energy collected by the solar collectors: 90 kWh
- Energy coming to the generator: 76 kWh
- Energy produced by the chiller at the evaporator: 33 kWh
- Energy rejected by the drycooler: 116 kWh
- Electricity consumption: 5,8 kWh
- Thermal COP = Eevap/Egen: 0.44
- Electrical COP = Eevap/electricity consumption: 5.8

Very good performances were thus monitored when the Perpignan installation was in cooling mode.

3.1.2 Sunny day for the heating mode: 15 July 2008

The switch to the heating mode was performed in October. The day studied here is the 15/10/2008.

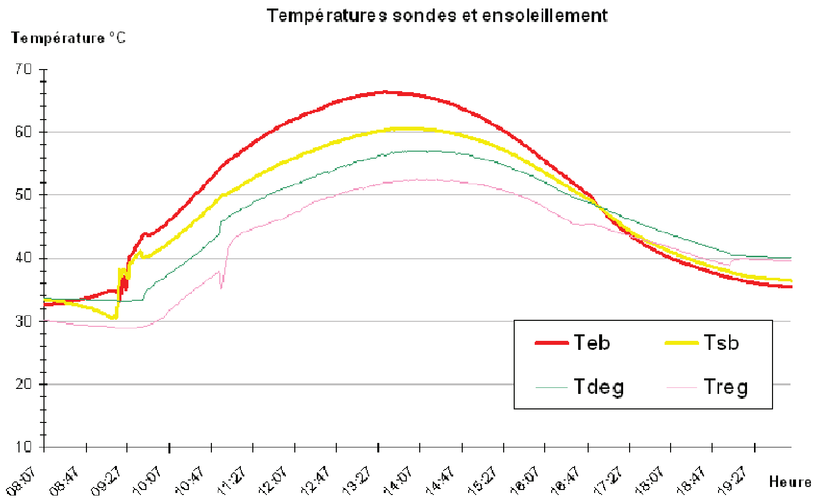
It was possible to evaluate the collector performances in heating mode. The solar radiation power and the power collected by the solar loop are showed on next chart:



Solar radiation and solar loop power

A daily average was calculated and leads to a collector field efficiency of 52%.

It was also possible to calculate the energies collected by the collectors, but also the energy produced and distributed to the building.

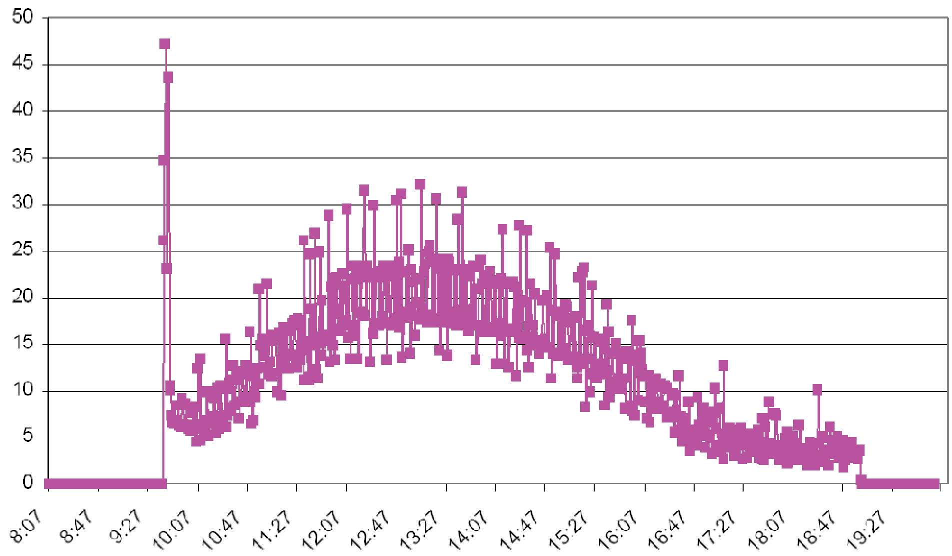


Inlet and outlet temperature of the collector and of the distribution

After calculation of the energy balance, the heating production is about 27 kWh.

The electrical COP was also calculated following the equation :

Electrical COP = heating energy produced / electrical energy consumed.



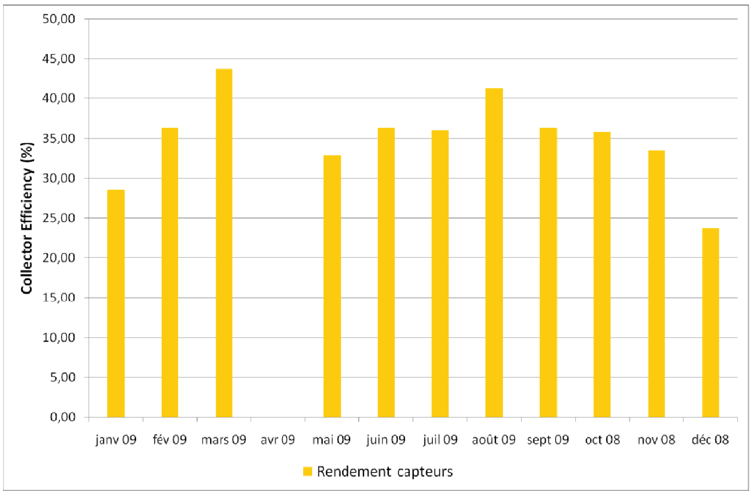
The average electrical COP for the day is 12.

3.2 Monthly and yearly analysis

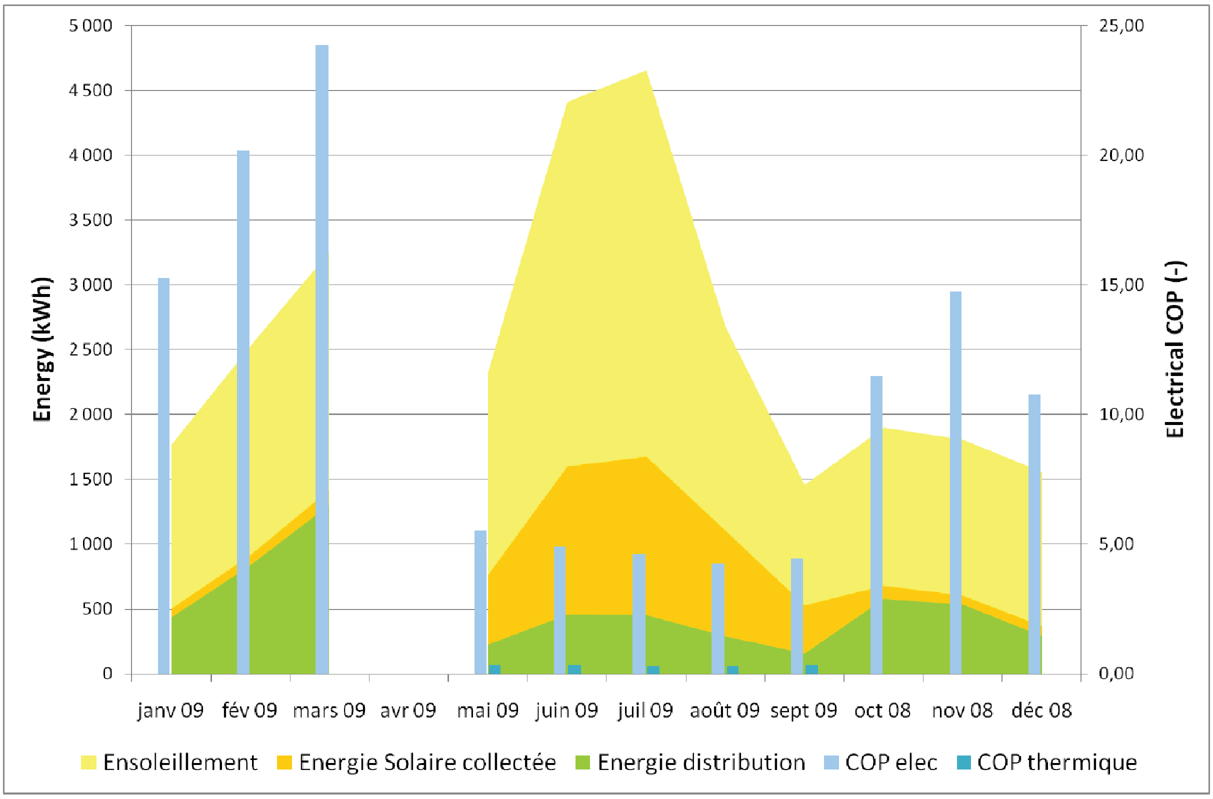
The complete analysis of an entire working year (2008-2009) showed excellent results. It is noticeable that the overall electric efficiency of the system for one entire year reaches about 10. That means this solar cooling and heating system produces 10 times more energy (hot and cold) than it consumes electricity. As a consequence the system is far more efficient that the best heat pumps currently on the market.

Bilan thermique :	TOTAL	janv.-09	févr.-09	mars-09	avr.-09	mai-09	juin-09	juil.-09	août-09	sept.-09	oct.-08	nov.-08	déc.-08
Ensoleillement	28357,04	1765,67	2527,81	3253,03		2328,92	4414,36	4657,99	2679,11	1460,24	1902,27	1807,95	1559,67
Energie Solaire collectée	10186,57	504,57	918,08	1422,82		765,12	1603,17	1678,43	1107,34	530,28	682,08	605,06	369,64
Energie distribution	5573,79	436,21	838,38	1305,40		226,61	457,35	454,43	290,21	157,68	579,77	536,12	291,63
Rendement capteurs	35,92	28,58	36,32	43,74		32,85	36,32	36,03	41,33	36,31	35,86	33,47	23,70
COP thermique	0,33					0,35	0,33	0,31	0,29	0,35			
Bilan électrique :													
COP elec	9,72	15,26	20,17	24,26		5,52	4,89	4,64	4,27	4,46	11,49	14,74	10,79
conso elec	615,97	31,07	45,18	58,48		43,51	98,93	104,66	72,70	37,70	54,83	39,53	29,37
conso sans distrib	573,47	28,59	41,57	53,80		41,02	93,48	97,90	67,93	35,35	50,44	36,37	27,02

Performances of the installation for the entire year October 2008 to September 2009



Efficiency of the solar collector field during the year 2008-2009



Graphic representation of the installation performances for the year 2008-2009

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning and Refrigeration

D-A3b: Appendix 6

Monitoring Results of INES Research Center Offices, Chambéry (France)

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30th November 2010

By Romain Siré[1] and Daniel Mugnier[1]

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Contents

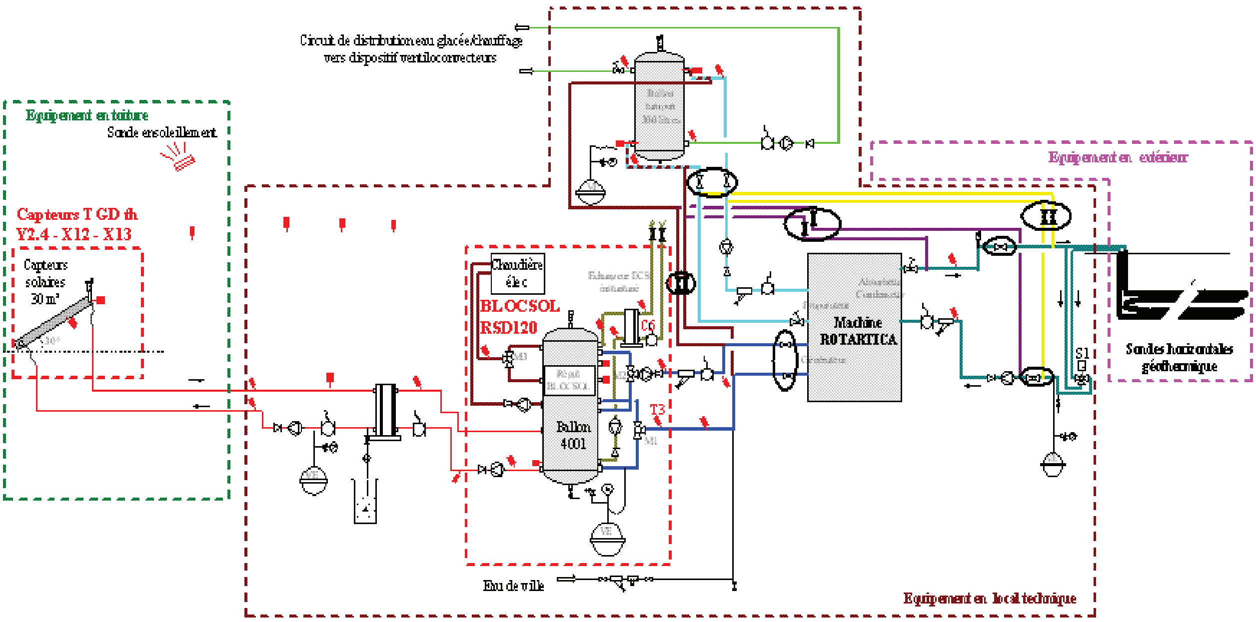
1 Introduction

The targeted building welcoming the solar cooling application is the PUMA3's INES research office. The INES (National Institute of Solar Energy) was created in 2006 by the public institutions to promote and develop solar technologies in France. To reach these objectives, the INES is divided into two centers: research, development and innovation on the one hand, and training and education on the other hand. The INES is located in the "Savoie Technolac" area which is very close to Chambéry in Rhônes-Alpes area (close to the French Alps and Lyon). The PUMA3 building is large, but only 3 mezzanine offices are cooled down by the solar system. The building was created recently so it has a good level of energy efficiency.

2 System design

The system is based on an absorption chiller of 4.5 kW coupled with 30 m² flat plate collectors. The installation is cooling the building in summer, heating in winter, and it also producing a small amount of domestic hot water all year long. The 400 liters heat storage tank is included in a packaged device supplied by Clipsol (SSC BlocSol RSD 120). Included in this device there is also an electric backup and all the security devices related to the heat storage. Another tank is installed in the system which is ensuring the cold storage in summer and a second heat storage in winter. The heat rejection system is done by a horizontal geothermal field.

Installation's scheme:



Building:

- Type of building: Office building
- Location: Chambery, France
- In operation since: April 2009
- System operated by: INES
- Air-conditioned area: 21m²
- System used for space cooling and heating (DHW production possible but not used)

System general properties:

- Technology: Closed cycle
- Nominal capacity: 4.5 kWcold
- Type of closed system: Absorption
- Brand of chiller unit: ROTARTICA
- Chilled water application: Fan coil
- Dehumidification: No
- Heat rejection system: Geothermal probes, exchange area about 138m²

Solar thermal:

- Collector type: Flat plate collectors
- Brand of collector: CLIPSOL
- Collector area: 30m²
- Tilt angle, orientation: 30°, 10°
- Collector fluid: Water glycol
- Typical operation temperature: 80°C

Configuration:

- Heat storage: 0.4 m3 water (part of:SSC BlocSol RSD 120 CLIPSOL)
- Cold storage: 0.3 m3 water
- Use of auxiliary heating system: Hot backup (heating the heat storage tank)

Photos:

Technical premises

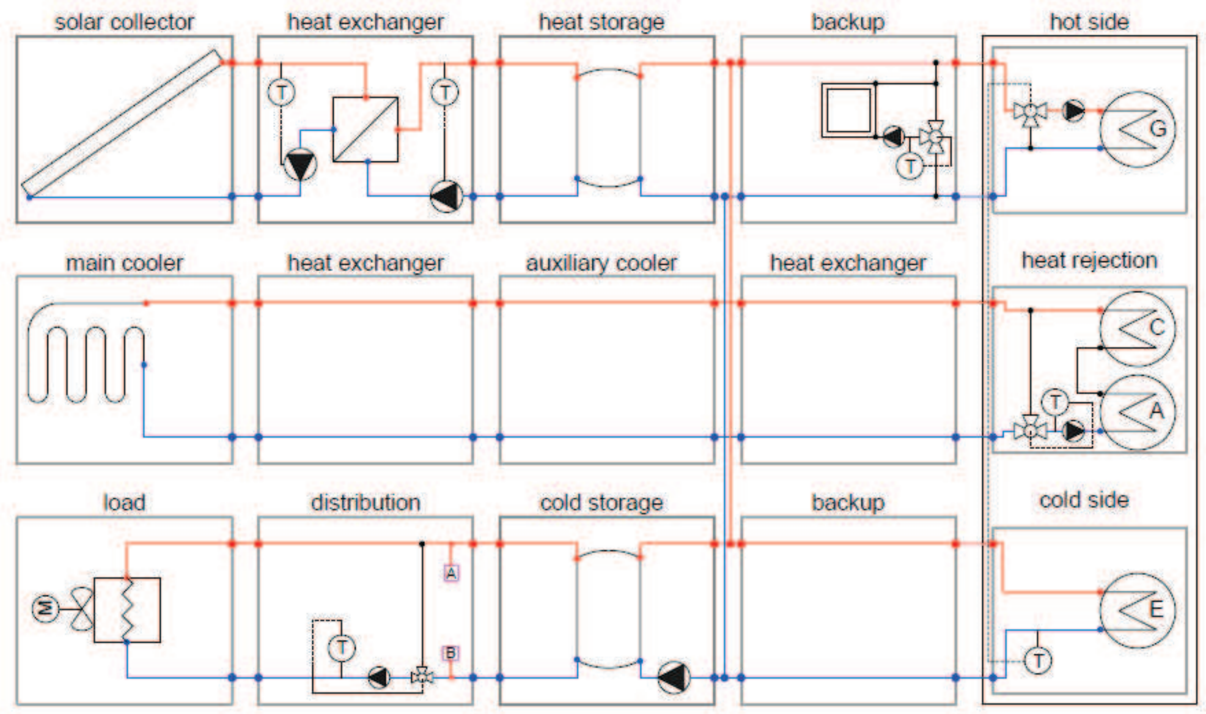
Collector field

Geothermal field

Distribution system

Task 38 standard Scheme for the installation:

If the standard Task 38 scheme is used to describe the installation, the scheme is as follows:



3 Performance factor used for the Chambery installation

3.1 Background

Within IEA Task 38 a universal monitoring procedure for solar heating and cooling systems was developed. This monitoring procedure can be conducted in three different levels of detail:

1. First level: Basic Information on Primary Energy Ratio and Costs
2. Second level: Simple analysis of the solar energy source management
3. Third level: Advanced monitoring procedure

The IEA Task 38 also developed a standard basic scheme adjustable for the large majority of systems. In this scheme, the energy flows of a solar heating and cooling system are nominated and shown.

Abbreviations:

SH Space Heating

DHW Domestic Hot Water

SC Space Cooling

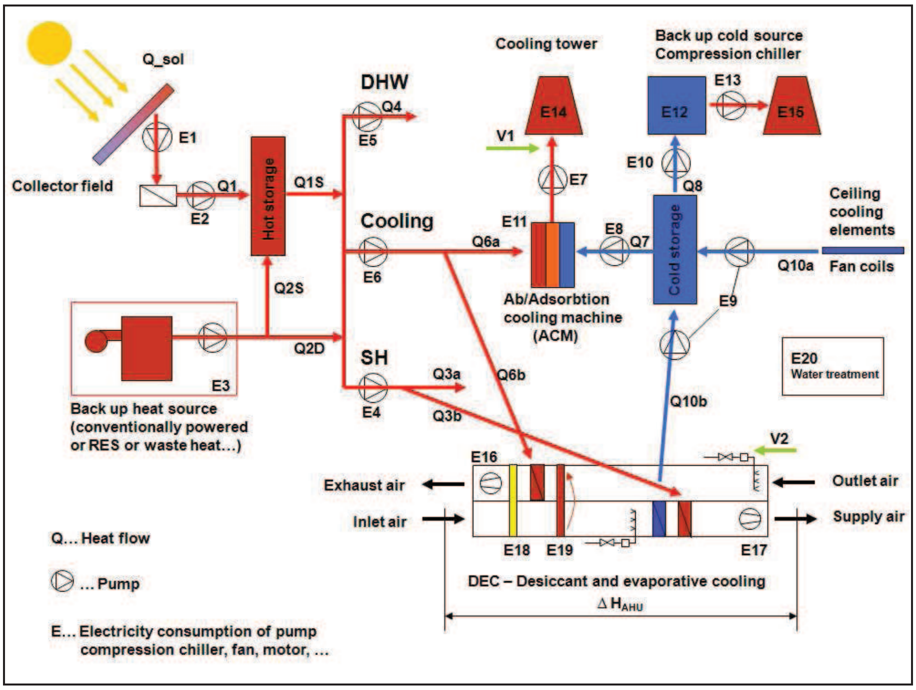
RES Heat recovery from cogeneration unit or biomass...

AHU Air Handling Unit

εelec Primary energy factor for electricity production (kWhel/kWhprim-fossil)

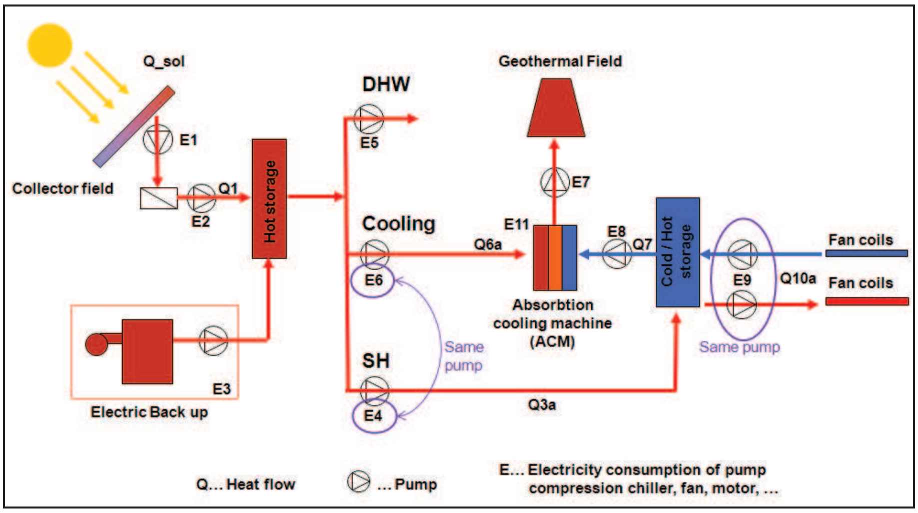
εfossil Primary energy factor for fossil fuel (kWhfinal/kWhprim-fossil)

εRES Primary energy factor for renewable sources (kWhfinal/kWhprim-fossil)



Task 38 base SAC maximum system scheme

This standard scheme was modified to match the characteristics of the Chambery installation:



Task 38 scheme adapted the the Chambery installation

Several performance factor were selected to characterize the installation performances:

3.2 Collector efficiency

Basically the collector efficiency characterise the performances of the solar collector field.

It is calculated as follows:

$$\eta_{coll} = \frac{Q_{sol}}{G}$$

With:

η_{coll} Collector efficiency (-)

Q_{sol} Solar energy collected in the solar loop (kWh)

G Solar radiation (kWh)

This performance factor is calculated for a certain period. Usually it is calculated for one day, or for one month.

Indirectly, the collector efficiency permits also to characterize the overall performances of the installation. Indeed, if the installation is turned off, because of a technical issue, or because there is no charge, the energy collected is equals to zero while the solar radiation is still there. So the collector efficiency value is low.

3.3 Thermal COP

The thermal coefficient of performance (thermal COP) characterizes the internal performances of the chiller. This performance factor is calculated only in summer (cooling mode of the installation) as follows:

$$thermal\ COP = \frac{Q_{evap}}{Q_{gen}}$$

With:

Thermal COP Thermal coefficient of performance (-)

Q_{evap} Energy taken from the evaporator loop (kWh)

Q_{gen} Energy coming from the generator loop (kWh)

Usually adsorption chillers have a thermal COP around 0.4 to 0.5 and absorption chillers have a thermal COP around 0.6 to 0.7.

3.4 Electrical COP

The electrical coefficient of performance (electrical COP) characterizes the electric performances of the entire solar installation. This performance factor is calculated all year long as follows:

$$electrial\ COP = \frac{Q_{distrib}}{E_{tot}}$$

With:

Electrical COP Electrical coefficient of performance (-)

$Q_{distrib}$ Energy (hot or cold) given to the distribution loop (kWh)

E_{tot} Electrical energy consumed by the solar system (except backups) (kWh)

This ratio is used to compare the performances of the installation to the performances of a conventional system such as heat pumps.

3.5 Primary Energy Ratio (PER)

This performance factor characterizes the overall efficiency of the installation in terms of primary energy. It is the ratio between all the energies given to the user (heating, cooling, and DHW) and all the primary energies consumed by the system (including backups).

The more this performance factor is high, the more the solar fraction is high and/or the overall efficiency of the installation is high.

$$\boldsymbol{PER} = \frac{\boldsymbol{Q}10 + \boldsymbol{Q}3 + \boldsymbol{Q}4}{\boldsymbol{E}_{\boldsymbol{système}} \cdot \quad_{\boldsymbol{élec.}} + \frac{\boldsymbol{Q}2}{\boldsymbol{Rg}} \cdot \quad_{\boldsymbol{x}} + \frac{\boldsymbol{Q}8}{\boldsymbol{EER}} \cdot \quad_{\boldsymbol{x}}}$$

See the SOLERA deliverable No. D 6.1, or the Task38 deliverables for more explanation about the meanings of the different terms of the equation.

3.6 Fractional solar heating and cooling savings

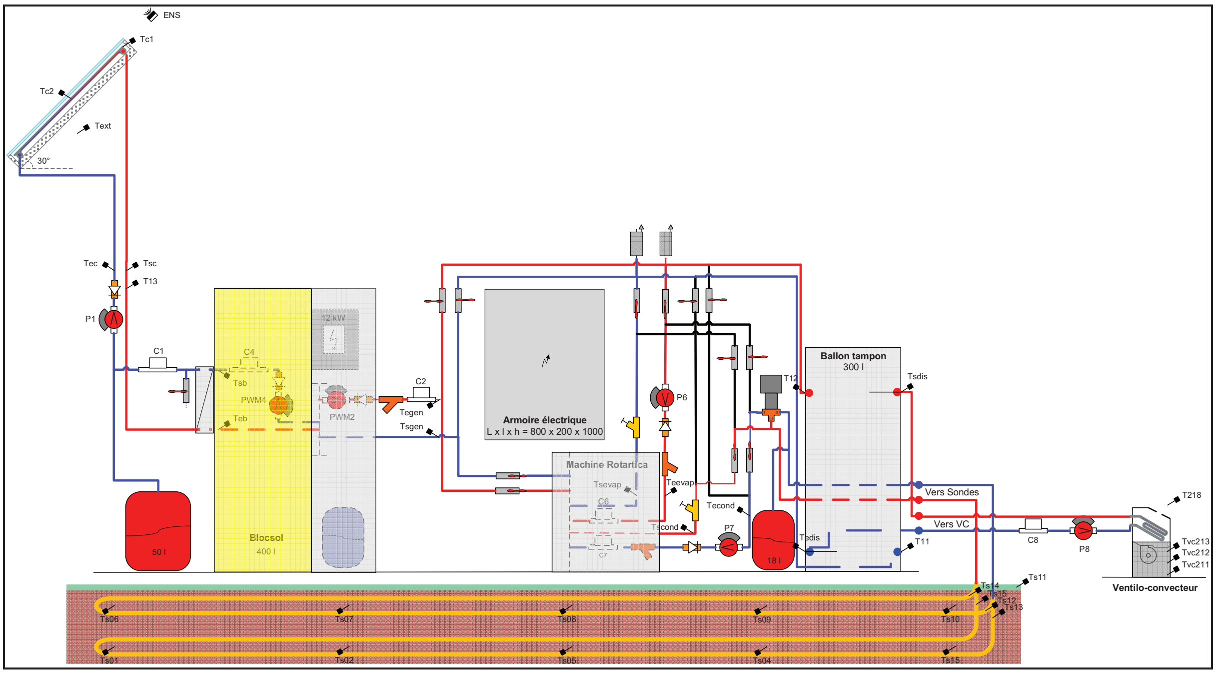
The global evaluation of the whole system was performed by using the fractional solar heating and cooling savings fsav,SHC as actually defined in the IEA-SHC Task 38. It describes the fraction of energy savings of the solar system compared to a conventional system that provides the same service for heating, cooling and domestic hot water demand. It is calculated as follows:

$$f_{sav,shc} = 1 - \frac{\frac{Q_{boiler}}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \frac{Q_{RES}}{\varepsilon_{RES} \cdot \eta_{RES}} + \frac{E_{el}}{\varepsilon_{elec}} + \frac{Q_{cooling,missed}}{SPF \cdot \varepsilon_{elec}}}{\frac{Q_{boiler,ref}}{\varepsilon_{fossil} \cdot \eta_{boiler,ref}} + \frac{E_{el,ref}}{\varepsilon_{elec}} + \frac{Q_{cooling,ref}}{SPF \cdot \varepsilon_{elec}}}$$

See the SOLERA deliverable No. D 6.1 for more explanation about the meanings of the different terms of the equation.

4 Monitoring scheme and list of measurements

In order to control the system, to obtain good monitoring data, and to be able to calculate the performance factors described above, the following sensors are used:



Position of the monitoring equipment

The sensors used are listed below:

Table 1: list of the Chambery installation's sensors

Type	***Locations***	***Names***
Temperature sensor	on the collector field	Tc1 & Tc2
Temperature sensor	in the primary solar loop	Tec, Tsc & T13 (Blocsol)
Temperature sensor	in the secondary solar loop	Tsb, Teb & 1 inside the Blocsol
Temperature sensor	at the top and bottom of the hot water storage tank	inside the Blocsol
Temperature sensor	in the generator loop	Teg, Tsg & 2 inside the Blocsol
Temperature sensor	for the electric back up	inside the Blocsol
Temperature sensor	in each branch of the condenser loop	Tecond & Tscond
Temperature sensor	in each branch of the evaporator loop	Teevap & Tsevap
Temperature sensor	At the top and bottom of the cold water storage tank	T11 & T10 (Blocsol);

Temperature sensor	in each circuit of the distribution loop	Tsdis & Tedis
Temperature sensor	in the sanitary hot water loop	inside the Blocsol
Temperature sensor	ambient temperature sensor	Text
Temperature sensor	in each office heated and cooled	Tvc211 → Tvc213
Temperature sensor	in the office located in the middle	T212
Temperature sensor	in the ground next to the geothermal probes	Ts1 → Ts11
Temperature sensor	on each loop of the two geothermal probes	Ts12 → Ts15
Flow meter	in the primary solar loop, on the inlet pipe of the pump	C1
Flow meter	in the secondary solar loop, on the inlet pipe of the exchanger	C4
Flow meter	in the generator loop on the outlet pipe of the pump	C2
Flow meter	in the condenser loop, on the inlet pipe of the absorption machine	C7
Flow meter	in the evaporator loop, on the inlet pipe of the machine	C6
Flow meter	in the distribution loop, on the inlet pipe of the pump	C8
Irradiation sensor	On the roof tilt & orientation like the collectors	ENS

See the SOLERA deliverables for more information about the monitoring itself.

5 Analyse of the monitoring data

5.1 Collector efficiency

The collector efficiency can be calculated and followed every two minutes thanks to an efficient monitoring. It is then possible to draw the collector efficiency during one day.

The Figure 1 shows the collector efficiency of the Chambery installation on the 6th June 2010.

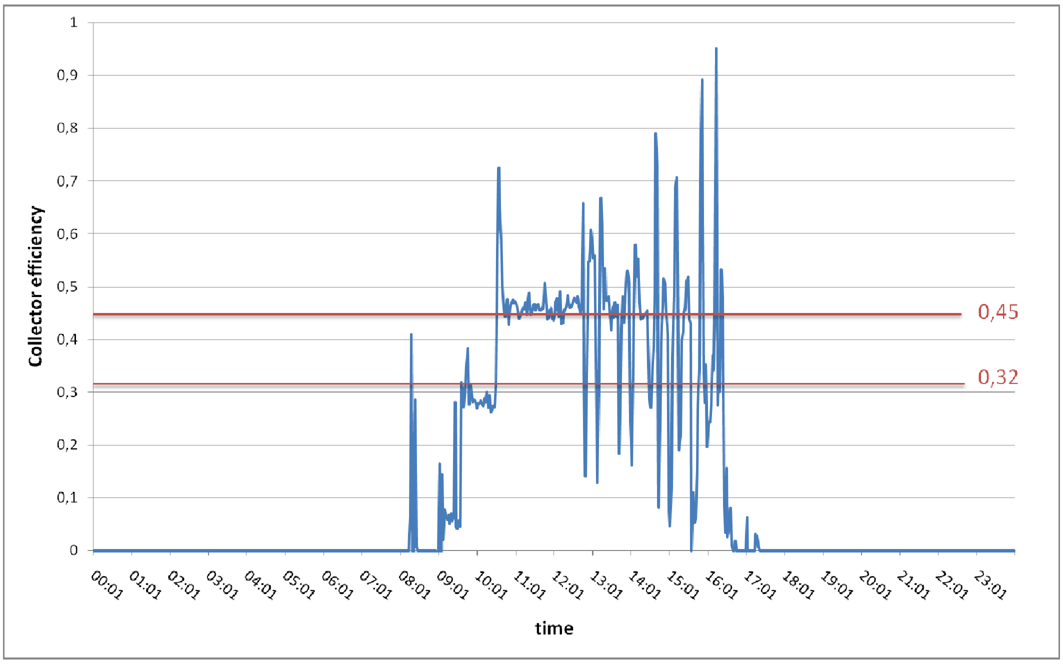


Figure 1: Collector efficiency of the Chamberry installation on 06/06/2010

During this day the solar collector field is working properly as the installation is running in stationary state between 10:30am and 4:30pm, the collector efficiency is around 0,45 which is a correct value with regard to the expected performances.

However, when the calculation is performed during the whole day, solar irradiation is 184,13kWh while solar energy received is 59,77kWh: the whole day collector efficiency is then 0,32.

This difference between the collector efficiency function of the time of observation can be explained because in the morning and in the evening, when the solar radiation is low, solar collectors can’t work at they full potential. Anyway, a collector efficiency value of 0,32 for the entire day is still a correct value.

The calculation for one month follows the same principles.

For the period between 01/05/2009 and 31/10/2010, the monthly collector efficiency is showed on Table 2.

Table 2: Collector efficiency of the Chambery installation during the observation period

May 2009	Jun 2009	Jul 2009	Aug 2009	Sep 2009	Oct 2009	Nov 2009	Dec 2009	Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
23,2 %	**27,7 %**	**30,8 %**	**31,2 %**	**20,3 %**	**21,2 %**	**11,9 %**	**9,7%**	**10,0 %**	**17,9 %**	**22,9 %**	**24,5 %**	**26,2 %**	**23,7 %**	**26,1 %**

The collector efficiency is good for flat plate collectors in the summer and for the sunny months. However in winter the efficiency goes low. It can be explained by the large amount of days in winter that are not sunny enough to start the solar system, or not sunny enough to keep the system working. However, for those days a solar radiation is still there even if it's not very high. As a consequence, the average collector efficiency is not very high.

Indeed, even in December, a few days appears with good collector efficiency, like on 08/12/2009 (see Figure 2) where the collector efficiency reaches 0,4. These days are just not so numerous than in summer.

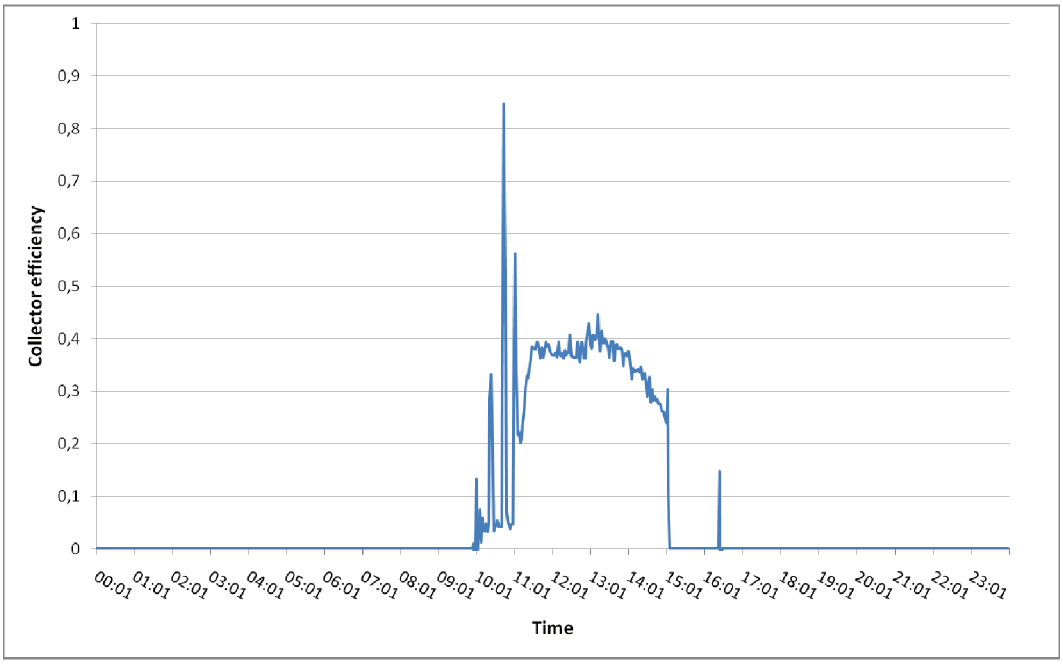


Figure 2: Collector efficiency of the Chamberry installation on 08/12/2009

5.2 Thermal COP

The Thermal COP can be calculated and followed every two minutes thanks to an efficient monitoring. It is then possible to draw the thermal COP during one day.

The Figure 3 shows the thermal COP of the Chambery installation on the 14th July 2009.

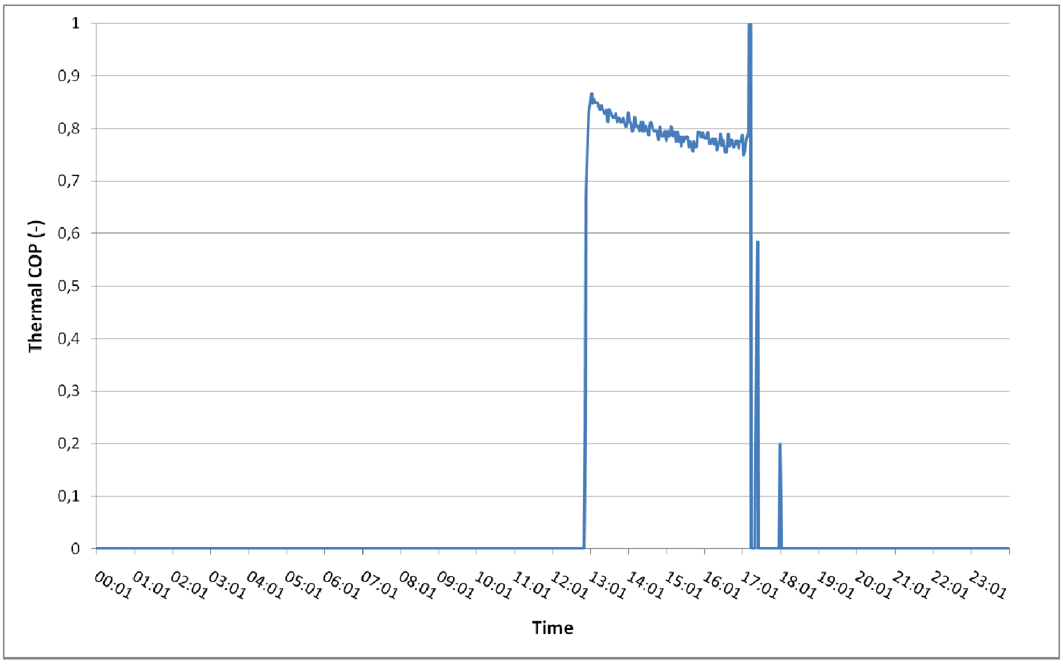


Figure 3: Thermal COP of the Chamberry installation on 14/07/2009

Looking at this day the Rotartica chiller exhibited an excellent (high and stable) coefficient of performance.

The chiller started on 14/07/2009 at about 12:50 and stopped in the evening around 5:10pm. During this time as the installation was producing cold, the internal thermal COP of the chiller was about 0,8 which is an excellent value even for an absorption chiller.

If the overall working period is studied, the thermal COP can be calculated month by month, and are showed on Table 3and on Figure 4:

Table 3: Thermal COP of the Chambery installation during the working period

May 2009	Jun 2009	Jul 2009	Aug 2009	Sep 2009	Oct 2009		May 2010	Jun 2010	Jul 2010
0,68	0,68	0,73	0,70	0,73	0,75		0,75	0,72	0,72

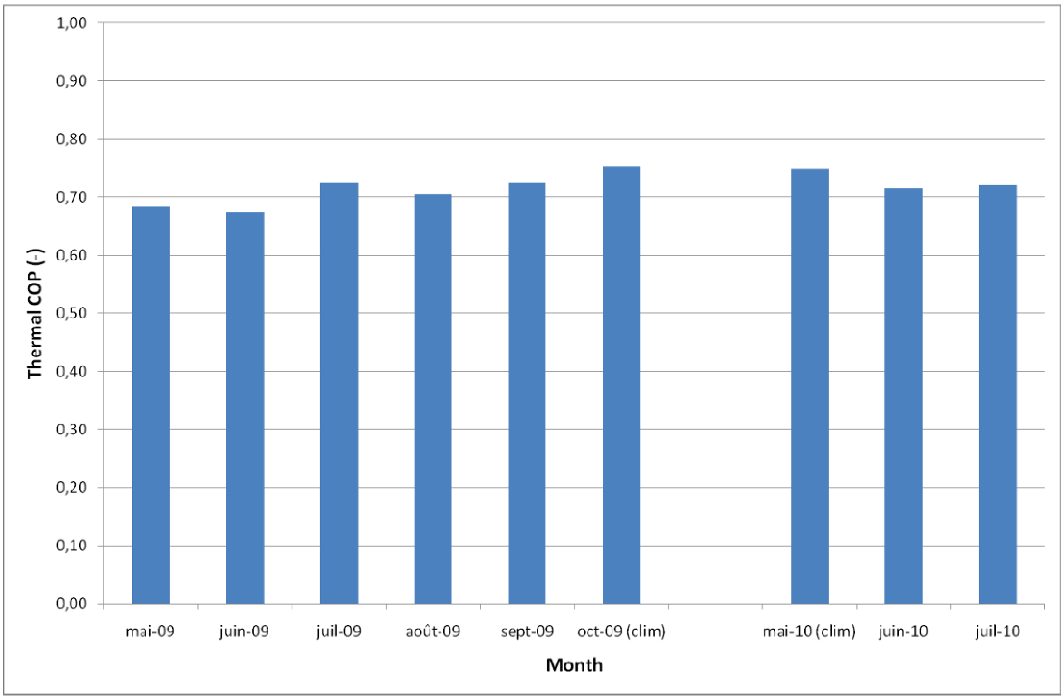


Figure 4: Thermal COP of the Chambery installation during the working period

So, during the cooling period, the thermal COP of the Rotartica chiller is quite excellent. Indeed, thermal COP reaches at least a value of 0,7 almost every month. Besides, this value is quite stable for the whole cooling period.

5.3 Electrical COP

The Electrical COP can be calculated month by month for the working period, and is showed in Table 4 and in Figure 5.

Table 4: Electrical COP of the Chambery installation during the working period

May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Jul
2009	**2009**	**2009**	**2009**	**2009**	**2009**	**2009**	**2009**	**2010**	**2010**	**2010**	**2010**	**2010**	**2010**	**2010**
1,94	**2,60**	**3,20**	**3,53**	**3,60**	**0,65**	**3,72**	**2,03**	**0**	**4,55**	**9,31**	**10,6**	**3,55**	**3,71**	**3,62**

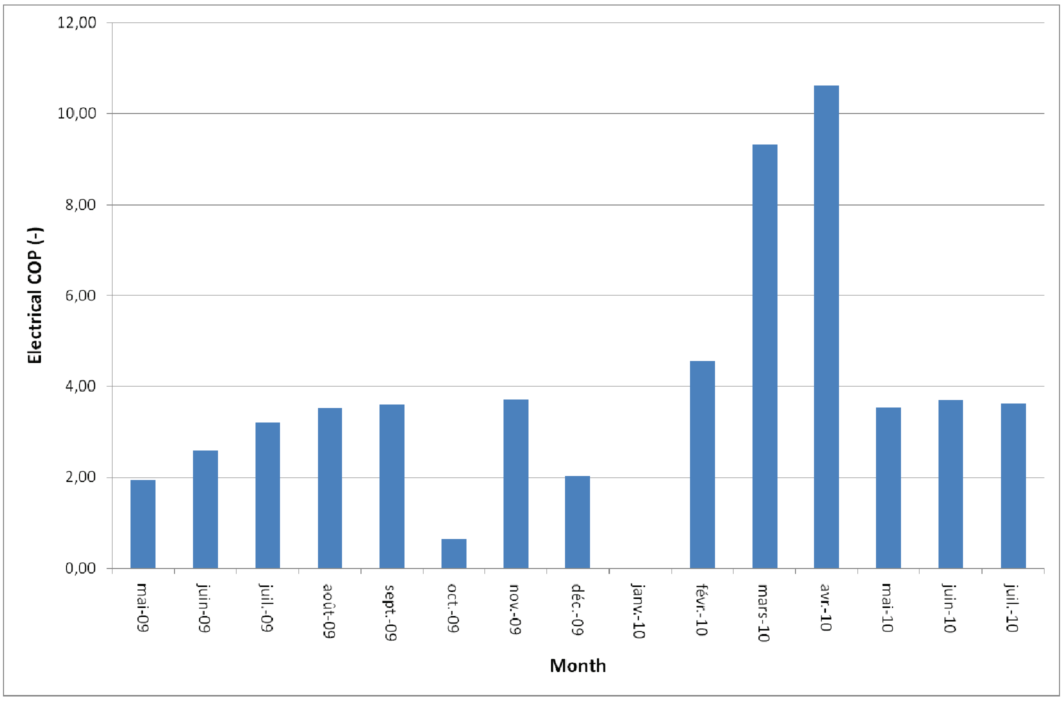


Figure 5: Electrical COP of the Chambery installation during the working period

Even with some very good month (March and April 2010), it appears that the overall average electrical COP for the working period is 3,14, which is not an extremely good result (even if it could have been worse). Indeed the objective of a solar cooling and heating installation is usually to reach an overall electrical COP of 5. In fact, an electrical COP around 3 could be almost reached by a conventional heat pump and the objective of a solar installation is to be far better than a conventional solution.

However, this low electrical COP can be at least partly explained by the design of the installation. Indeed, the installation, because of its "experimental state" has numerous devices consuming electricity. The number of pumps is high, the Clipsol's Blocsol device consumes electricity too, as well as the Rotartica which consumes more electricity than an "average" absorption chiller, and finally a not negligible part of the electricity consumption is due to the metrology devices (sensors more numerous than in a "basic" solar cooling installation).

5.4 Primary Energy Ratio (PER)

The overall efficiency of the installation in terms of primary energy was calculated using the PER (Performance Energy Ratio). It is the ratio between all the energies given to the user (heating, cooling, and DHW) and all the primary energies consumed by the system (including backups). This value is calculated monthly for the working period, and the results are shown in Table 5.

Table 5: Monthly Primary Energy Ratio for the Chambery installation

May **2009**	Jun **2009**	Jul **2009**	Aug **2009**	Sep **2009**	Oct **2009**	Nov **2009**	Dec **2009**
0,68	**0,94**	**1,14**	**1,29**	**1,18**	**0,38**	**-**	**0,73**

Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
0,77	0,90	2,05	3,68	1,22	1,45	1,43

Again the results are strongly related to the studied month. The Primary energy ratio can reach 3,68 in April 2010. But they also can be lower than 1.

An average based on the year from July 2009 to June 2010 was calculated, and it leads to a PER equals to 1,10. Once again, even if this value is higher than 1, it is not very high.

In addition to the large amount of electricity consumed by the different electrical devices (as explained in the previous paragraph 5.3), the backup is an electrical one. And the amount of primary energy to create one kWh of electrical energy is very high, as a consequence, the PER is lower than in the case where another type of backup is used.

5.5 Fractional solar heating and cooling savings

The global evaluation of the whole system was performed by using the fractional solar heating and cooling savings fsav,SHC performance criteria. It calculates the ratio of the fraction of energy savings of the solar system to energy consumption of a conventional system that fulfils the same demand for heating, cooling and domestic hot water. The calculation was achieved by considering the conventional system as using fossil electricity. This value is calculated monthly for the working period, and the results are shown in Table 6.

Table 6: Mothly fractional solar heating and cooling savings for the Chambery installation

May 2009	Jun 2009	Jul 2009	Aug 2009	Sep 2009	Oct 2009	Nov 2009	Dec 2009
-63,74%	-19,18%	2,14%	13,15%	4,73%	-195,8%		-12,99%

Jan 2010	Feb 2010	Mar 2010	Apr 2010	May 2010	Jun 2010	Jul 2010
-5,88%	9,12%	60,08%	77,70%	10,11%	22,83%	21,51%

The results are closely related to the studied month. The fractional solar heating and cooling savings can be high up to 77,7% (for April 2010), but they can also be negative.

An average value based on the year from July 2009 to June 2010 was calculated, and it leads to a fractional solar heating and cooling savings equals to 14,80%. Even if this value is positive, it is not very high, and the explanations are the same than in the previous paragraph (5.4): because of the “experimental state” of the system, a lot of devices consuming electricity are part of the installation design, and in addition an electrical boiler is used in the hot tank which is disadvantageous when the calculations are based on primary energy.

6 Conclusion

Then, thanks to a precise monitoring it was possible to obtain the necessary values to calculate these performance factors with the real experimental data.

After the calculation and analysis of this set of performance factors, the performance of the overall installation has been evaluated as well as the performance of some specific element of the system.

The main conclusions obtained regarding the performances analysis carried on are:

- ✓ Excellent performances of the Rotartica chiller.
- ✓ Good behaviour of the geothermal horizontal probes.
- ✓ Good performances of the collector field considering they are flat plate collectors, but a lot of days are not sunny enough to start the solar system.
- ✓ Difficulties to reach a high electrical COP due to the “experimental state” of the installation: explained by the presence of numerous pumps, of devices consuming more electricity than usually (hot tank, chiller), and of a large set of sensors (more numerous than in a “basic” solar cooling installation). The same conclusions are also valid for the Primary Energy Ratio and for the fractional solar heating and cooling savings.

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 7

Monitoring Results of Spain: Gymnasium of the University of Zaragoza, Zaragoza

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date:

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Contents

1 Background

The installation is located in Zaragoza (Spain) at the indoor sports centre of the University of Zaragoza and it is used to cool a gymnasium. This installation was designed as a consequence of the overheating in the existing solar collectors. In summer, the solar field was oversized because solar power was higher than needed.

2 System Design

- System scheme

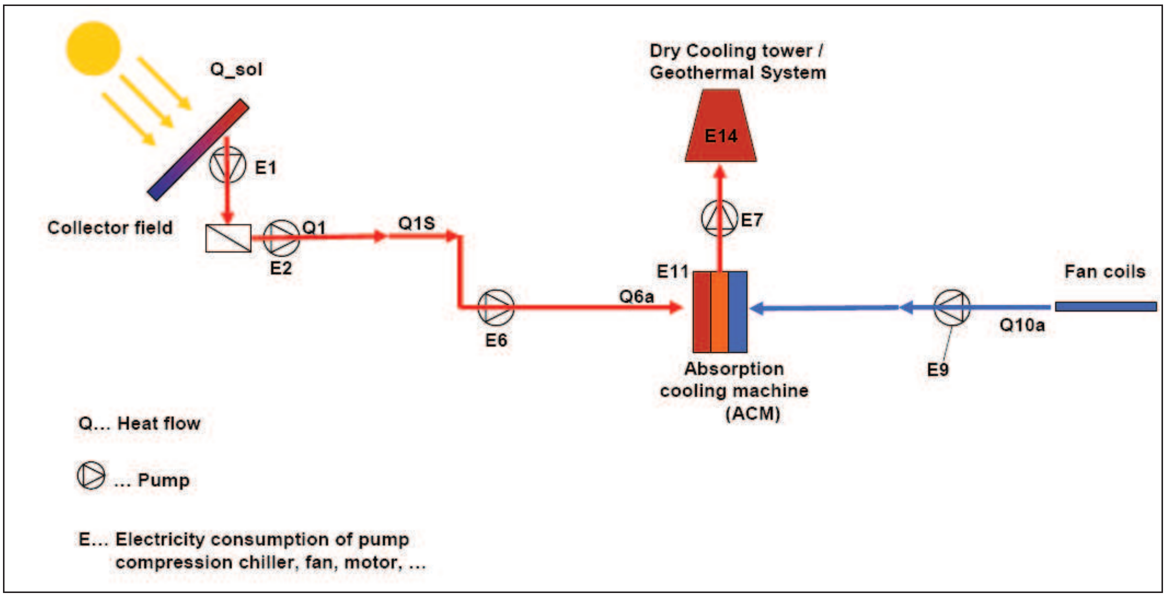

Fig. 1. – Solar Cooling System scheme

- Reasons for designing the system the way it has been designed

This solar cooling installation was designed as a consequence of the overheating problems of the existing solar filed used to contribute to the domestic hot water supply of the building. In the summer, to solve this problem and to use this solar waste energy, the chosen solution was the installation of an absorption chiller. Therefore, the solar collector field of the solar air-conditioning system has 37.5 m^2 of useful area. Problems encountered

- Photo documentation

		
Solar field	Machinery room	Gymnasium

		
Fan coil	Initial heat rejection system	Geothermal heat rejection system (2009)

3 Control Strategy

The solar pump of the system starts to work when the temperature of the solar field is around 86ºC, in this moment the water circulates in the primary loop. The secondary pump, which supplies thermal energy to the absorption chiller, is activated when the solar heat exchange temperature on this secondary side is over 80ºC.

The monitoring system is configured so that the absorption machine is turned on when the inlet temperature of the generator is 80ºC.

It has to be noticed that the installation contains a hot water storage tank and an auxiliary boiler as the back-up system. The aim of the hot water tank is to store the solar energy in those periods without cooling demand. And the mission of the gas boilers is to supply thermal energy to the absorption chiller when the solar energy isn't enough to produce chilled water. However, both devices have never been used. On the one hand the gymnasium always demands cooling demand when there is enough solar energy to activate the chiller, so it isn't necessary a buffer to store the heat water. On the other hand the installation is focused on the chiller performance when the chiller is only solar powered.

4 Monitoring Equipment

4.1 Installed Equipment

The installation is completely monitored. The monitoring system was designed perform the energy balances of the different components of the installation. There are a PLC unit and a web controller, which form the controlling and recording system. In this way, the outdoor and indoor conditions of the installation are well defined.

The monitoring system consists mainly of two temperature probes (near to the absorber and the condenser of the absorption machine) which measure the inlet and outlet temperature of the flow between the absorption chiller and the dry cooling tower (finned tube heat exchanger). Two temperature probes measure the inlet and outlet temperature of the flow between the absorption chiller and the fan coils (the measure is taken near the evaporator in the absorption machine).

Two temperature probes measure the inlet and outlet temperature of the flow between the absorption chiller and the heat exchanger (the measure is taken near to the generator in the absorption machine). While the temperature values for the exterior temperature are measured with NTC sensors, the ones located around the absorption chiller are registered with PTC sensors.

Furthermore, there is a flow meter in each one of the circuits of the chiller; the water flow that goes to the generator, the water flow that goes to the fan coils and the water flow that goes to the finned tube heat exchanger.

In figure 2 it can be seen the location of probe of the monitoring system.

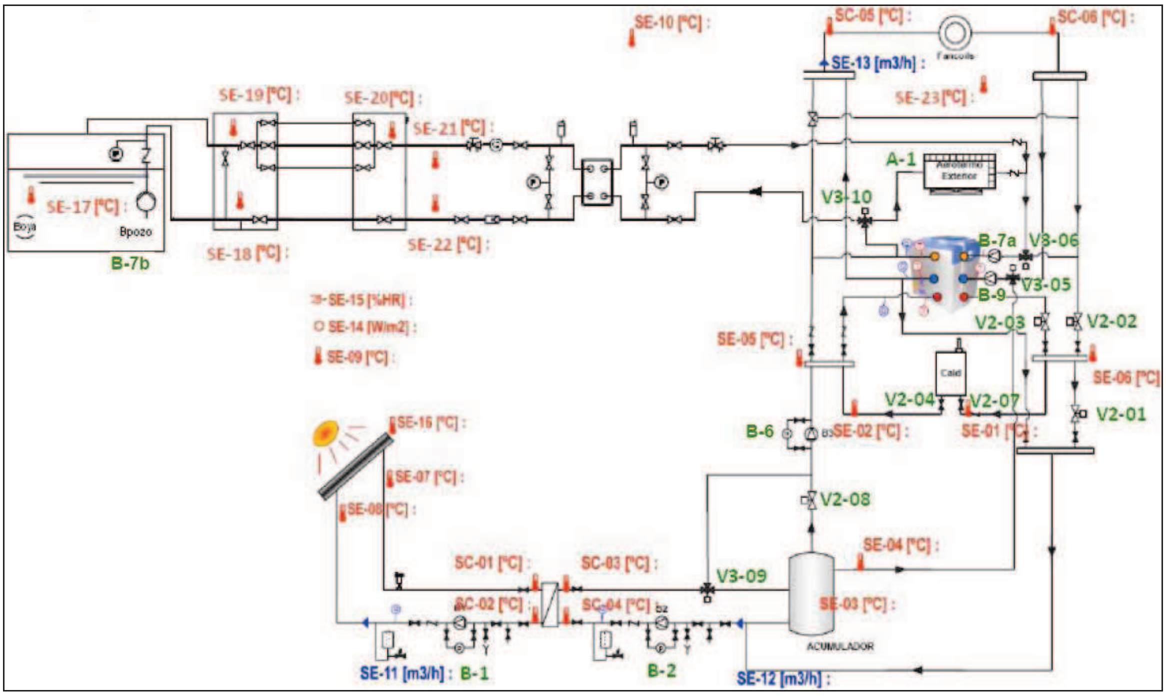

Fig. 2. – Monitoring Equipment

In Table 1 the features of the main sensors of the monitoring system are shown.

DEVICE	MEASUREMENT	UNITS	COMMERCIAL MODEL	RANGE	ACCURANCY
SC-01	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-02	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-03	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-04	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-05	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+18 0ºC	Máx. ±1,5%
SC-06	Temperature	℃	SIEMENS Ultraheat 2WR5(NTC)	+2ºC;+1 80ºC	Máx. ±1,5%
SE-01	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-02	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-03	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-04	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-05	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-06	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-07	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-08	Temperature	℃	SIEMENS QAD22	-30ºC;+130ºC	Máx. ±0,8 º%
SE-09	Temperature	℃	SIEMENS QFA 3160	0ºC;+50ºC	±0,8ºC (from 15ºC to 30ºC); ±1ºC (0ºC a 15ºC; 30ºC a 50ºC)
SE-10	Temperature	℃	SIEMENS QAC22	-35ºC;+50ºC	±0,05ºC in 0ºC (DIN 43760)
SE-11	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. ±4%
SE-12	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. 4%.
SE-13	Flowmeter	m^3/h	SIEMENS Ultraheat 2WR5	15l/h-3m^3/h	Máx. ± 4%
SE-14	Solar Radiation	W/m^2	QUIMISUR IQ-5.0	0-2000W/m^2	±2%
SE-15	Relative Humidity	%HR	SIEMENS QFA 3160	0-100% h.r.	±2%h.r.
SE-16	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-17	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-18	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-19	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-20	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-21	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-22	Temperature	℃	SIEMENS QAE2120.010	-30ºC;+130ºC	Máx . ±0,8º%
SE-23	Temperature	℃	SIEMENS QAC22	-35ºC;+50ºC	±0,05ºC en 0ºC (DIN 43760)

Table 1.- Features of the installed equipment

4.2 Period of Measurement

The solar cooling system has been measured from June to September since 2007.

5 Monitoring Results

5.1 Annual / Monthly Data

Table 2 shows the average experimental results of the chiller analysis in the year 2007 and 2008, operating in the steady state period.

Year	W_{ch} (kW)	W_c (kW)	W_g (kW)	COP	T_{dbo} (ºC)
2007	5.78	9.7	15.4	0.57	27.7
2008	4.4	8,0	12.5	0.51	31.2

Table 2.- Experimental mean values of the installation in the years 2007 and 2008

5.2 Analysis of Typical Days

In Fig. 3 it is shown the chiller operation temperatures for one day (11/07/2008). The operation in this day can be considered as representative for the chiller performance. The daily operation process is described as follows.

At 11:30 due to the solar field temperature, the pump of the secondary circuit starts to pump water heated in the solar heat exchanger to the generator of the chiller. As soon as the temperature of this water flow overcomes 80 ºC at the generator inlet, the chiller begins to produce chilled water.

The temperature difference in the heat driven of the absorption cycle is up to 7 ºC between the inlet and the outlet of the generator. The average generator power for this day is 7.9 kW. This generator power makes the evaporator outlet temperature ($T_{ev,o}$) decrease, reaching 12 ºC on this day. When the climatic conditions are the optimal ones, these values can decrease to 9 ºC. The temperature difference in the evaporator is around 2 - 3ºC, achieving a chilling capacity of 5.8 kW in the last part of the 2007 and 4.4 kW in 2008. According to the heat rejection system, the inlet temperature ($T_{he,i}$) increases when the absorption chiller operates, being its maximum value 42 ºC, coinciding with the maximum values of $T_{g,i}$.

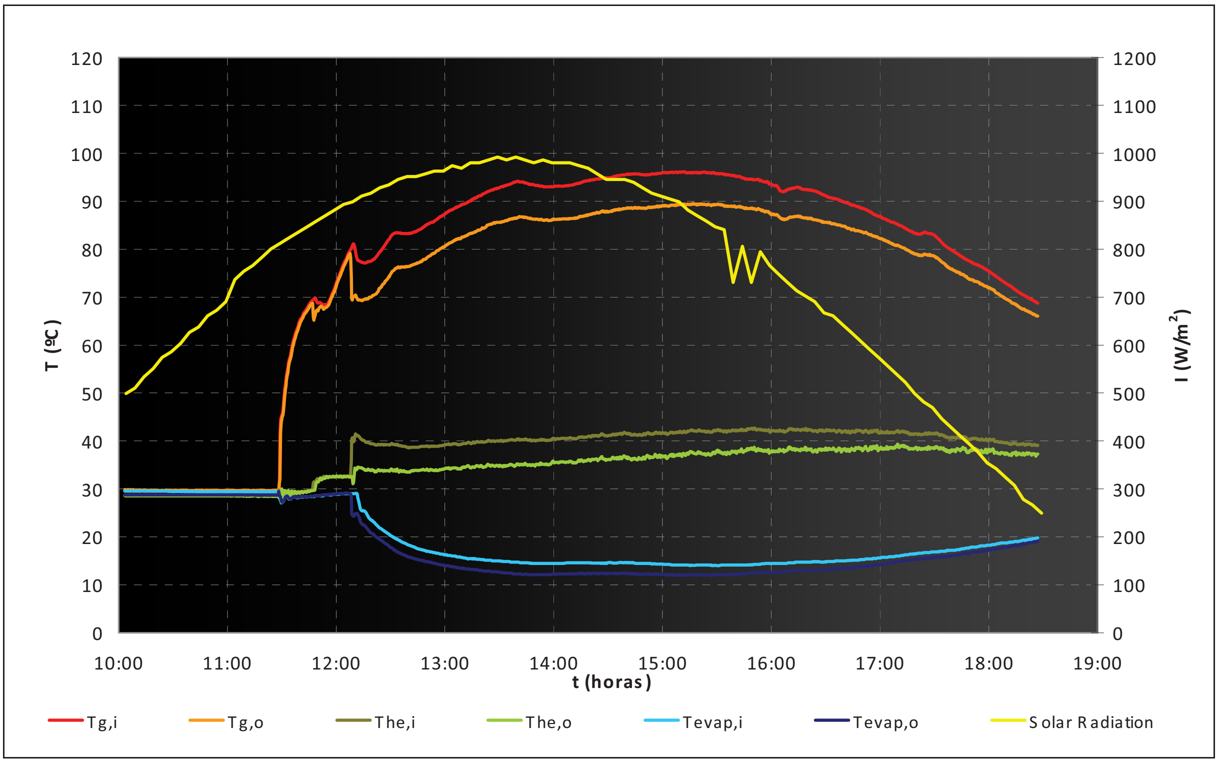

Fig. 3. - Operation temperatures of the chiller and solar radiation in 11/07/2008.

Because of the fact that a dry cooling tower is used to reject the waste heat of the absorption chiller, in figure 3 it can be seen that the heat rejection temperatures depend on the outdoor temperature. The influence of the ambient temperature can be seen in the Table 3 too. The average values of the most important parameters of the chiller are presented for three representative days in this table. It can be seen that the average generator power is similar during the three days, but the deviations between the chilling capacities differ up to 19%. This happens because at high values of the outdoor temperature, the capacity of the dry cooler to reject the heat of the absorption cycle to the ambient decreases. Therefore, the higher the outdoor temperature is, the lower the chilling capacity and the COP produced by the absorption chiller are.

Date	T_{dbo} (ºC)	$T_{g,i}$ (ºC)	$T_{ev,o}$ (ºC)	W_{ch} (kW)	W_c (kW)	W_g (kW)	***COP***
11/07/2008	35	94	12.4	4.02	12	7.9	0.48
29/07/2008	30.4	92.2	11.6	4.5	12.8	8.2	0.53
08/08/2008	29.4	91.9	9.5	4.8	13.3	8.7	0.53

Table 3.- Average values of three typical days.

5.3 Detailed Analysis

The steady state performance of the chiller was analyzed. In this way figure 4 shows the influence of the outdoor temperature on the COP during 2007 and 2008 when the chiller operated in the steady state period. The COP decreases when the sink temperature increases. Both trends have the same slope, although the mean values of the year 2008 are lower because the mean ambient temperature during 2008 was higher.

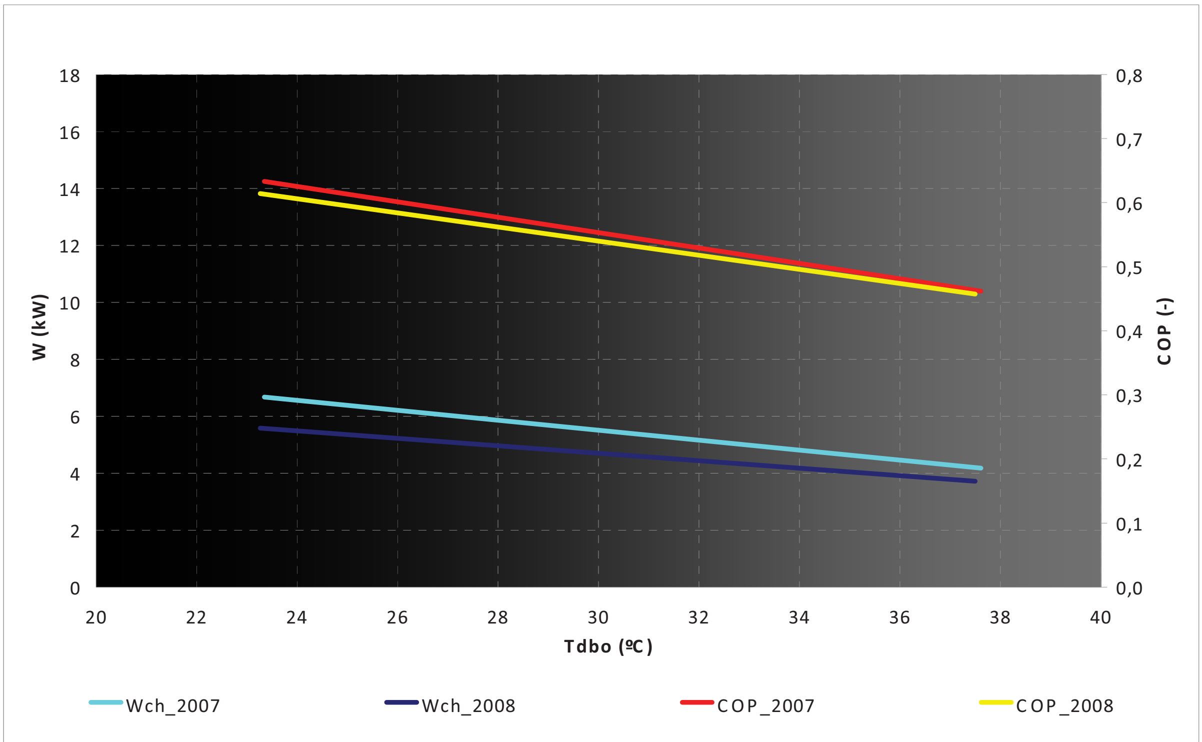

Fig. 4. – Influence of the outdoor temperature on the COP and in the chilling capacity

6 Experiences / Lessons Learned

As it was mentioned, during this first two years, the installation worked with a dry cooler tower. The studies showed the great influence of the temperature of the heat rejection sink on the machine performance. Therefore, the substitution of the initial heat rejection system was proposed in order to improve the performance of the absorption chiller. Finally, it was decided to use a geothermal system using a water well placed in the surroundings of the solar cooling installation. With this modification the chiller operated with a constant temperature in the heat rejection sink. This temperature, according to others water wells placed near by was around 17 ºC. This water well supply water to a 25 m^3 water tank, in which the heat produced in the absorption cycle, is rejected. The tank is usually used to irrigate the sport grounds placed in the surroundings of the solar cooling installation in summer. This use means the contained water of the tank is renewed every day, so, daily, the water temperature kept daily a constant value, resolving the possible problems of thermal saturation in the tank.

Another feature of the geothermal system is the following. The overall length of the new circuit is 190.5 m, of which 90.5 m was divided into three pipes with a diameter smaller than the rest of the circuit. This has been done in order to increase the heat exchange surface between the pipes and the ground. Hence the rejection of the heat generated by the absorption machine will take place in two places: the water tank and the geothermal horizontal exchanger. Besides this, the initial heat rejection sink, the dry cooling tower hasn't been removed from the installation, so that the solar cooling has a hybrid heat rejection system.

In order to estimate the performance of the chiller working with the geothermal sink a TRNSYS model was developed. Previously the model of the initial installation was created and validated with the experimental results taken from the monitoring system.

In 2009 the new heat rejection system started to work. Unfortunately the improvement of the chiller performance was partial. Although the influence of the ambient temperature was removed, due to the operational temperature of the water tank was 25 ºC instead of 17 ºC, the mean values of the chiller capacities were not the expected ones (Figure 5).

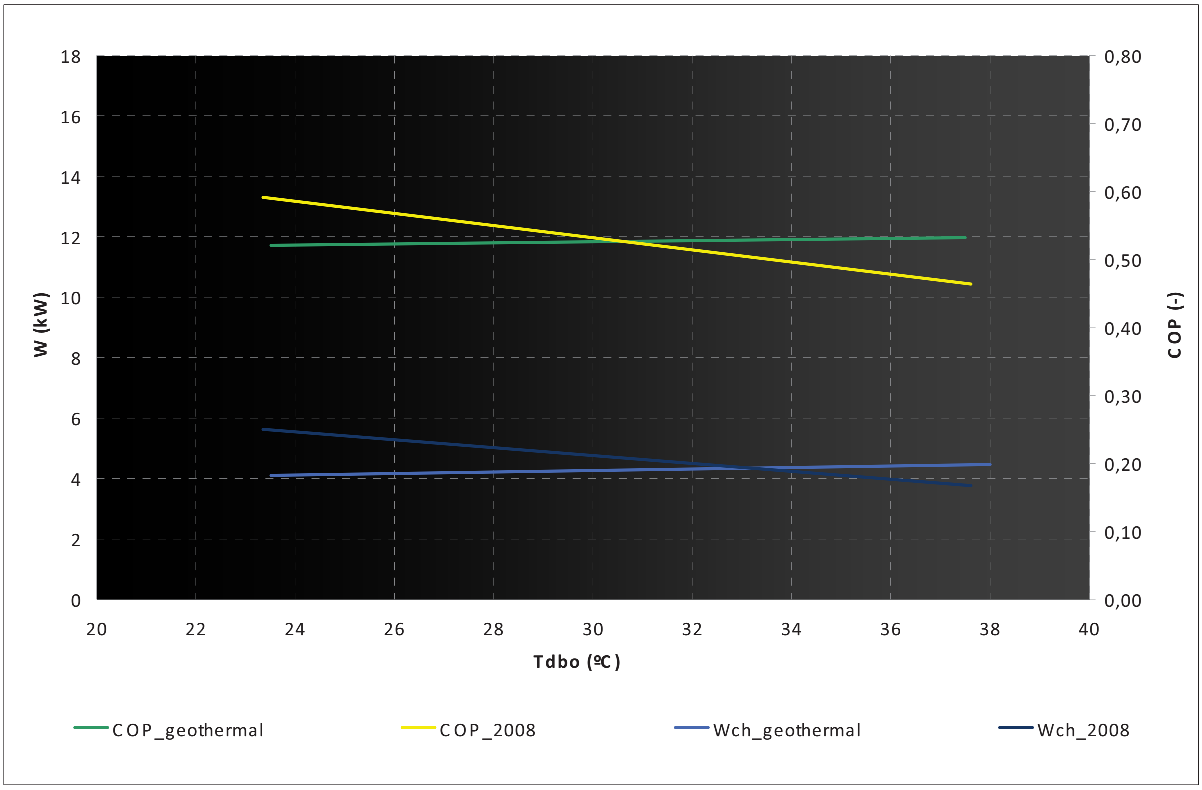

Fig. 5. – Comparison of the COP and the chilling capacity

7 Conclusions

The analyses of the solar cooling installation placed in the University of Zaragoza have allowed enlarging the knowledge of this kind of solar thermal systems.

The absorption chiller was installed in order to use the solar waste energy in summer time. Before its installation, the solar thermal system suffered overheating problems in this period.

Besides that the users of the gymnasium are satisfactory because the ambient conditions in the gym have been improved.

From the initial configuration of the solar installation, the system has been modified in order to increase its energy efficiency.

8 Bibliography

- Monné, C.; Alonso, S.; Palacín, F.; Serra, L., (2011). ***Monitoring and simulation of an existing solar powered absorption cooling system in Zaragoza (Spain).*** Applied Thermal Engineering, 31, 28-35. doi:10.1016/j.applthermaleng.2010.08.002.
- Monné, C.; Alonso, S.; Palacín, F., (2011). ***Experiencia en una instalación de refrigeración por absorción con energía solar***. Información tecnológica, 22 (3).
- Monné, C.; Palacín, F.; Alonso, S., (2011). ***Analyses of a new dissipation system for a solar cooling Installation in Zaragoza (Spain***). Journal of thermodynamics.
- Monné, C.; Alonso, S.; Guallar, J.; Palacín, F., (2011). ***Stationary analysis of a solar LiBr-H_2O absorption refrigeration system.*** International Journal of Refrigeration. Article in Press. doi: 10.1016/j.ijrefrig.2010.11.009.
- Palacín, F.; Monné, C.; Alonso, S., (2010***). Evaluation, diagnosis and improvement of a solar cooling plant by means of experimental analysis and dynamic simulation***. 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S.; Izquierdo, J. M.; Heredero, J., (2010). ***Design and operational experiences of an alternative heat rejection sink for an existing absorption solar cooling system.*** 8th International Congress on Solar Heating, Cooling and Buildings. Eurosun 2010. 28th September to 1st October 2010. Graz (Austria).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Experiences on heat rejection sinks of a solar absorption cooling system.*** 3rd International Conference on Passive and Low Energy Cooling for the Built Environment (PALENC 2010), 5th European Conference on Energy Performance & Indoor Climate in Buildings (EPIC 2010) and 1st Cool Roofs Conference. 30th September to 2nd October 2009. Rhodas Island (Greece).
- Palacín, F.; Monné, C.; Alonso, S., (2010). ***Comparing different sinks of heat rejection of an existing solar powered absorption cooling system.*** International conference on renewable energies and power quality (ICREPQ'10). 25th to 26th March 2010. Granada (Spain).
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Instalación experimental de refrigeración solar - primeros resultados.*** XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th to 21th June 2008. Vigo (Spain). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte I)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th June 2008. Vigo (Spain). SPE.
- Monné, C.; Guallar, J.; Alonso, S.; Palacín, F., (2008). ***Influencia de la velocidad de giro en las máquinas rotativas de absorción (parte II)***. XIV Congreso Ibérico y IX Congreso Iberoamericano de Energía Solar. 17th al 21th June 2008. Vigo (Spain). SPE.

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 8
Monitoring Results of Technical College Butzbach

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 16.12.2010

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Contents

1 Background

The low-energy building of the Technical College in Butzbach has a demand for summer air-conditioning due to high occupation rates and the frequent use of computational equipment. Besides the regular school days the building is occupied throughout the summer season intensively as well. This fact was determining for the promotion of this project within the frame of the German Solarthermie 2000plus funding programme.

Two ventilation systems with heat recovery and a 1,250 m³/h air volume flow rate each were already installed in the building. These systems were not sufficient in order to remove the sensible and latent cooling loads in summer. Therefore, a solar autonomous chilling plant was installed which consists of two absorption chillers of the type Suninverse from Sonnenklima, Berlin. Each of the chillers has a nominal chilling capacity of 10 kW. They are driven exclusively by solar heat from a collector field of 60 m² aperture area. It consists of evacuated tube collectors of the type CPC Star azzurro, Paradigma. In addition, the ventilation units were extended by cooling coils. Furthermore, chilled ceilings and a cooling shaft were installed. The area which is air-conditioned comprises 335 m².

2 System Design

The Fraunhofer ISE supported the preliminary planning by carrying out simplified system simulations. With the help of the simulation results estimations could be given for the dimensioning of the system. The variation of the building simulation, which considers an external shading device, leads to estimations on the peak cooling loads (sensible and latent) of more than 20 kW. The sensible cooling loads often showed daily peaks of about 15 to 20 kW. As the type of chiller was almost already decided (Suninverse, 10 kW nominal chilling capacity) the results supported the installation of two absorption chillers.

A simplified scheme of the complete air-conditioning plant of the Technical College in Butzbach can be seen in figure 1.

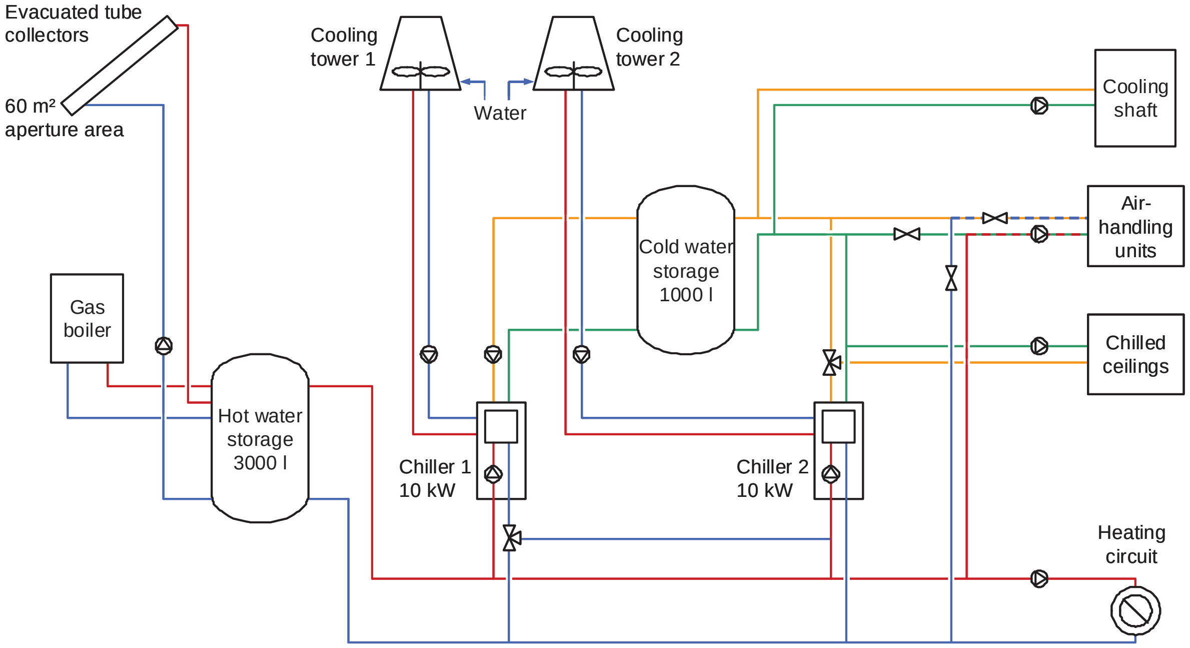


Figure 1: Simplified scheme of the heating and cooling system in the low energy building of the Technical College Butzbach. Source: Fraunhofer ISE

The hydraulic connection permits a single or a simultaneous operation of the chillers. Both chillers can deliver chilled water to the cold distribution systems. In the system concept it is planned that for high cooling demands in summer one of the chillers provides chilled water on a low temperature level for the dehumidification of the supply air. The second chiller supplies chilled ceilings on a higher cold water temperature level. This chiller cannot charge the cold water buffer storage. The last mentioned chiller could not be operated in the cooling season of 2009 due to a defect. Therefore, only operating results from one chiller can be provided for 2009. Since the end of June 2010 the second chiller could also start its operation.

The whole heat production and distribution system is one continuous water circuit. There is no heat exchanger integrated into the loop. This means also that the collector is a pure water system. In times where anti-freeze protection is necessary heat from the hot water storage is returned to the collector in short pump intervals.

Both, the solar collectors as well as the gas boiler can charge the hot water buffer storage. However, the boiler is switched off in the cooling period. By doing so, it can be assured that only solar thermal energy is used for the cooling of the building.

Figure 2: Cold distribution systems in the seminar rooms. Top: chilled ceiling elements; right: cooling shaft (without cover) for silent cooling and air dehumidification. Bottom: one of two air handling units with heat recovery and supply air cooling and dehumidification. The air handling unit in the seminar room was insulated additionally against acoustic noise. Source: Fraunhofer ISE

Figure 3: Evacuated tube collector field at the main building of the Technical College Butzbach. Also an effective shading of the rooms can be achieved by the assembly of the collectors. Source: Fraunhofer ISE

3 Control Strategy

The control strategy of the system was realized by one of the project partners (Hindenburg Consulting). Information on the strategy is not at hand.

The collector sub-control system for freezing protection runs independently from the overall system control and was prepared by the collector manufacturer Paradigma.

4 Monitoring Equipment

4.1 Installed Equipment

For the system monitoring different measurement devices have been installed. Beside several temperature and volume flow meters also the specific radiant power is measured by the help of two irradiation pyranometers (CMP11); one in horizontal position and the other one in the collector plane. The pressure in the collector circuit is measured as well. Ten electricity meters record the electrical power input of pumps, absorption chillers, cooling towers, air handling units and the controller (three-phase electricity counters with 1000 impulses per kWh). The operating hours from most pumps and some valve engines are registered as well.

The devices to measure the volume flow rates are multiple-jet impeller water meters with impulse transmitter. For measuring the temperatures in the system Pt100 temperature sensors (accuracy 1/3 DIN class B) were installed.

The following figure 4 shows all measurement points in the system.

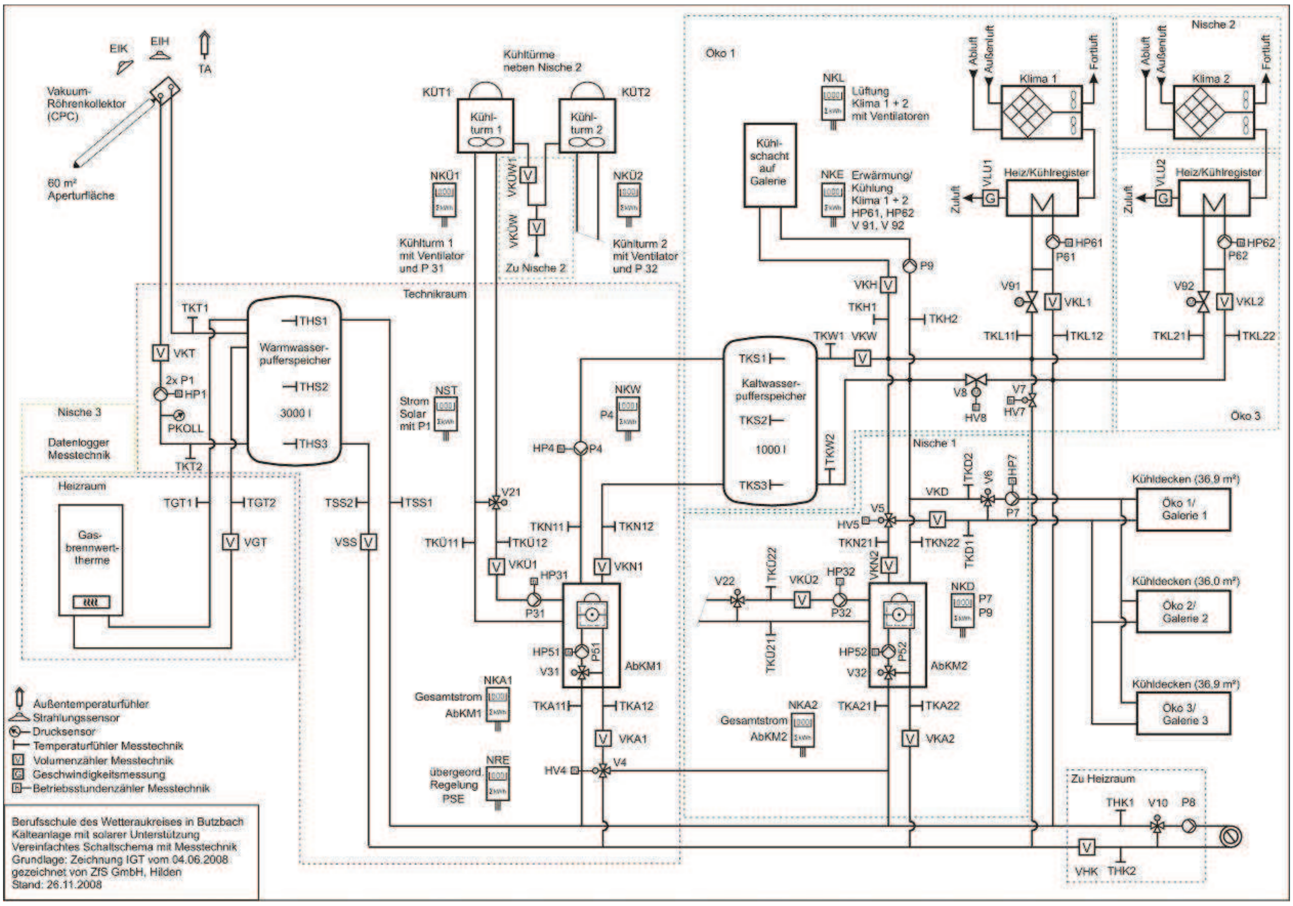


Figure 4: Scheme of the solar cooling plant with the measurement points. Source: ZfS GmbH

4.2 Period of Measurement

The monitoring system started its operation in December 18th 2008. Since then, the Fraunhofer ISE receives the data (one minute values) each day from the project partner ZfS. Since the beginning of the data monitoring there did not occur any complete or severe data loss. Just some measurement points needed to be adjusted at the beginning.

The Fraunhofer ISE will receive the data presumably until the end of 2011.

5 Monitoring Results

5.1 Annual / Monthly Data

As figure 5 shows the specific collector yields as well as the collector efficiency tend to be slightly higher in 2010 than the monthly values in 2009. A reason for this tendency can be a reduced shading of the collector in 2010 compared to 2009. In 2009 a scaffold shaded the collector.

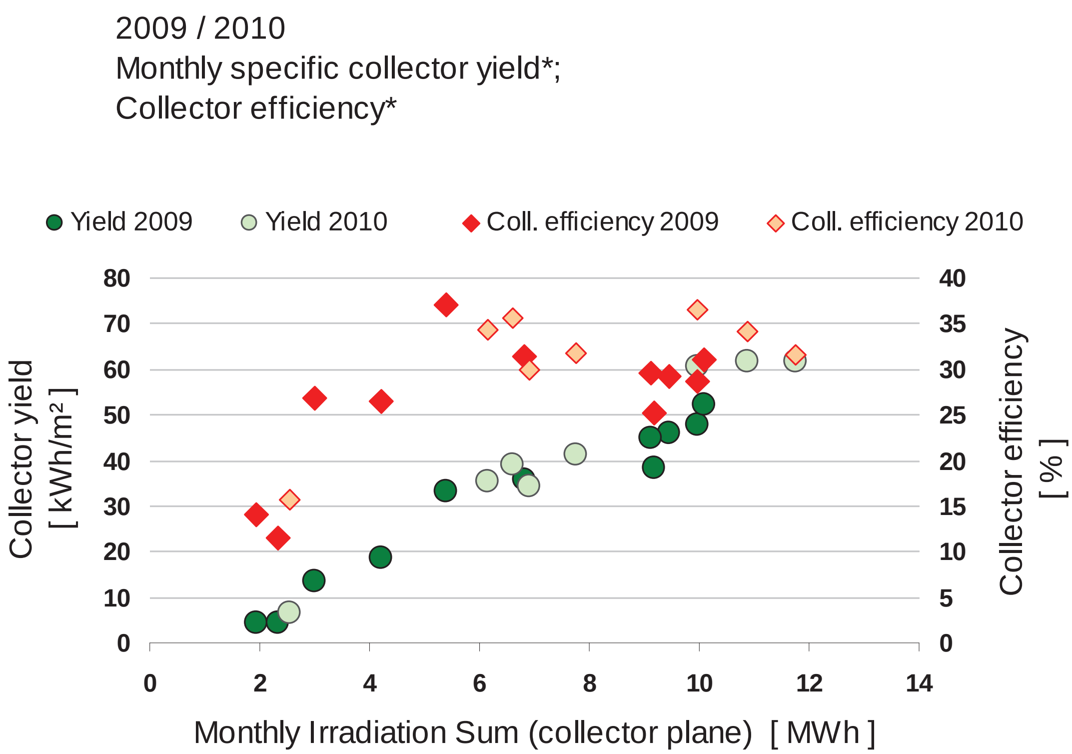


Figure 5: Monthly values of the specific collector yield and the collector efficiency in 2009 and 2010 (up to and including October 2010) plotted against the monthly irradiation sum on the collector plane. The values refer to the heat which the collector supplied to the storage; * includes returned heat to collector; no storage losses considered. Source: Fraunhofer ISE

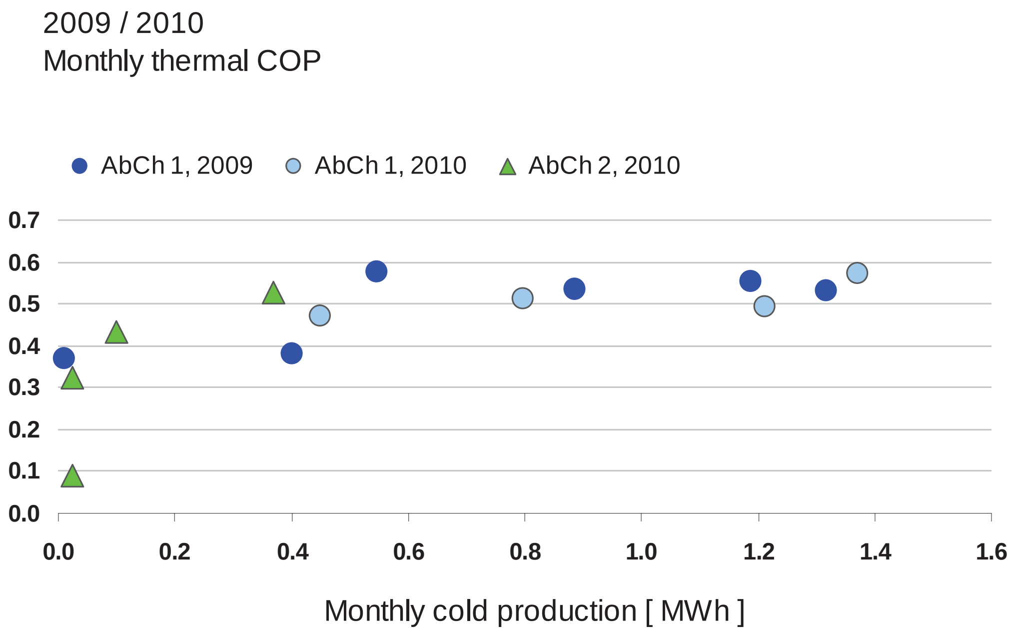


Figure 6: Monthly values of the thermal COP of the absorption chillers plotted against the monthly produced cold in 2009 and until September 2010. The cold which was produced by the absorption chiller 2 in June and September 2010 was very low. In June the chiller 2 started its proper operation at the end of the month and in September there was only little cooling load. Source: Fraunhofer ISE

2009	Collector efficiency* [%]	Share of solar heat returned to collector [%]	Share of solar heat on total heat input*** [%]	COP thermal absorption chiller 1 [-]	COP electric absorption chiller 1 [-]
January	11.5	57.4	4.1	---	---
February	26.9	20.6	16.5	---	---
March	37.0	6.7	48.7	---	---
April	29.2	9.4	100	---	---
May	29.7	5.0	92.8	0.39	2.9
June	25.2	7.1	100	0.55	4.6
July	28.7	4.3	100	0.56	5.4
August	31.1	3.5	100	0.53	5.0
September	31.5	4.7	100	0.58	5.6
October	26.5	12.6	29.8	0.37	2.4
November	14.0	39.5	5.1	---	---
December	< 0 **	165.2	< 0 **	---	---
Annual	27.7	10.8	40.6	0.53	4.7

* includes returned heat to collector; no storage losses considered

** collector yields slightly below heat for freezing protection

*** corrections in winter due to measurement error in heat input from gas boiler

Table 1: Monthly and annual balances for some system operation data of 2009. In the heating season when the gas boiler is in operation the share of solar heat is based on an estimate because the heat input of the boiler into the storage was measured incorrectly due to pollution in the hydraulic circuit. The values were adjusted by fixing the monthly hot water storage losses at 10% in winter. Source: Fraunhofer ISE

5.2 Analysis of Typical Days

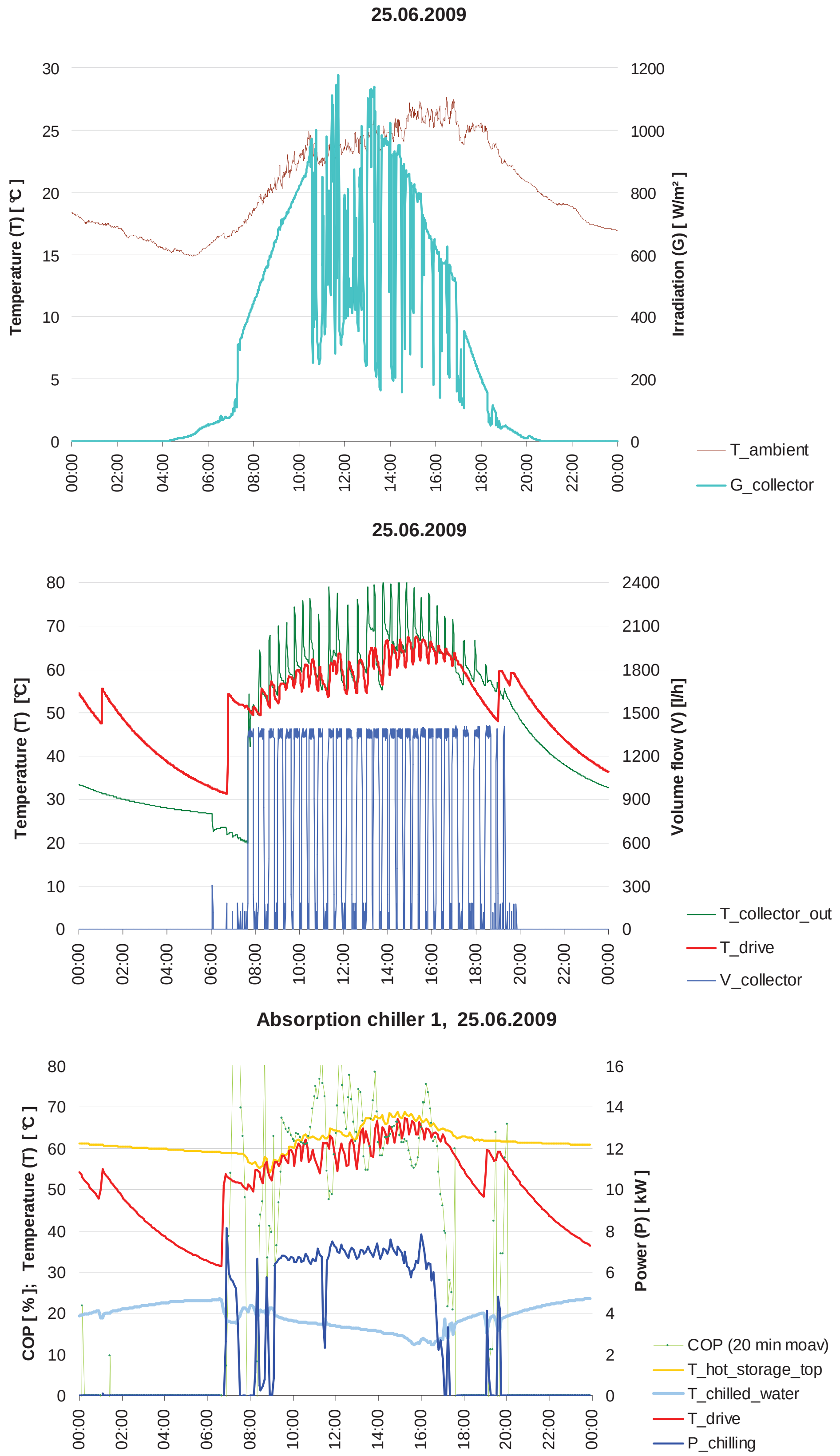


Figure 7: From the irradiation on the collector plane up to the cold production: Cooling with the absorption chiller 1 on June 25th 2009. (Values based on one minute average; just the thermal COP is based on the moving average of 20 minutes for a better visibility.) Source: Fraunhofer ISE

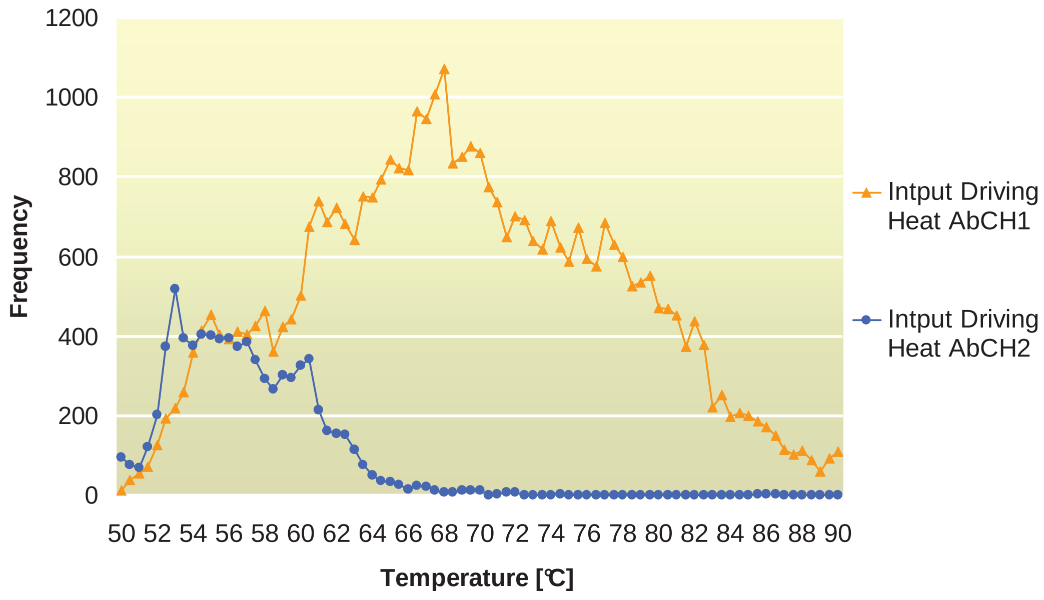
Often it can be observed that the chillers run continuously for several hours thereby taking advantage of the big range in the driving temperature. Figure 7 shows an exemplary day with cooling operation of absorption chiller 1. In contrast to the relatively steady temperature in the upper part of the storage the driving temperature to the chiller oscillates periodically in a stronger manner. This oscillations correlate exactly with the collector outlet temperature and the periodical pump operation in the collector circuit ("bucket principle"). It is still not clear why the driving temperature is evened by the storage just on a small scale. The measuring point of the upper storage temperature shows clearly a temperature course which is more straightened.

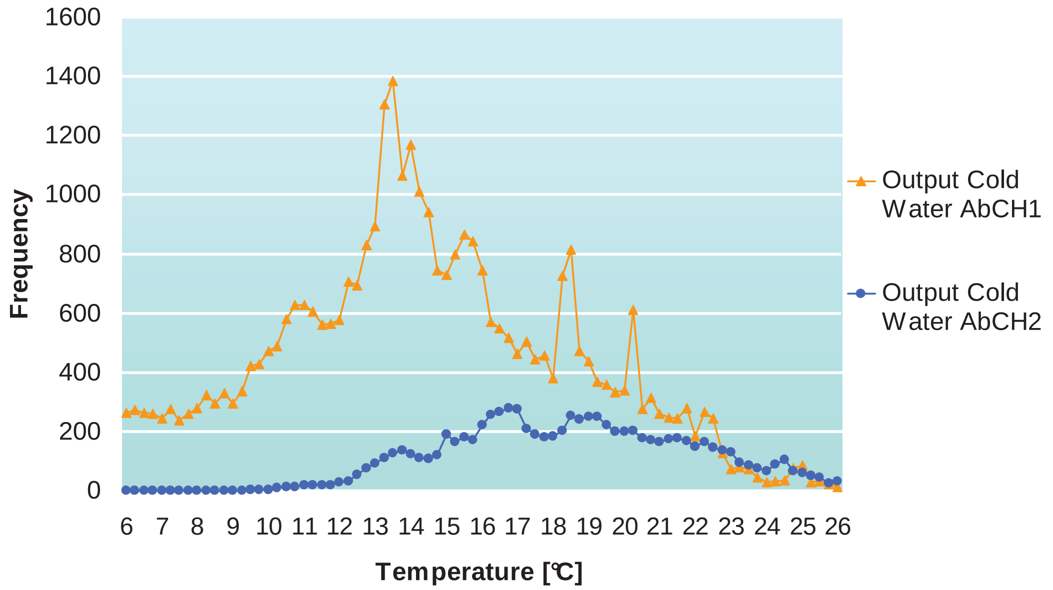
In contrast to the pump in the collector circuit the pump in the driving circuit of the chiller runs continuously from 9:00 until 17:00 with a short interruption before 12:00. Altogether with the low driving temperature chilling capacities between 6 and 8 kW are achieved. According to the temperature and capacity oscillations in the hot water circuit of the chiller the thermal COP varies strongly as well.

5.3 Detailed Analysis

Figure 8 shows the frequency distribution of driving, cooling water and chilled water temperatures of both chillers in serial connection in 2010. The frequency maximum for the driving temperature of chiller 1 lies between 65℃ and 70℃. Temperatures greater than 83℃ appear rarely. Chiller 1 produces most frequently chilled water temperatures in the range of 13℃ to 15℃. The chilled water range of chiller 2 is widely distributed and on a higher temperature level as well. The driving temperature level of chiller 2 is shifted about 10 K downwards. The maxima of the cooling water temperatures are relatively clear. However, the values differ with 4 K from each other.

The frequency distributions show just values when the pumps in the three circuits are in operation. Though, the first operation minutes after the chiller starts can for example show higher chilled water temperatures until a steady operation is reached. However, these values are not filtered out from the shown graphics.





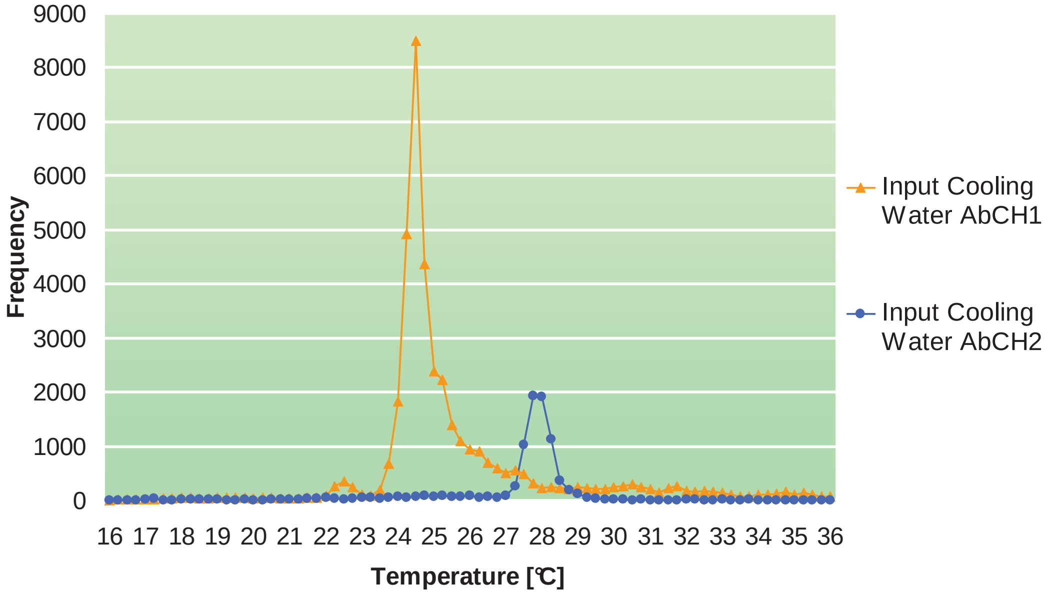


Figure 8: Frequency of the operation temperature ranges of both chillers in 2010 at serial connection in the driving circuit. (Values based on one minute average.) Source: Fraunhofer ISE

6 Experiences / Lessons Learned

As the collector is a pure water system it is necessary to return heat from the storage to the collector for the anti-freeze protection. Therefore, the pump runs in short intervals (bucket principle). The data analysis on an annual base showed that in one year about 10% of the collector heat gain is returned to the collector. In these 10% it is not only included the heat for freezing protection but also the losses which appear during the normal collector pump starts (cold water in the pipes). It should be kept in mind that in normal collector monitoring these start losses are not recorded. Rough estimated: approximately 1/2 to 2/3 of the 10% heat losses may be assigned to freezing protection.

As the plant is a pure water system no heat exchanger is installed. Consequently, the collector water is identical with the water in the heating system and driving circuit of the chillers. Thus, small particles from fabrication, e.g. swarfs, circulate through the whole hot water circuits and may cause pollution of filters which can lead to measurement errors. Actually, the flow meter in the gas boiler circuit was not working accurately due to the problem mentioned above. For avoiding this kind of problem a careful flushing of the total circuit after the installation is recommended.

The entry of small particles through the open wet cooling towers caused the pollution of the cooling water circuit. This led to a serious problem as the volume flow rates were much lower than normally and consequently too high cooling water temperatures entered the chiller. The solution of the problem was reached by the installation of an automated filter backwashing system, which starts every 16 minutes, and by a daily elutriation of the cooling towers. This circumstance should be considered by the installation of open cooling towers in areas with a high density of plants.

At the beginning, one of the two absorption chillers was not working due to a vacuum leakage. As an insolvency procedure of the chiller manufacturer was in progress in 2009 it took one year to solve the problem in the respective chiller. However, a general recommendation in order to avoid such problems cannot be given.

A special characteristic of the suninverse absorption chiller is the great flexibility in terms of driving temperature range. After the start phase of the chiller the chilled water production can be kept up until the driving temperatures drops slightly below 60℃. Since the second chiller was repaired and thus both chillers could be operated, the driving circuits were connected in series in order to test the above mentioned characteristic. The advantage of the serial connection is an increase in the temperature spread for the collector system. The experience that could be made with this connection variation was consistently positive. The first absorption chiller produces chilled water for supply air cooling and dehumidification, the second chiller supplies preferentially chilled ceilings and a cooling shaft on a higher chilled water temperature level. The operation of the chillers is stable; there are no start and stop phases and the operation of the chillers can be kept up continuously for several hours. Anyway, the common parallel connection of the chillers' generators with the hot water storage is also possible by switching a valve.

The acceptance of the solar autonomous air-conditioning during summer by the occupants (teachers, students) is big and it is stated as fully sufficient.

Furthermore, there is no extra input of fossil fuel for the chillers' operation necessary in order to overcome short periods with room air states outside the comfort range.

7 Conclusions

The plant for solar autonomous air-conditioning in summer in the low energy building of the Technical College Butzbach is built up relatively complex. In this case the reason lies in the testing of different connection variations in the driving and chilled water circuit, also for educational purposes.

The intended system operation was not possible due to the malfunction of one chiller in 2009 and the big delay of the reparation due to the insolvency process of the chiller manufacturing company. Further disturbances of the system's operation were caused by increased shading of the collector as a result of construction measures and pollution problems in the cooling water circuit. It is evident that the heat rejection is still a weak link in the component chain and it deserves closer attention also from the manufacturer's part.

Beyond that, the operation of the collector, of the absorption chiller 1 and the other hydraulic components has been very reliable. No damage results from collector stagnations. In spite of the above mentioned disturbances the occupants affirmed their satisfaction with the system at the final project meeting in September 2010. They also pointed out that the room air states in the seminar rooms improved significantly. The chosen concept, especially with respect to the autonomous solar thermally driven cooling, can be applied very well to the conditions of the cooling loads in the building.

The system serves at the same time as support in the education of students from the Technical College. This results in a big multiplier effect. In case the system should be transferred to a similar application it can be simplified considerably, as for example by installing just one chiller (for appropriate cooling loads) and thereby reducing respectively the hydraulic complexity. On the hot water part, the system is already simplified by being a pure water system and therefore not needing a heat exchanger in the collector circuit.

In the following points optimization potential can be given or a further revision is reasonable:

- possibly testing of the chiller operation with higher driving temperatures;
- a further examination of the second chiller’s capacity in the coming cooling season;
- a further examination of the backwashing intervals in the cooling water circuit: at present a high fresh water demand is required. There might be found an optimum between water consumption, backwashing intervals and pollution degree in the cooling water circuit.
- checking if a careful pruning of the trees shading the collector is possible.

Estimations show that in comparison to conventional heating and cooling systems primary energy savings and CO_2 emission avoidance was achieved.

8 Bibliography

E. Wiemken, J. W. Wewiór, A. R. Petry Elias: Betriebserfahrungen aus solar unterstützter Kaltwassererzeugung zu Gebäudeklimatisierung. Tagungsbeitrag zum 20. Symposium Thermische Solarenergie, 5. – 7. Mai 2010, Bad Staffelstein

E. Wiemken, J. W. Wewiór, A. R. Petry Elias, B. Nienborg, L. Koch: Performance and Perspectives of Solar Cooling. Tagungsbeitrag zur EuroSun2010, 28. September – 1. Oktober 2010, Graz, Österreich

SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 9

Monitoring Results of Germany: ZAE Bayern Office Building, Garching

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 20.12.2010

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Contents

1 Background

In solar thermal installations, both solar cooling and solar heating can be provided synergistically, yielding a complete annual utilization. During the cold season, solar heat serves for space heating. During the warm season, solar heat can be converted into useful cold by means of sorption cooling. A favorable situation is given when low temperature heating and cooling facilities, e.g. floor or wall heating systems or activated ceilings, are applied for heating and cooling. During heating operation, a low temperature latent heat storage can be used to balance the heat generation by the solar thermal system and the heat supply to the heating system. Thus, a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

In cooling mode, the same storage is used in combination with a dry cooling tower to absorb/reject the waste heat in order to replace a wet cooling tower. By that means heat rejection of the chiller is shifted partly to periods with lower ambient temperatures, i.e. night time, or to off-peak hours.

2 System Design

In conventional absorption cooling installations, wet cooling towers designed for a coolant supply/return temperature 27/35 °C are applied. To use a dry air-cooler, cooling water temperatures have to be increased to 40/45 °C. As a consequence of the increase of the cooling water temperature, the temperature level of the driving heat supplied to the regenerator of the absorption chiller has to be increased accordingly.

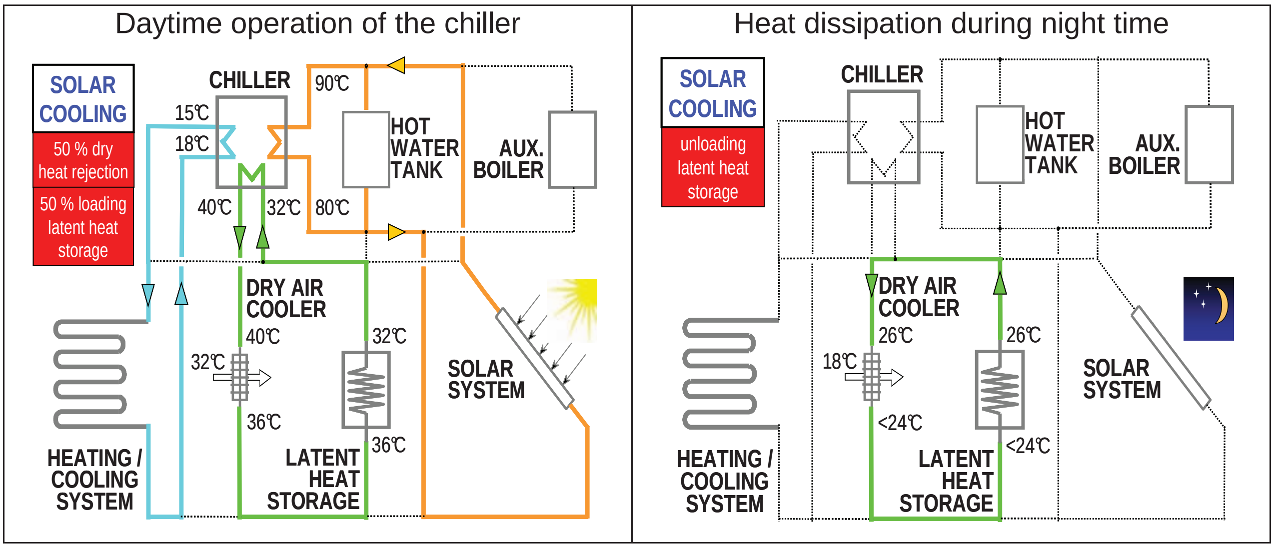


Fig. 1. System scheme of the solar cooling installation supported by latent heat storage.

By integrating a heat storage into the heat rejection system of the absorption chiller, a part of the reject heat can be buffered during peak load operation of the chiller, allowing the application of a dry air cooler with coolant temperatures of 32/40 °C. During off-peak operation or at night time when lower ambient temperatures are available, the stored reject heat can be discharged.

As a consequence of the reduced coolant temperature arising from the integration of the latent heat storage, lower temperatures of the driving solar heat are feasible to operate the absorption chiller. Thus a higher solar gain is obtained for a given size of the solar collector system (see Figure 1).

During the heating season, the latent heat storage buffers the solar surplus heat and balances the heat supply to the consumer by boosting the return temperature of the heating system (see Figure 2). Thus, a low operating temperature of the solar thermal system is accomplished yielding efficient operation with optimum solar gain.

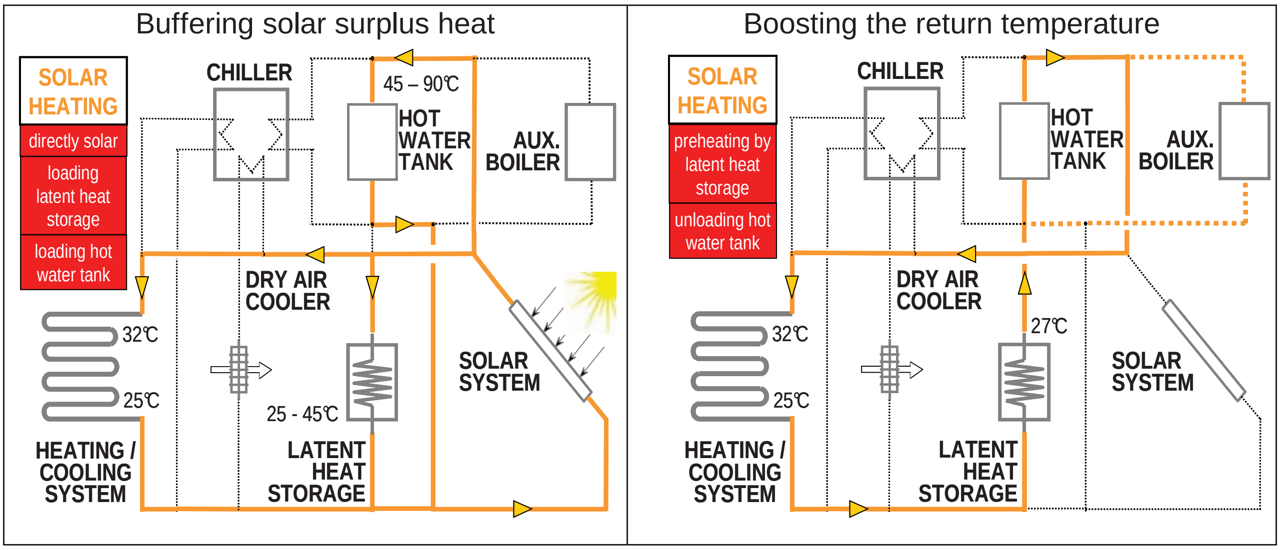


Fig. 2. System scheme for solar heating with latent heat storage

Simplified system configurations for cooling and heating operation are given in Figure1 and Figure 2. The realised piping and instrumentation is given in Figure 3. This kind of installation facilitates a very flexible and efficient use of solar energy in existing buildings with heating and cooling systems operating at moderate heating and chilled water temperatures.

- Conventional design of the solar thermal system with primary loop in water/glycol and a secondary loop connected to the heating system, the hot water tank, and the generator of the absorption chiller via the high temperature (HT) heat distributor.

- The dry air-cooler and the latent heat storage are integrated into a secondary loop, linked to the heating system by the plate heat exchangers WT 4-2. A set of valves is applied for switching between the different operating modes in summer and winter, enabling for boosting the temperature of the return flow of the activated ceilings, the loading and unloading of the latent heat storage, and the emergency cooling of the solar thermal system during summer operation. The latent heat storage had to be integrated in the secondary reject heat loop due to safety reasons and simplified unloading by the dry air cooler in cooling mode. In further installations also the chiller’s absorber and condenser should be integrated into this water/glycol loop to avoid a temperature increase of the cooling water due to the heat transfer in the plate heat exchanger. Furthermore an increase of the electrical COP could be achieved.

- During the heating season low (NT) and high temperature (HT) distributor are connected to each other.

- A ground water well linked to the NT distributor and a pellet boiler linked to the HT distributor serve for backup in cooling and heating mode, respectively.

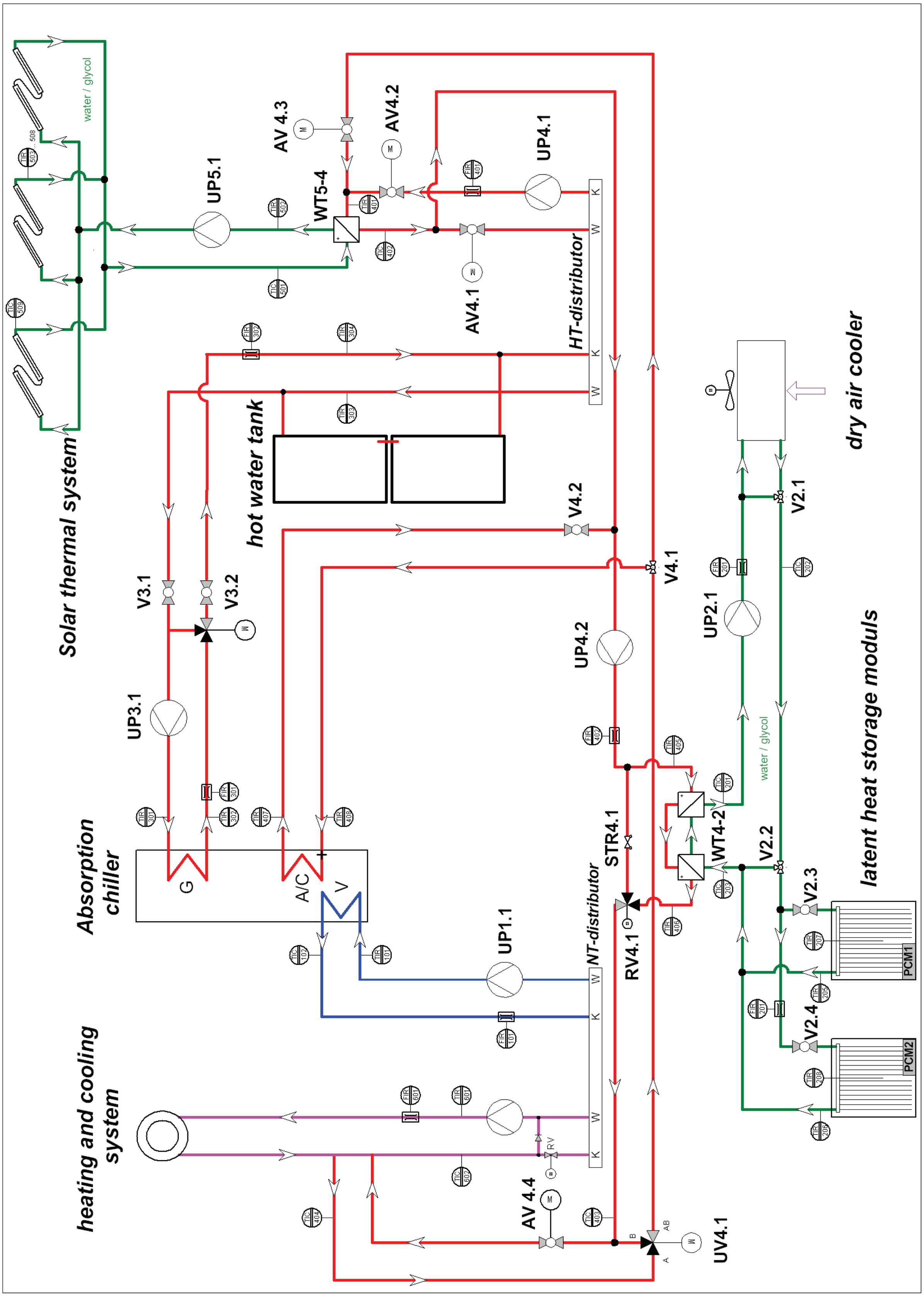


Fig. 3. Hydraulic scheme for the solar heating and cooling system with latent heat storage

3 Control Strategy

The control strategy for this solar cooling and heating system with all parameters, interactions and safety switches is very complex and can not be discussed in detail. The following list only gives a short overview about the control units and their major tasks.

(BMS) Building management system

- Selects heating or cooling mode (dis/connects NT and HT distributor).
- Controls supply temperature of the heating and cooling system as a function of the ambient temperature or the dew point temperature, respectively.
- Activates backup sources and valves of the building.

(Beckhoff SPS) PLC control of the absorption chiller

- Controls the internal pumps, hot and chilled water pump.
- Controls the chilled capacity: Controlled parameter is the outlet temperature of the chilled water; manipulated parameter is the driving heat input with control of the chilled water supply temperature.
- Requests external pumps and dry air cooler.
- Observes internal process parameters and activates the automatic purge procedure with 30 minutes purging every five days.

(UVR1611) Universal programmable logic controller

- Enables the solar thermal system when the temperature of the collector exceeds the:
  → Winter: return temperature of the heating system (activated ceilings) + 5 K
  → Winter: phase change temperature of the latent heat storage + 5 K
  → Winter: temperature in the top of the hot water tank + 5 K
  → Summer: minimum driving temperature of the absorption chiller 60 °C + 5 K
- Adjusts the speed of the different pumps (solar secondary loop to the HT distributor, heating loop to activated ceilings, flow rate through latent heat storage), according to the requested temperature.

  → direct solar heating < 35 °C
  → direct solar heating and/or loading the latent heat storage <50 °C
  → loading the hot water tank 50 °C up to 90 °C
  → emergency cooling of the solar thermal system 90°C
- Switches and controls all valves in the solar heating and cooling system.
- Enables the absorption chiller when cooling mode is selected and driving heat in the solar thermal system or buffer tank is sufficiently high (> 65 °C).
- Enables the reject heat pumps and the dry air cooler and adjusts fan speed for constant cooling water outlet temperature 30 °C.
- Calculates optimized unloading time and duration for the latent heat storage depending on the ambient air temperature or the return temperature of the activated ceilings.

4 Monitoring Equipment

4.1 Installed Equipment

Each heat sink or source is metered with two temperature sensors (inlet and outlet) and one electromagnetic flow meter. Apart from the TIC sensors for control purposes all sensors are connected to an Agilent 34980A Multifunction Switch/Measure Unit with shielded and grounded cables.

Installed sensors:

32	TIR	Pt-100 temperature sensors	1/10 DIN B	4-wire	resistance
	- Ambient air - Surface temperature of one solar collector - Outlet of each solar collector array - Solar heat exchanger, primary and secondary inlet and outlet - hot water tank - absorption chiller inlet and outlet of generator, evaporator and absorber/condenser - Reject heat exchanger, primary and secondary inlet and outlet - Dry air cooler outlet - Latent heat storage modules outlet and PCM temperature				
18	TIC	Pt-1000 temperature sensors	1/10 DIN B	2-wire	resistance
	- only for control purposes				
8	FICR	Electromagnetic Flow meters	± 0.2 %	2-wire	0-10 V
	- solar secondary loop - primary and secondary reject heat loop - latent heat storage module1 loop - absorption chiller, chilled water loop - absorption chiller hot water and hot water tank loop - heating and cooling system loop of the building				
4	LIR	Wattmeter	± 0.1 %	2-wire	0-10 V
	- Solar thermal system components and UVR1611 control unit - Reject heat loop and latent heat storage components - fan of the dry air cooler - absorption chiller and control unit				
1	RIR	Pyrometer	< 2 %	2-wire	mV
	- direct and indirect solar insolation on the aperture area of the solar collector				

4.2 Period of Measurement

The period of measurement started in January 2008 and is ongoing. Due to latent heat storage performance tests and capacity tests the data of the following days have not been taken into account for the evaluation of the annual performance data:

30.01. – 31.01.2008	21.04. – 22.04.2009	23.03. – 28.03.2010
20.04. – 23.04.2008	09.07. – 10.07.2009	
17.06. – 18.06.2008	17.09. – 18.09.2009	
29.10. – 30.10.2008		

4.3 Annual / Monthly Data

Monthly data of the solar cooling and heating system for the years 2008, 2009, and 2010 are given in the tables 1 to 3 below. By replacing some pumps with new high efficiency models in June 2009 the electrical COP in cooling mode has been increased significantly.

Table 1: Monthly energy values for the measuring period 2008

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2008	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	3182	6029	6404	6811	10009	8439	9660	8736	5980	4689	3094	1567
solar thermal gain	1167	2300	2110	2006	2804	2220	2831	2743	1968	1793	1060	749
Absorption chiller												
consumed heat	0	0	0	77	1419	1428	1874	1938	701	26	0	0
provided cold	0	0	0	44	972	949	1279	1378	490	11	0	0
Building												
cold demand	0	0	0	56	1897	4132	4035	3585	1134	95	0	0
heat demand	4763	3116	2456	958	0	0	0	0	1120	2277	3809	8ς
Secondary energy												
Total Electricity Consumption	125	262	208	204	211	267	230	255	203	165	130	104

Table 2: Monthly energy values for the measuring period 2009

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2009	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	2797	3034	4162	6776	9674	8683	9168	10074	7553	4475	3376	2097
solar thermal gain	938	919	1364	3143	2825	2195	2717	3421	1827	1367	1215	504
Absorption chiller												
consumed heat	0	0	0	61	731	981	1676	2225	540	53	0	0
provided cold	0	0	0	31	503	671	1221	1639	392	32	0	0
Building												
cold demand	0	0	0	133	1368	2198	3596	3781	965	176	0	0
heat demand	6046	4975	3209	426	116	30	0	0	8	2127	2742	♦*ϑ
Secondary energy												
Total Electricity Consumption	88	102	155	125	155	129	175	227	83	111	130	62

Table 3: Monthly energy values for the measuring period 2010

	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
2010	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]	Energy [kWh]
Solar thermal system												
solar irradiation on collector	1671	3239	5385	8782	6266	8374	9683	7417	6980	4989	2808	
solar thermal gain	255	1022	1731	2369	1253	2388	3227	2128	2126	1770	880	
Absorption chiller												
consumed heat	0	0	0	140	0	1292	2061	1134		0	0	
provided cold	0	0	0	92	0	956	1488	828		0	0	
Building												
cold demand	6234	4135	2407	779	639	284	0	55	575	2320	3278	
heat demand	0	0	0	215	751	2974	4661	2171	137	0	0	
Secondary energy												
Total Electricity Consumption	42	91	112	141	165	194	216	139	82	139	101	

4.4 Analysis of the absorption chiller and the reject heat loop

Figure 4 shows the capacity of the chiller and the heat rejection loop during the course of a typical summer day. The absorption chiller provides about 11 kW chilled water by means of 14.5 kW driving heat from 10 a.m. to 5 p.m. Until 9 p.m. chiller operation is continued with surplus driving heat from the buffer tank, gained during daytime. The chiller's reject heat is primarily dissipated by the dry cooler. Although auxiliary energy saving has not been the major focus of this first pilot installation, the electrical COP (chilled water capacity per driving electric input) is about 7 to 10. The daily electrical COP comprises also the pumping energy during regeneration of the latent heat storage during night time and reaches values between 4.5 and 8 depending on the stored heat in the latent heat storage and the ambient temperature.

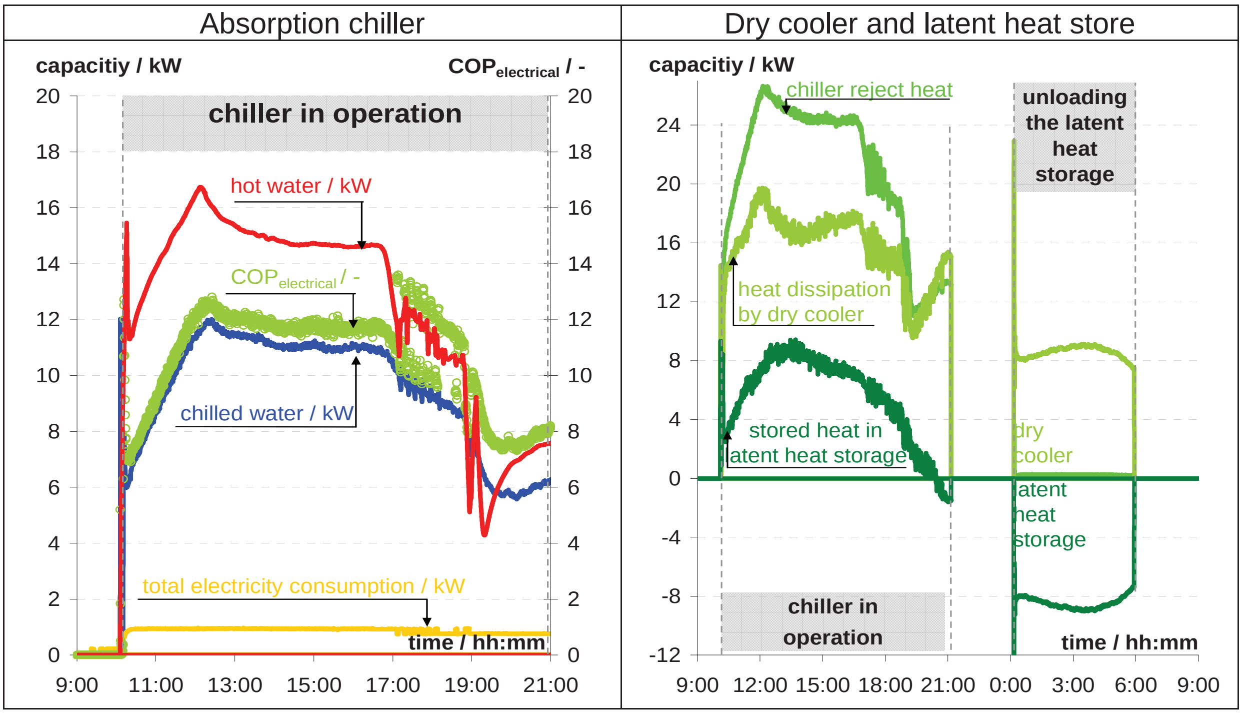


Fig. 4. Operational data of the solar cooling system during the course of a typical day

During hot ambient conditions (10 a.m. to 8 p.m.) the latent heat storage supports the dry cooler with maximum cooling water temperature about 33°C. The discharging of the latent heat storage is controlled with regard to the actual heat content (here from 0 to 6 a.m.).

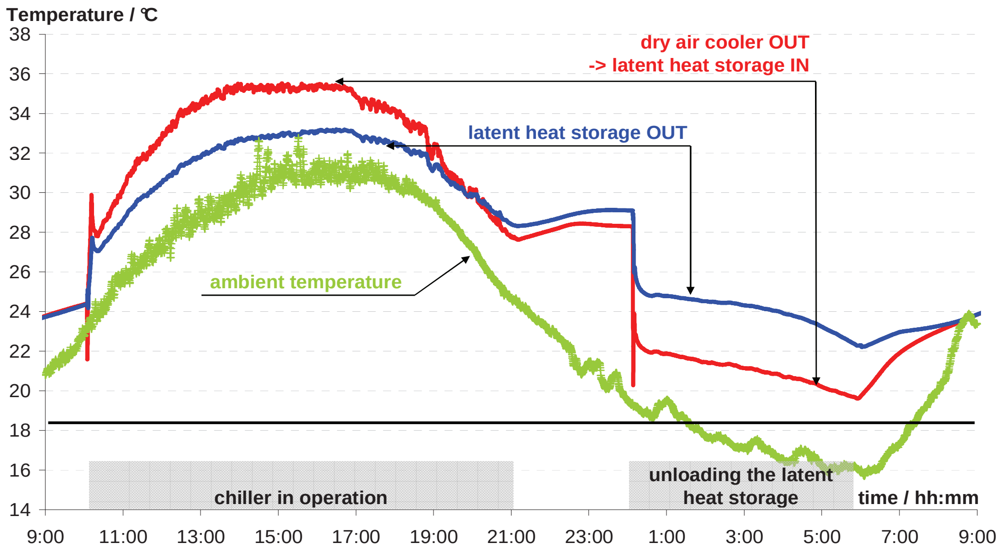


Fig. 5. Temperature profile of the reject heat loop

4.5 Detailed Analysis of the long term stability of the latent heat storage

Between 2007 and 2010 recurring measurements of loading and unloading the storage were carried out to determine long term effects on the PCM. These measurements showed no degradation of thermal power or capacity (see Figure 6), confirming the assumption that the separation of the PCM could be prevented successfully.

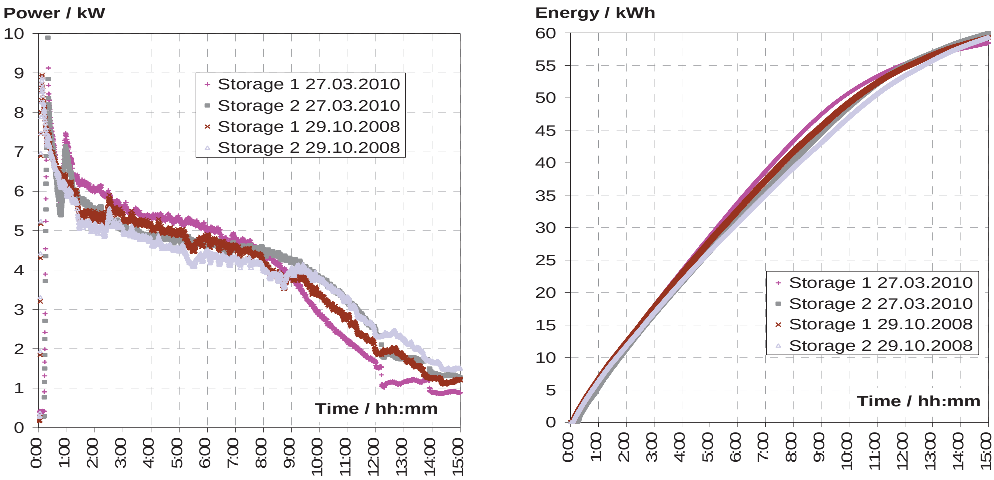
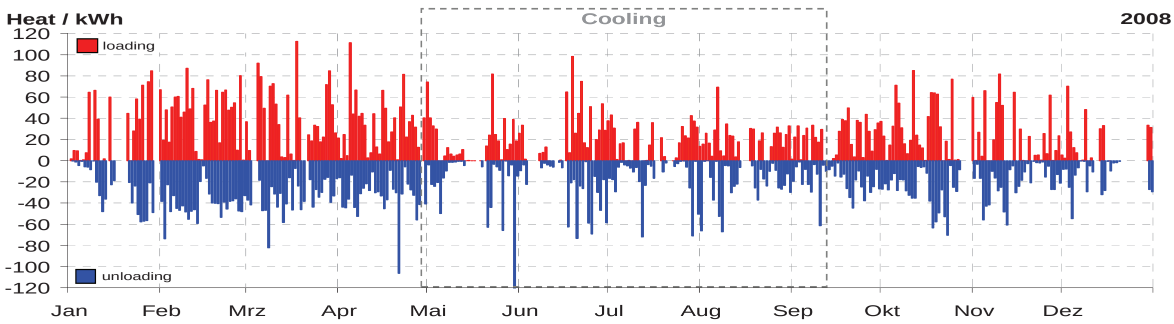


Fig. 6. Power and the stored energy of the two PCM storage modules during loading with a heat carrier supply temperature of 36 ℃ and a flow rate of 1.5 m^3/h

Figure 7 shows the charging and discharging cycles of the latent heat storage during summer and winter operation in 2008 and 2009 (data for 2009 in the following text in brackets), respectively. In total 293 (223) charging and discharging cycles have been performed. During the heating season a total of 6478 (3922) kWh solar heat has been stored, whereof 5741 (3323) kWh heat could be discharged and transferred to the heating system. In the cooling period 2105 (2053) kWh reject heat of the absorption chiller have been stored. In this case with close temporal coherence of loading and unloading a storage efficiency of 96,6% (85,4%) has been accomplished. Figure 7 furthermore illustrates the large impact of the weather situation on the utilization of the latent heat storage: Due to mild and sunny weather in spring 2008 rather large amounts of heat have been processed whereas a substantially lower utilization of the storage has been accomplished in winter 2008/09.



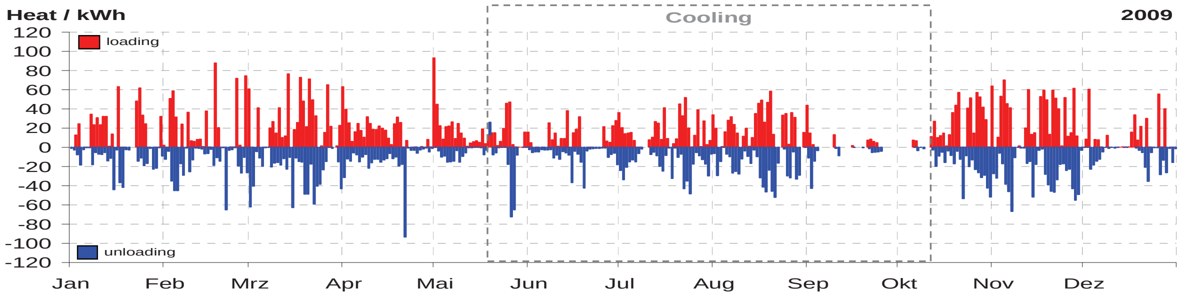


Fig. 7. Loading and unloading cycles of the PCM storage

In order to operate a latent heat storage as efficient part of a heating or cooling system, precise information about the actual energy content of the storage is essential. Due to the fact that a metering method for the thermal content of a latent heat storage is still a matter of development, up to now the potential for efficient charging and discharging of the storage could not be fully exploited. As a first attempt, different approaches using a heat meter to indicate the actual state of charge had only little success due to the small temperature difference of the heat carrier between storage inlet and outlet and the resulting inaccuracy of the control. To integrate the storage into the solar heating and cooling system despite the missing charge control, for the moment different temperature criteria are used to achieve a rough estimate of the actual state of charge. The development of a more precise metering procedure is ongoing.

Figure 8 shows the transferred heat for a series of loading and unloading cycles of the mentioned latent heat storage during solar cooling operation in summer 2008 an 2009, respectively. In cooling mode complete discharge of the accumulated heat during the following night is strived for. Yet , due to the imperfect charge control a substantial mismatch between loading and unloading of the storage occurred in 2008. Finally, by improving the control strategy in 2009 the daily disbalance of the heat storage has been minimized.

As a second conclusion from Figure 8, it can be stated that the storage allows for efficient mid-term storage, e.g. storage periods of some days, without substantial heat loss: A proof is given by the period May 23 to 30, 2008, when the solar gain accumulated during 8 days is almost completely extracted on two days only.

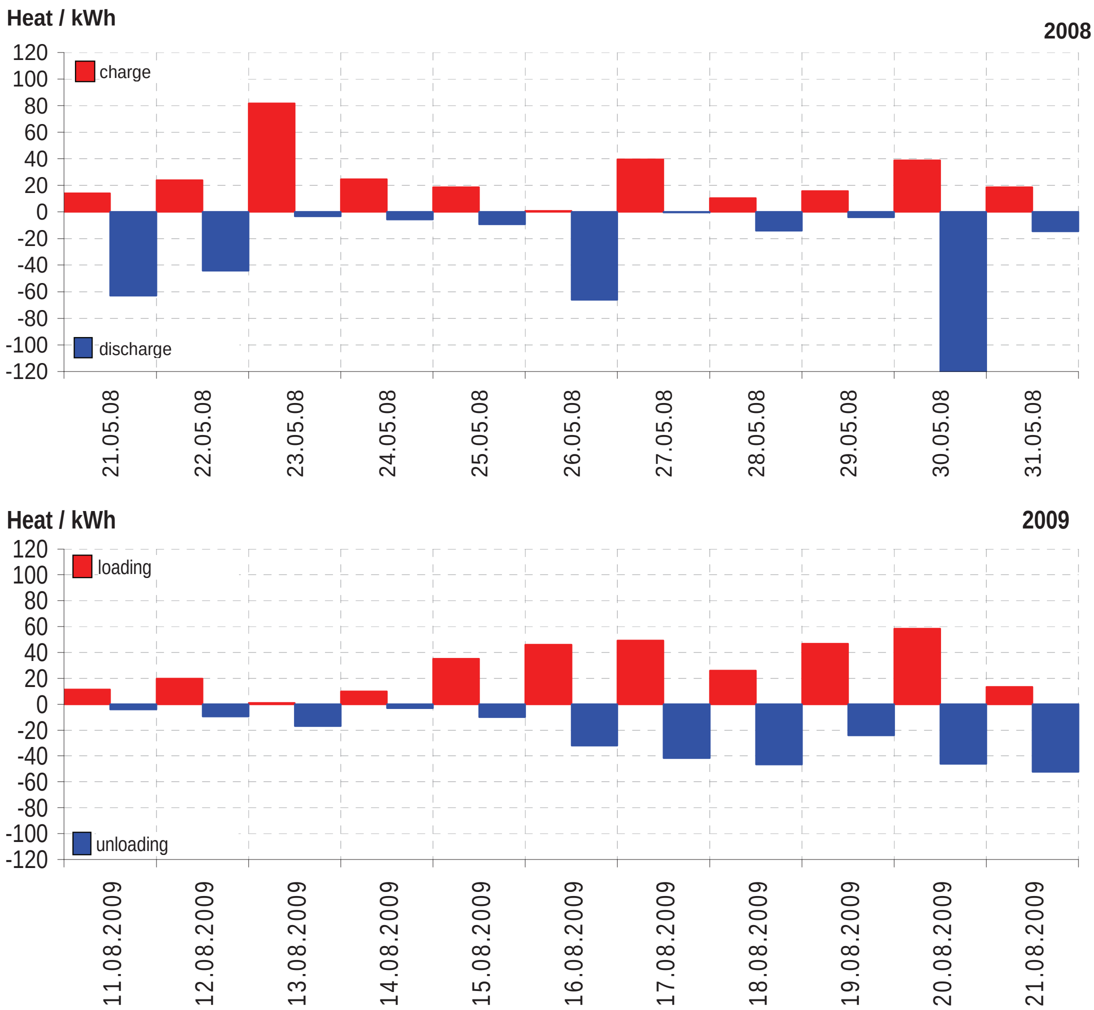


Fig. 8. Loading / unloading cycles of the PCM Storage 2008 / 2009

In 2008 a substantially higher level of insolation compared to 2009 allowed for a higher solar heat accumulation accompanied by a lower building heat demand. Consequently, a monthly solar fraction of 24 % to 85 % was reached in the period Jan – March 2008, whereas under severe winter condition in January and February 2009 the solar coverage was limited to 15 to 18 % only. In the transitional period March/April 2008 and April/May 2009 almost fully solar coverage has been reached. For the whole winter periods 2008 and 2009 solar fractions of 48 % and 33 % have been achieved, as shown in Fig. 9.

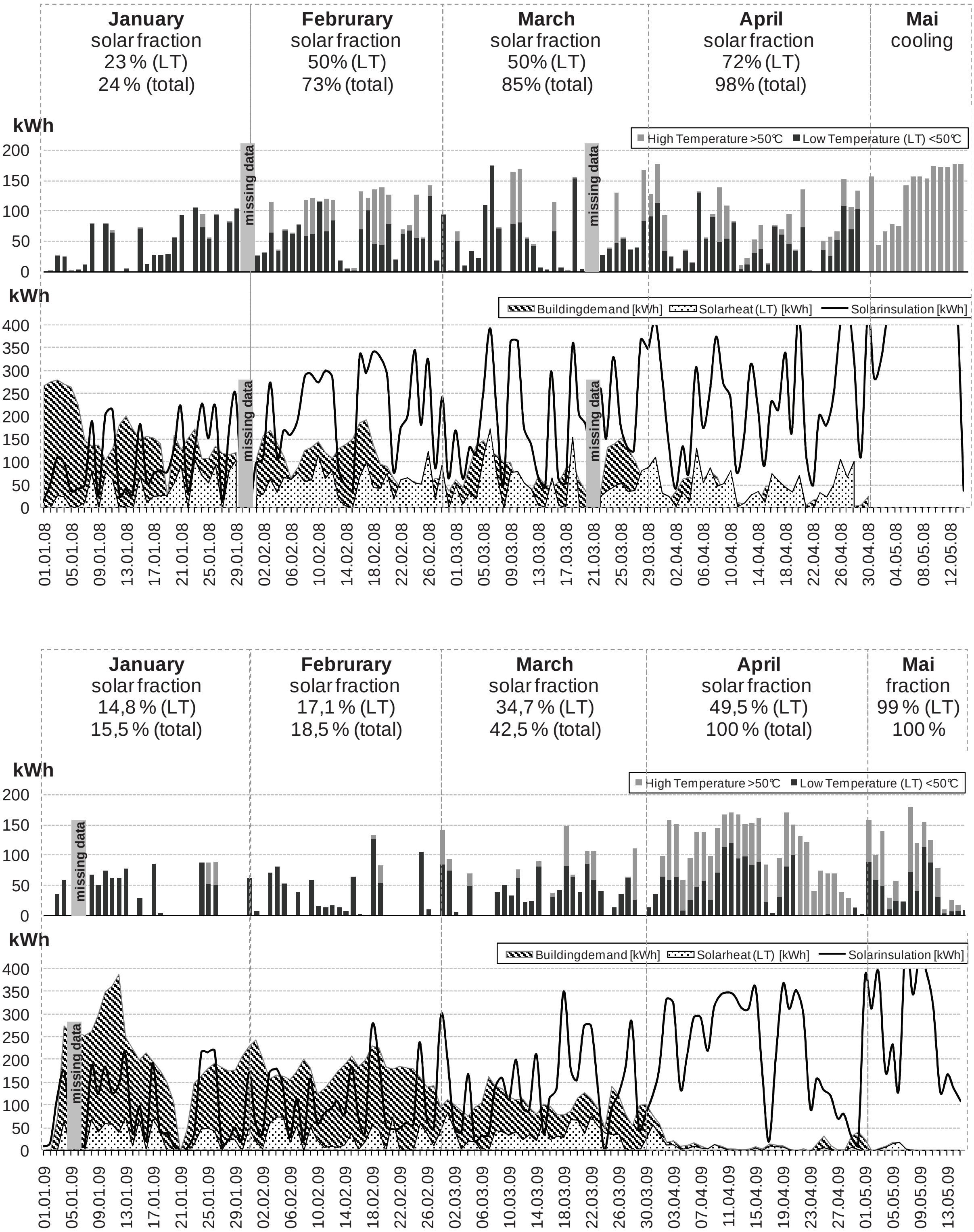


Fig. 9. Measurement data from solar heating period 2008 (top) and 2009 (bottom)

Simulations confirm the significant increase of the solar gain e.g. during the heating period from October to March of about 20 %, compared to a hot water tank system with the same specific cubic content per square meter collector surface area (in this case 40 Liters/m²).

All data of the solar fraction given above refer to the total amount of solar heat supplied to the building. These results by far exceed the performance of conventional large-scale solar installations. Pre-requisite for the obtained efficiency of the solar thermal system is the operation at moderate collector temperatures stabilized by means of the low-temperature latent heat storage. As a consequence, the major part of the solar heat supplied to the building passes through the low-temperature (LT) heat storage and only minor parts of the heat are generated at higher temperature levels, as given by the monthly data for the total heat supply to the building (total) and the contribution of the latent heat storage (LT).

5 Experiences / Lessons Learned

Within the monitoring period of 3 years various efforts in increasing the energy efficiency have been made. As a result reliable and efficient components, e.g. EC technology for pumps and fans, have been applied. Apart from this, the system effectiveness also depends on:

- the thermal COP has a crucial impact on the electrical COP. A poor COP results in an increase of the secondary energy used for dissipating the reject heat to the ambient.
- the standby electricity consumption should be reduced to a minimum. At best, unutilized components are shut down completely.
- the knowledge of the real heat content in the latent heat storage is essential for an optimized and efficient storage management aiming at high solar fraction and minimized electricity consumption.
- the pressure drop in all components should be minimized, obeying economic and thermodynamic limitations.

Further improvements will be realized in the near future:

- Replacement of the 3-phase AC fan of the dry air cooler by a 1-phase EC model. This reduces the electric energy consumption of the cooler by about 50 % and the parasitic energy demand in standby by about 15 %.
- Integration of an optimized latent heat storage with increased performance and reduced setup area (-50%).
- Implementation of a 3-way mixing valve in parallel to the latent heat storage allowing for an easy adjustment of the cooling water temperature and preventing unintended loading of the latent heat storage.
- Test of a heat content sensor for the latent heat storage.
- Control of the flow rate of the primary solar loop depending on the current insolation on the solar collectors.

6 Conclusions

A solar heating and cooling system with absorption chiller and latent heat storage has been operated for more than three years with high energetic efficiency and only minimal maintenance effort. In recurring measurements from 2007 until 2010 no aging effects of the phase change material (PCM) used in the latent heat storage could be observed.

Due to the implemented latent heat storage solar cooling with a dry cooling tower has been accomplished during the whole cooling period independently from ambient temperatures. In the heating periods an average solar fraction of 48% in 2008 and 33% in 2009 has been achieved owing to the reduced collector temperature stabilized by the latent heat storage. Aiming at series-production and wider distribution of the heat storage, in Oct 2009 a follow-up project for further development of the solar heating and cooling system has been started. By integrating a new optimized latent heat storage, accompanied by further system improvements, an electrical COP of 12 seems to be feasible for this solar cooling system.

7 Acknowledgment

The project is supported by funds of the German Federal Ministry of Environment (BMU) under contract number 0329605O and 0329605D.

8 Bibliography

M. Himpel, S. Hiebler, M. Helm, C. Schweigler, “Long term test results from a Latent Heat Storage developed for a Solar Heating and Cooling System”, Proceedings of EuroSun 2010, Graz, 2010

M. Helm, C. Keil, S. Hiebler, H. Mehling, C. Schweigler, Solar heating and cooling system with absorption chiller and low temperature latent heat storage: Energetic performance and operational experience, International Journal of Refrigeration, 32, 2009, 596–606.

S. Hiebler, H. Mehling, M. Helm, C. Schweigler, Latent heat storage developed for solar heating and cooling with melting temperature 29 °C, Proceedings of Effstock 2009 – 11th International Conference on Thermal Energy Storage, Stockholm, 2009

M. Helm, S. Hiebler, H. Mehling, C. Schweigler, “Solar Heating and Cooling with Absorption Chiller and Latent Heat Storage”, 3rd International Conference Solar Air-Conditioning, Palermo, Italy, 30.09.–02.10.2009

C.Schweigler, S.Hiebler, C.Keil, H.Köbel, C.Kren, H.Mehling, “Low temperature Heat Storage for Solar Heating and Cooling Applications”, ASHRAE Transactions, ASHRAE Winter-Meeting, Dallas, USA, Jan. 2007

F.Storkenmaier, M.Harm, C.Schweigler, F.Ziegler, P.Kohlenbach, T.Sengewald, “Small-capacity LiBr Absorption Chiller for Solar Cooling or Waste-Heat driven Cooling”, Proc. of the 30th International Congress of Refrigeration, Washington, USA, Sept. 2003

Task 38
Solar Air-Conditioning
and Refrigeration

D-A3b: Appendix 10

Monitoring Results of Solar air-conditioning with adsorption chiller of the canteen kitchen area at Fraunhofer ISE, Freiburg, Germany

A technical report of subtask A (Pre-engineered systems for residential and small commercial applications)

Date: 30.9.2011

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Contents

1 Background

In the frame of the finished EU project MODESTORE a solar driven cooling system with a 5.5kW reversible adsorption chiller from the German company SorTech, a 20m² flat plate collector field with a 2m³ buffer storage and a borehole array of three 80m boreholes, was installed at the Fraunhofer Institute for Solar Energy Systems. In the summer period, the adsorption machine is operated as a chiller driven by solar energy while the boreholes are used for heat rejection. In winter it is operated as a heat pump driven by the heat from a heating network and using the boreholes as low temperature heat source.

Ever since the commissioning (2006) of this solar cooling system, the operation has been monitored constantly. During the years of operation the chiller device has been replaced several times by later generations of the Sortech ACS (see chard 5) in order to follow the progress made in the development of small scale adsorption chillers.

2 System Design

The Fraunhofer Institute for Solar Energy Systems (ISE) institute building is an energy efficient building with passive cooling measures. An exception is the canteen kitchen area, where due to high internal loads active cooling of the supply air is appropriate. This is done by means of a small size thermally driven chiller.

The system technology is a closed cycle chilled water system with an adsorption chiller. Heat is provided by a solar thermal system and by the heat network of the institute which is fed by a CHP unit. During summer, the system runs in cooling mode (left schematic in Fig.1). The medium temperature heat of the chiller is rejected by three ground boreholes of 80 m each. In winter, the heat pump function of the machine is activated and the ground tubes act as low-temperature energy source (right schematic in Fig.2). The system thus cools the supply air into the kitchen (max. flow rate 3000m³/h) and pre-heats the air in the main channel of the air handling unit (max. flow rate 9000m³/h).

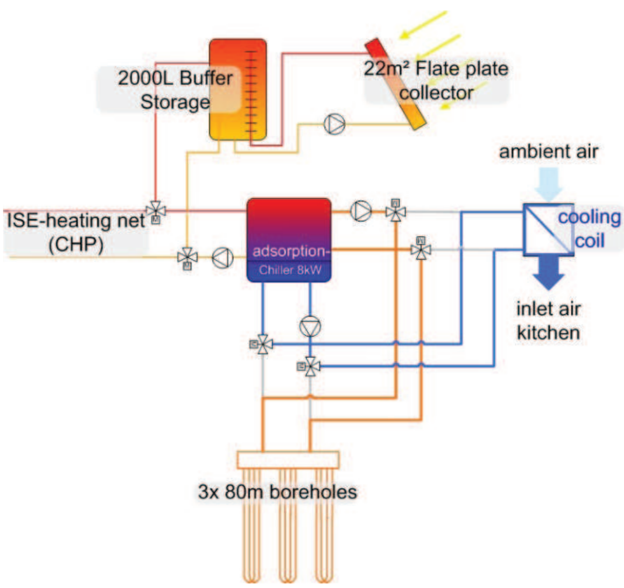


Figure 2(left): Simplified hydraulic scheme of the system in cooling Mode. The boreholes function as heat sink for heat rejection of the adsorption process.

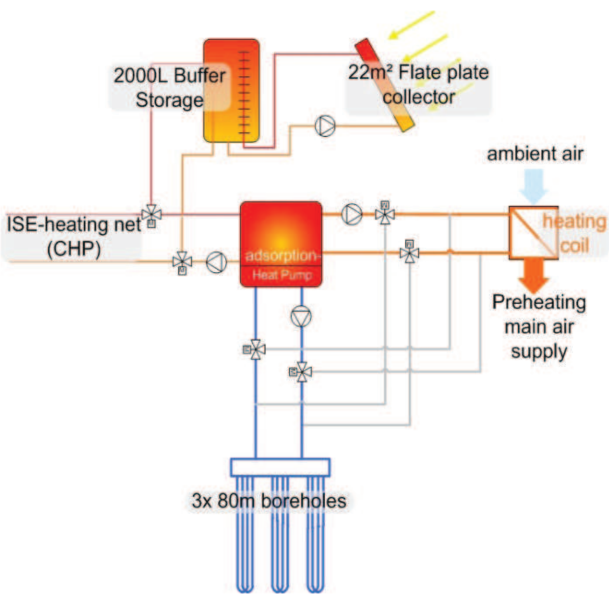


Figure 1(right): For driving the system in heating Mode the boreholes are switch on the cooling circuit and become a low temperature heat source for heat pump operation.

Current System specifications

Collector Field		
Type		Flat Plate Collector
Model		Solvis F35
Manufacturer		Solvis GmbH
Heat Fluid		Water/Glycol
Collector Field Area	[m²]	21.9 (Aperture Area)
Slope		36.6°
Azimuth		0° (direct south)
Number of Modules; hydraulic design		6; 3 x modules in series, 2 x parallel rows
Collector pump		Grundfos 25
Electric Capacity	[W]	measured: 130
Flow Rate	[l/h]	mesured: 620
Control strategy		Diff. ON/OFF; variable Flow Rate
Commissioning date		April 2001
Additional note		Former system design for hot water supply of the canteen / currently: heat support only for solar cooling system

Hot water Storage		
Number		1
Type		Glazed Steel
Charging method		Stratified charger / “Low Flow System”
Storage Volume	[m³]	2,0
Insulation		100mm PU Foam, Plastic Cover

Chiller		
Type		Adsorptions-Chiller
Model		ACS08
Manufacturer		SorTech AG
Number		1
Cooling Capacity	[kW]	8.0
Heating Capacity	[kW]	21.5
Electric Energy Consumtion	[W]	Measured: 11.9
COP (thermal)	[-]	Cool-Mode: 0.6 / Heat Mode: 1.5

Terrestrial Heat Probes		
Type		double-U-pipe, 40mm PE100
Boreholes		3x 80m, each 152mm diameter
Borehole filling		Thermocem – AZBUT GmbH
Thermal Capacity		Above 20 kW
Heat conductivity	[W/mK]	3.22
Thermal Borehole Resistance	[mK/W]	0.088
True Ground Temperature	[°C]	13.5

Figure3: ***Top Left***: Flat Collector Area about 22m². ***Top right***: former Adsorption Chiller ACS05. ***Bottom Left:*** Borehole with injected pipes. ***Bottom right:*** Main Air supply unit of the ISE office department with integrated cold and heating coils

3 Control Strategy

The operational concept foresees an operation only during weekdays. The operation conditions are as follows:

Cooling operation is carried out when:

- the inlet air temperature exceeds 20°C (2K hysteresis),
- the air temperature in the kitchen is above 23°C (2K hysteresis),
- the time is between 6:45 and 16:00 o'clock.

Heating operation is carried out when:

- the inlet temperature in the main air duct is below 14.5°C (3K hysteresis),
- the inlet air temperature is above 3°C (freeze protection of the machine),
- the time is between 6:45 and 19:00 o'clock.

Solar heat is used whenever the mean temperature in the upper part of the storage is above 73°C with a 5K hysteresis for turning off solar heat supply. In spring and autumn it may happen that the air temperature falls below 14.5°C in the mornings and thus the heating mode is activated, but later during the day temperatures in the canteen kitchen rise above the threshold for cooling operation. In these cases the system is operated in the heating mode first and later in the cooling mode. This operation is called ‘alternate mode’. The volume flows in the three circuits are kept constant and correspond to the nominal flows required by the chiller.

4 Monitoring Equipment

The data acquisition system consists of internally integrating heat meters with matched Pt100 type temperature sensors. The integrator has a sampling rate of 1s and calculates cumulated energy amounts and mean temperatures and powers. This internal sampling rate assures a correct collection of energy data for the highly dynamic temperature patterns characteristic of adsorption systems. The integrator and further temperature sensors are read out by a computer with a sampling rate of 15s. The monitoring software further reduces these values to cumulated energies and mean temperatures which are stored with an interval of 5 minutes in the raw data measurement file. The storage interval can be set by the system operator and thus allows a flexible data management. The post processing of the raw data further reduces the values to hourly accumulated and mean values – depending on the quantity considered. For the hourly mean temperature values also a standard deviation is calculated in order to judge the stability of the temperature within the evaluated hour.

4.1 Installed Equipment

The Monitoring System can be separated into four sub levels.

- Linux based Embedded PC for system control as well as data recording
- ICP AD-Modules for measuring Temperature values of 4Wire Pt100 and signals of the electrical energy counters
- M-Bus string to read out several Heat-Meters of the different hydraulic circuits
- Separated weather station of the ISE

On the **Embedded PC** is one system control program operating as well as two other sub-programs recording the measurement values of the ICP string and the M-Bus string. All sensor values and all actor set values are recorded at one single text file for further system analysis.

About 40 Pt100 Temperature sensors, 4 electric energy meters and 11 actors are attached to different **ICP AD Changers**. Due to the fact, that the main purpose of those temperature sensors is for system control issues, low accurate and non calibrated sensors were installed. To monitor the electricity consumption of the three circuit pumps, the chiller and the solar system, energy meters about the accuracy of 0.5 Wh per Impulse were installed.

To monitor the thermal energies at the different hydraulic circuit of this system, commercial heat meters, controlled by the **M-Bus** system, were installed. The temperature sensors of each of these units are paired Pt100 couples with a tolerance of 1/10B class. The accuracy of the flow sensor of these units is below 3% of the measured value. The overall failure of each heat meter comes out to 4 - 8% of measured thermal heat flux, depending on the temperature spread.

The radiation (GHI,DHI) upon the collector field as well as the ambient temperature is monitored by an external **weather station** of the ISE. The local Irradiation has to be post processed by using Trnsys to get the true radiation on the tilted aperture area of the collector field.

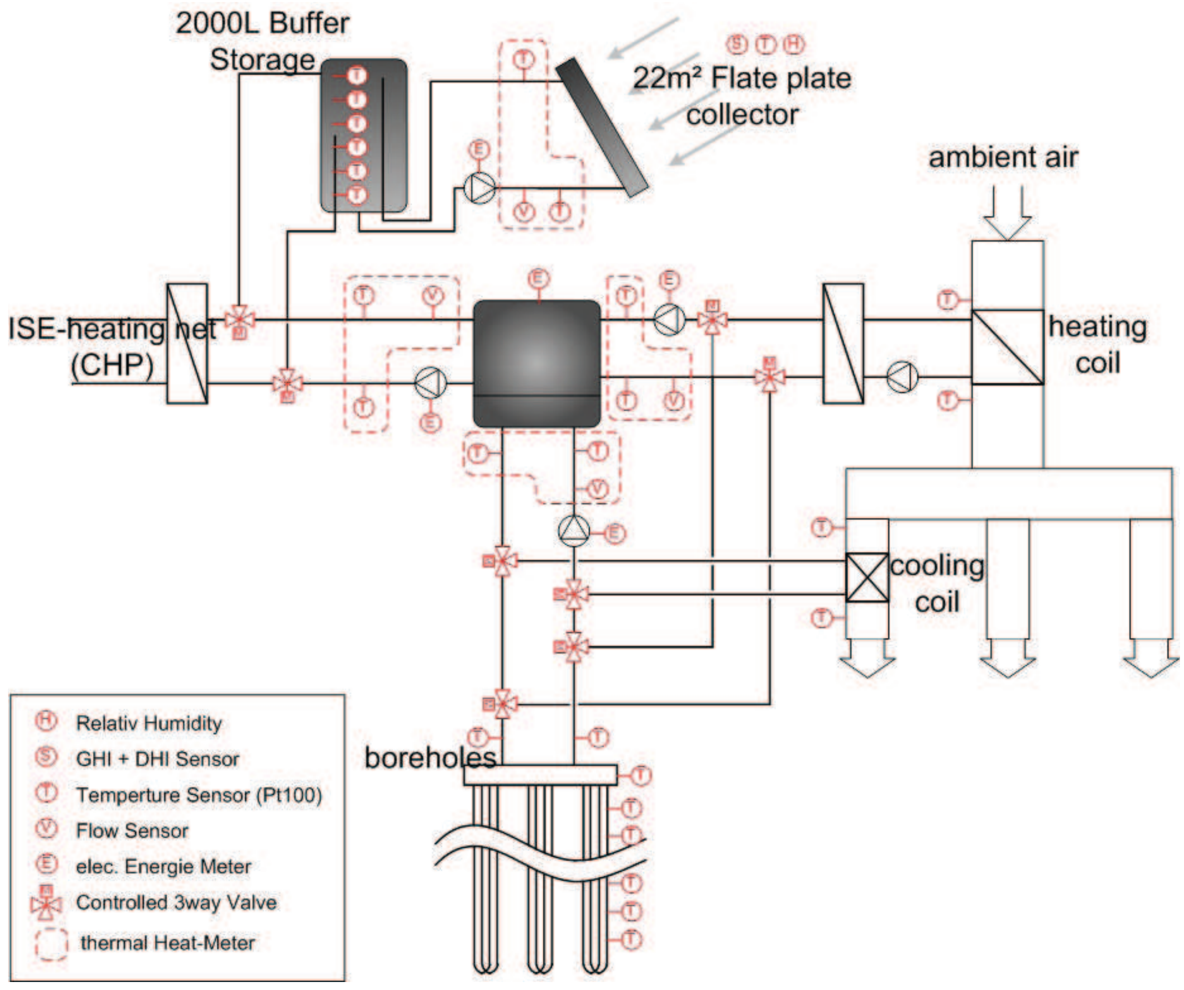


Figure4: Scheme of the installed measurement equipment

4.2 Period of Measurement

After the installation of the solar cooling system in 2006 the final commissioning of the entire monitoring system and plant operation take place in June 2007. Ever since, the monitoring has been done continuously with two exceptions of less than five months. During the system operation over the past five years the adsorption chiller has been replaced by the latest available chiller generation. Due to the increasing chiller power, a replacement of the circular pumps by a high efficient pump generation to gain suitable flow rates has become necessary in 2010.

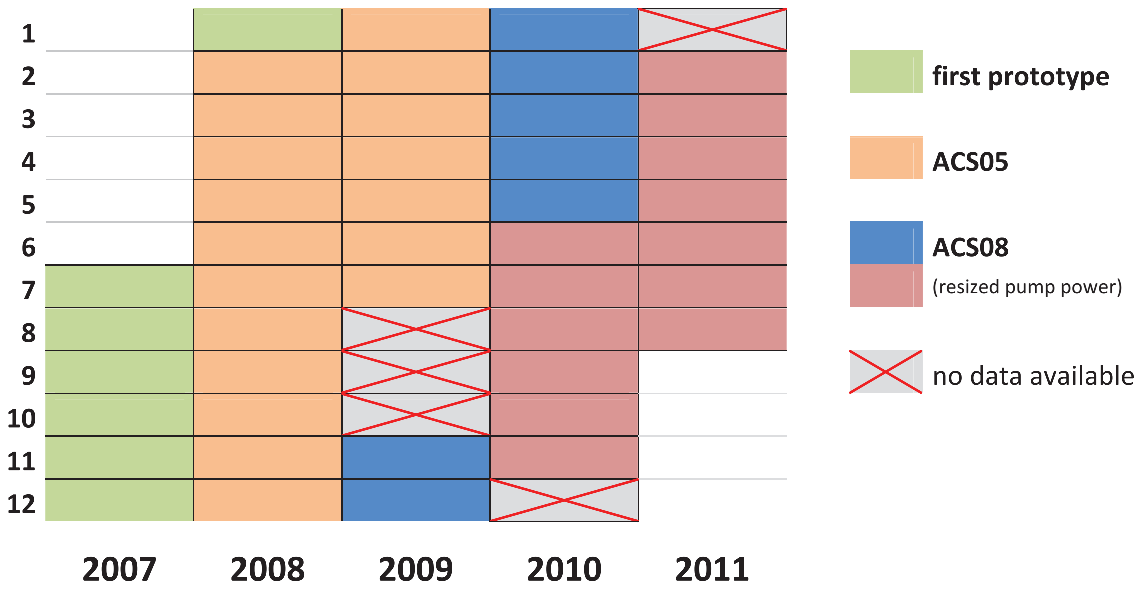


Figure5: Overview of monitoring periods

5 Monitoring Results

5.1 Annual / Monthly Data

At the following charts the results were sorted by several installed generations of adsorption chillers. No separation by cooling or heating mode operation has been done by analyzing the collector system performance, thus the seasonal feedback of the plant operation on the solar hot water system is included. Due to the fact that the hot water storage has to reach the temperature of 75°C minimum before the stored heat can be utilized by the system, there is a threshold of irradiation upon the collector performance improves.

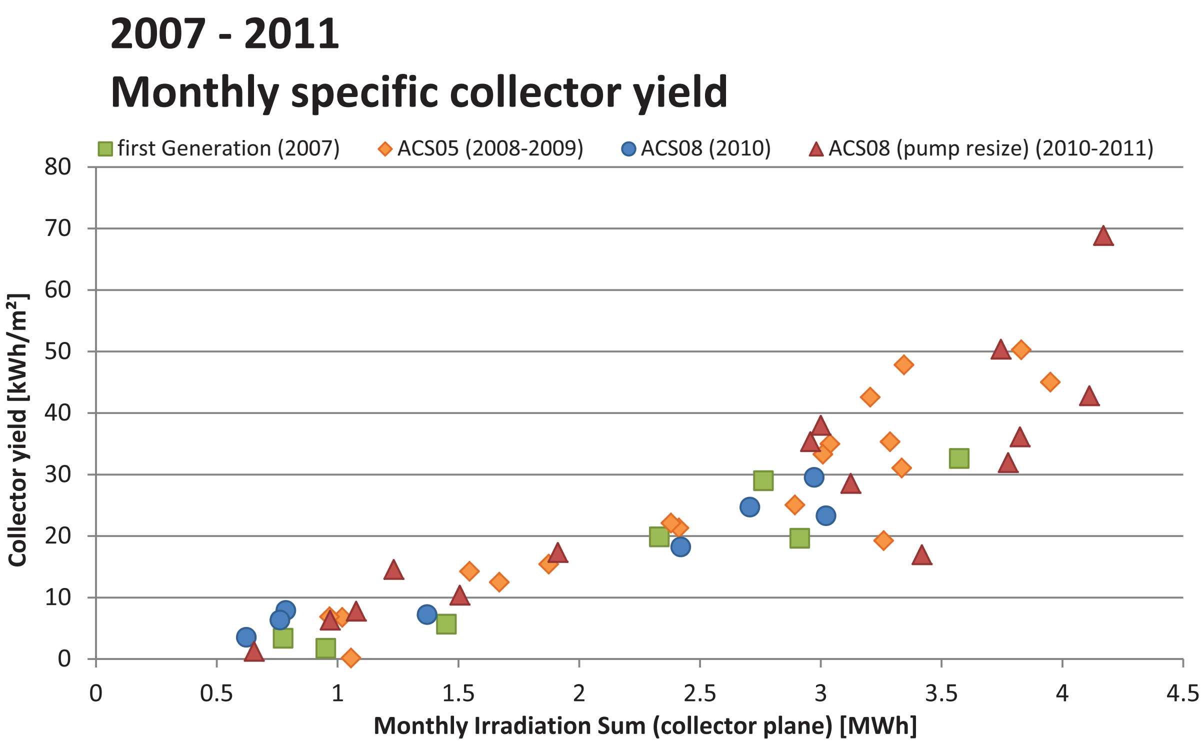


Figure 6: Monthly values of the specific collector yield of all the years of plant operation (2007-2011) plotted against the monthly irradiation sum on the 22m² collector plane. The values refer to the gained heat, supplied to the hot water storage.

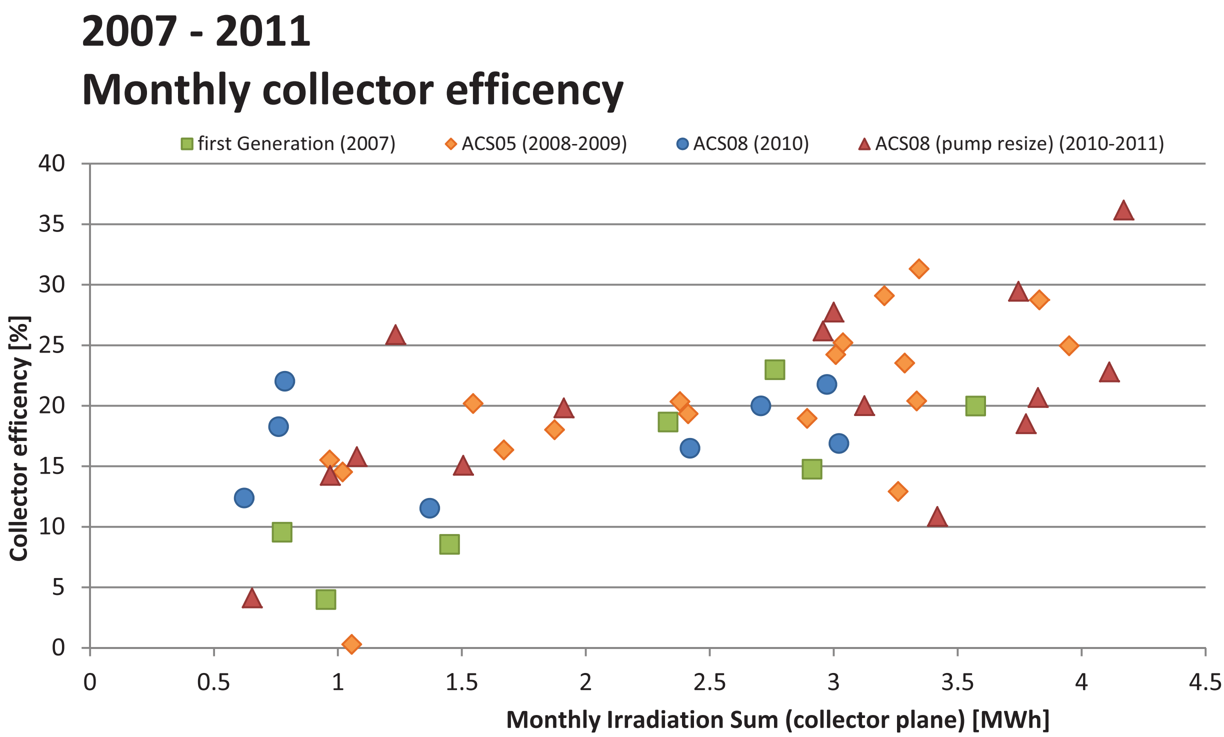


Figure 7: Monthly values of the collector field efficiency according to the previous Figure 6.

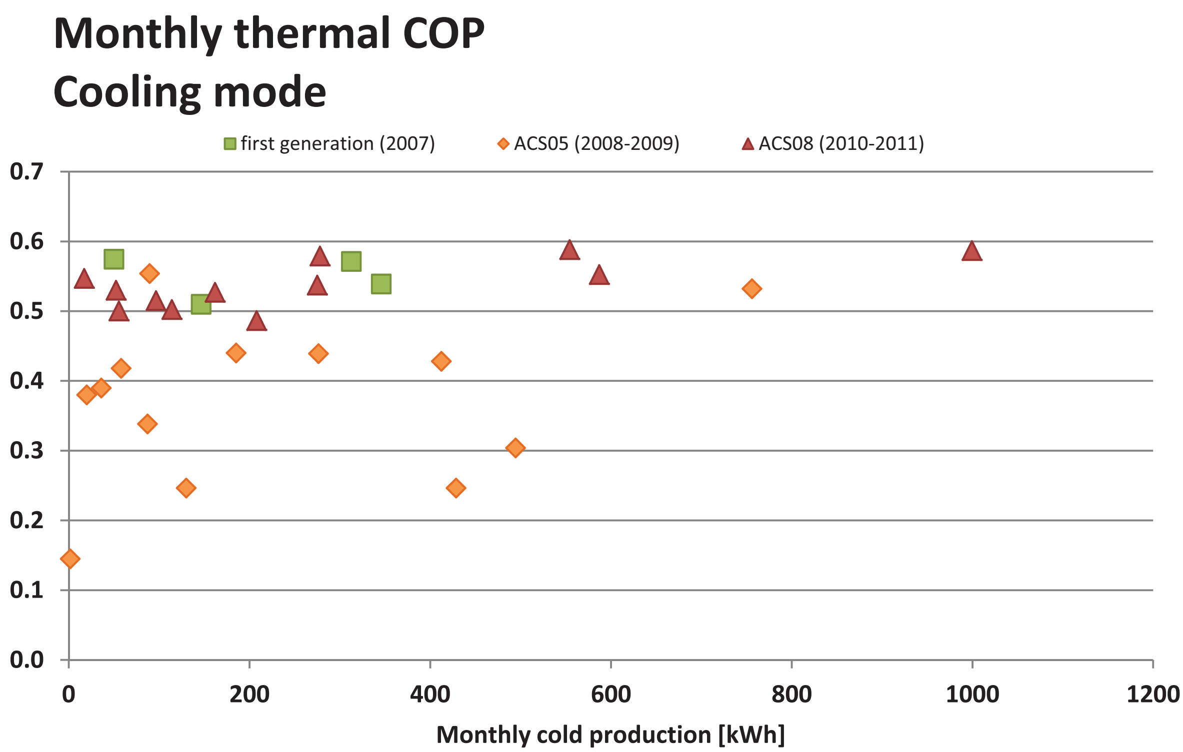


Figure 8: Monthly values of the thermal COP vs produced cooling energy sorted by chiller capacity of the different ACS generations

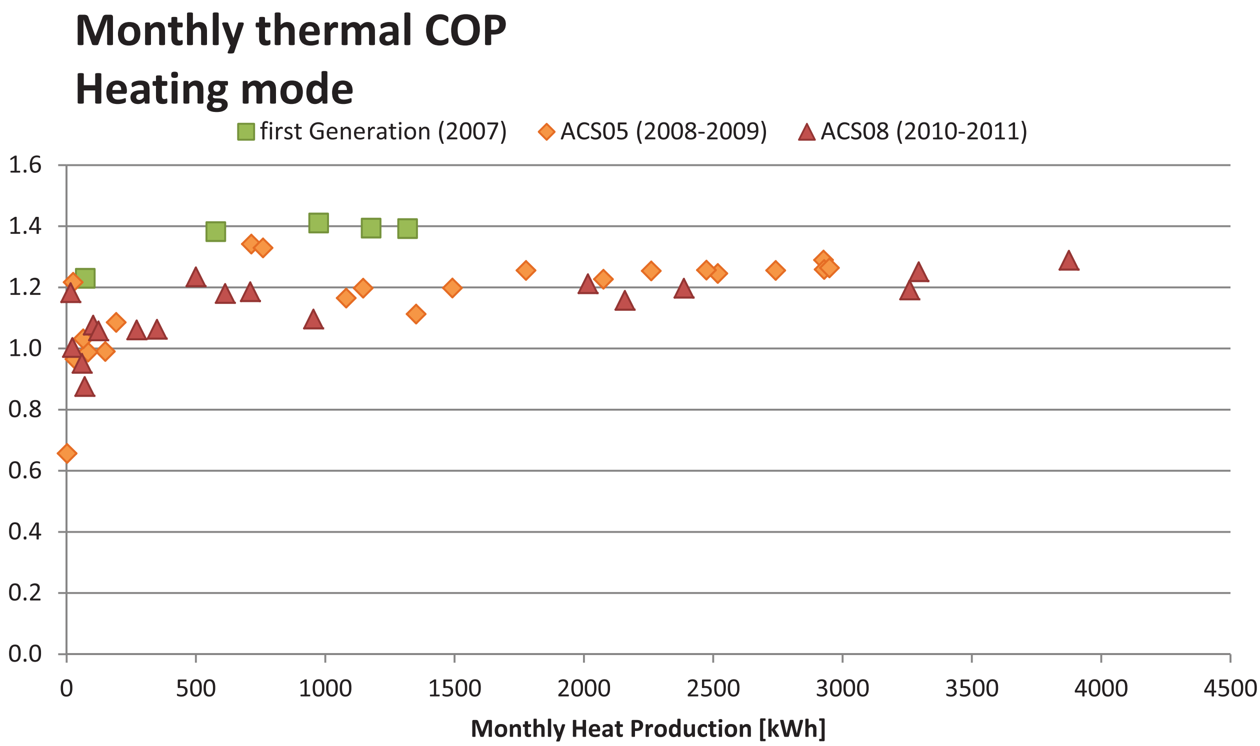


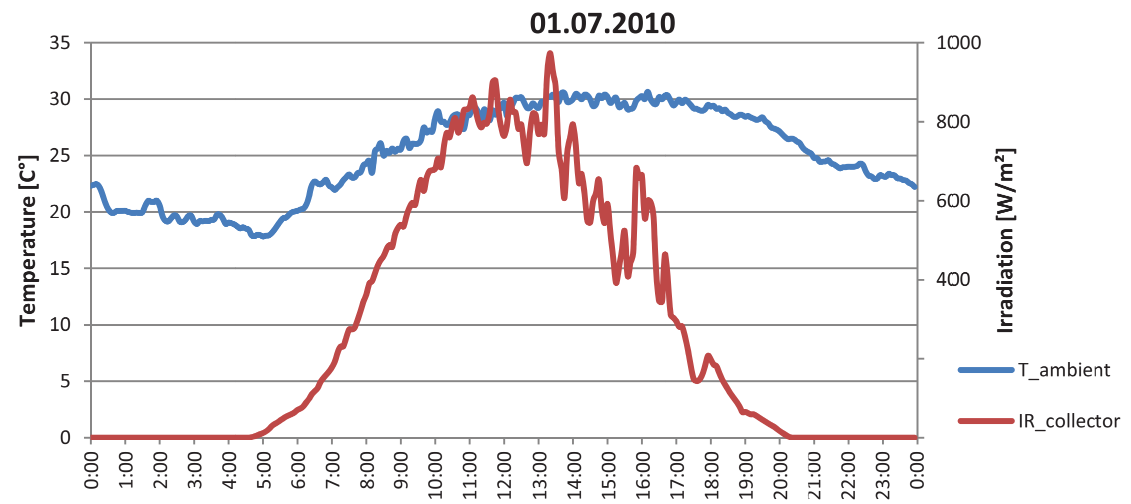
Figure 9: Monthly values of the thermal COP vs produced heat energy sorted by chiller capacity of the different ACSgGenerations

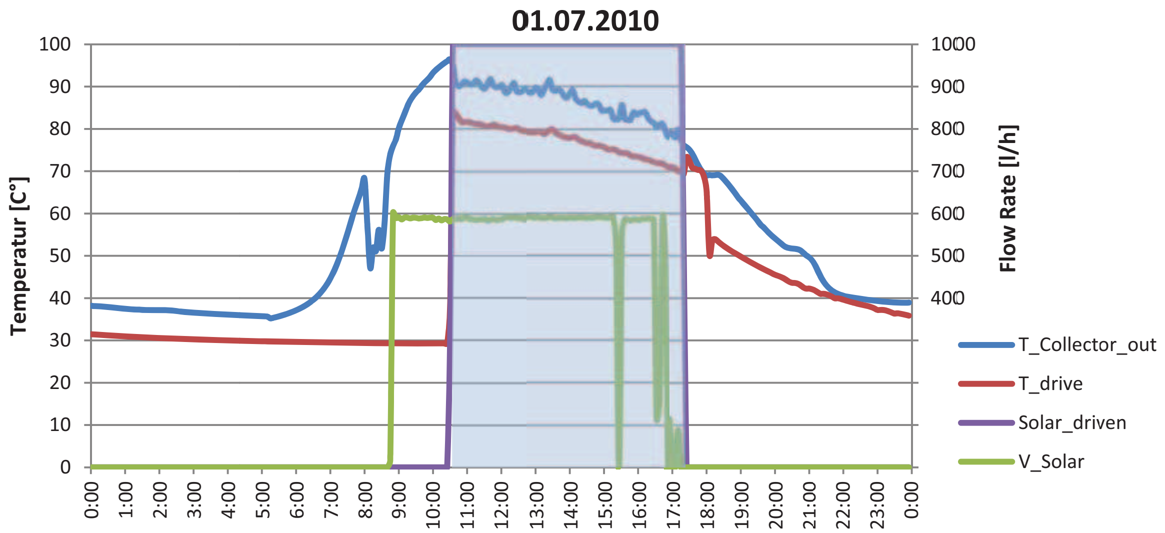
2010	Collector efficency	share of solar heat on total heat input		COP thermal		COP electric		Cold Production	Heat Production
		Cool Mode	Heat Mode	Cool Mode	Heat Mode	Cool Mode	Heat Mode		
	[%]	[%]	[%]	[-]	[-]	[-]	[-]	[kWh]	[kWh]
January	18,3	--	--	--	1,29	--	39,0	--	3008
February	16,5	--	3,4	--	1,25	--	33,1	--	2634
March	21,8	--	13,4	--	1,20	--	28,1	--	1994
April	20,0	81,0	18,1	0,53	1,19	6,0	22,7	52	599
May	16,9	70,9	18,1	0,52	1,06	4,6	19,5	97	256
June	36,2	73,5	76,3	0,59	1,00	6,1	20,9	999	22
July	26,2	49,9	8,0	0,59	1,08	6,4	13,4	554	96
August	27,7	54,9	35,9	0,58	1,18	5,9	19,5	278	520
September	19,8	99,7	8,7	0,55	1,21	4,4	27,9	17	1664
October	15,8	--	--	--	1,23	--	16,5	--	405
November	4,1	--	--	--	1,20	--	31,9	--	7593
December	14,2	--	--	--	1,18	--	28,1	--	6917
Annual	**22,8**	**64,7**	**3,4**	**0,58**	**1,17**	**6,1**	**29,7**	**1998**	**25709**

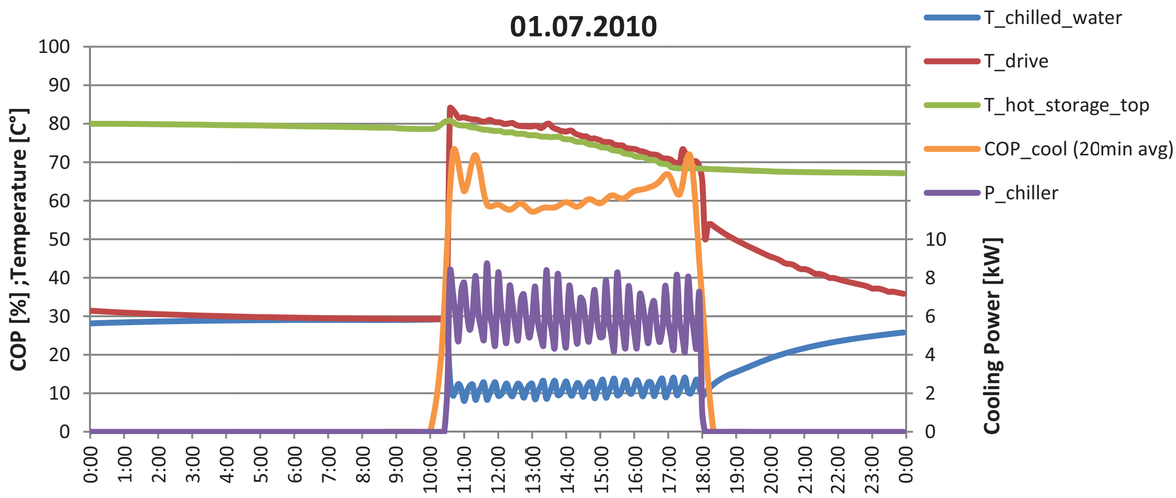
Table 1: Monthly and annual balances of system operation with ACS08 in 2010.

5.2 Analysis of Typical Days

Cooling Mode:







of the
minute

Heating Mode

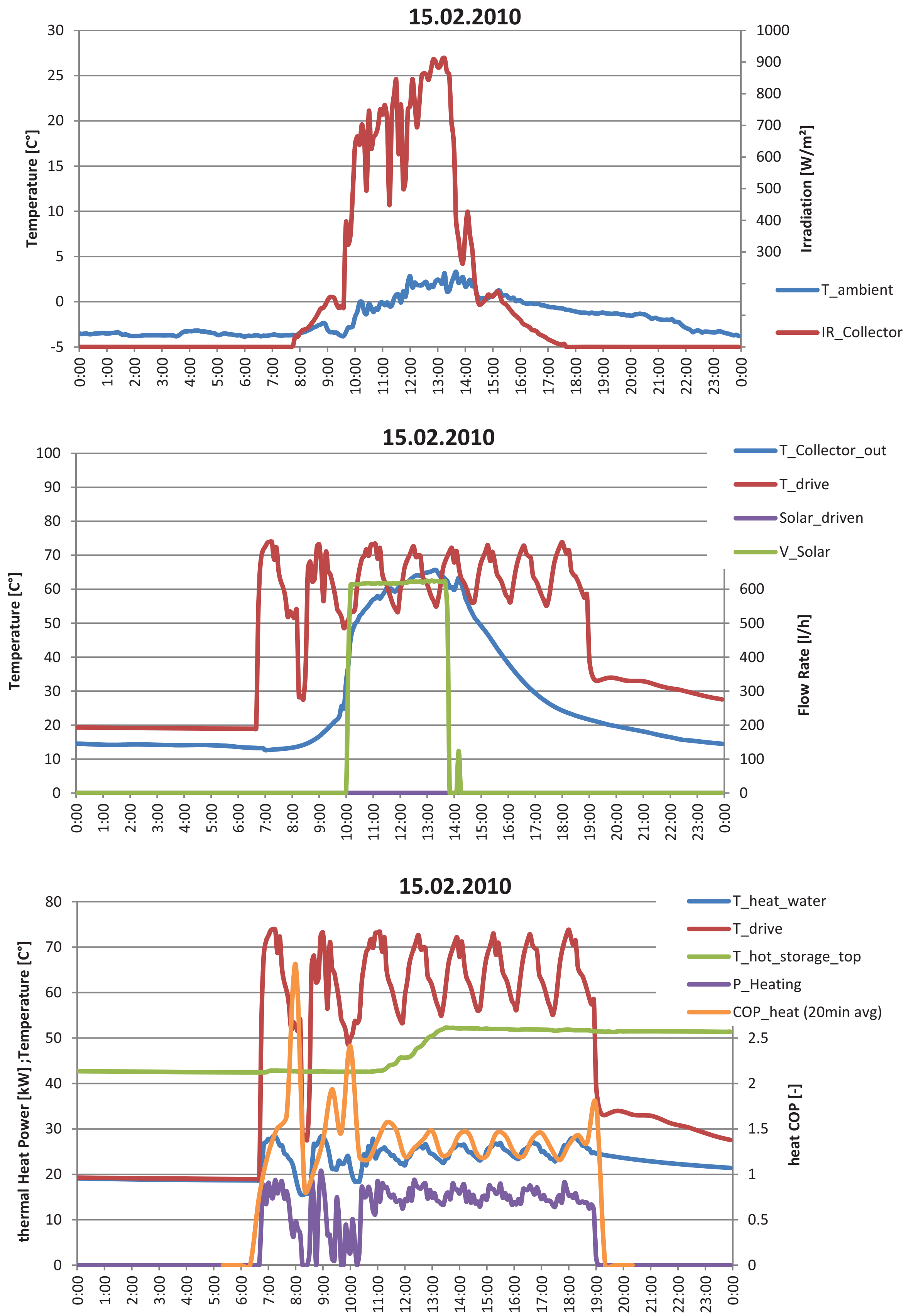


Figure 11: From the Irradiation on the collector field up to the heat production of the current system (ACS08) at the 15th of February 2010. Instantaneous values recorded in 5 minute time steps; COP is displayed in 20min average for a better visibility

5.3 Detailed Analysis – electrical COP

The following charts compare the electrical COP (ECOP) about the different generations of the chiller. The hydraulic pipe design of the solar cooling system is based on the thermal cooling capacity of the first ACS generation. Because of the increase of the chiller capacity, higher flow rates in all three circuits, specially the cooling circuit, have become necessary. Due to the increasing pressure drop of the existing hydraulic system by running higher flow rates the pump power have to be adjusted. Finally, after commissioning the latest Generation (ACS08), the pumps have to be replaced.

The Impact on the ECOP by installing higher pump capacities to coupe the increasing pressure drop of the hydraulic system is displayed in the following charts:

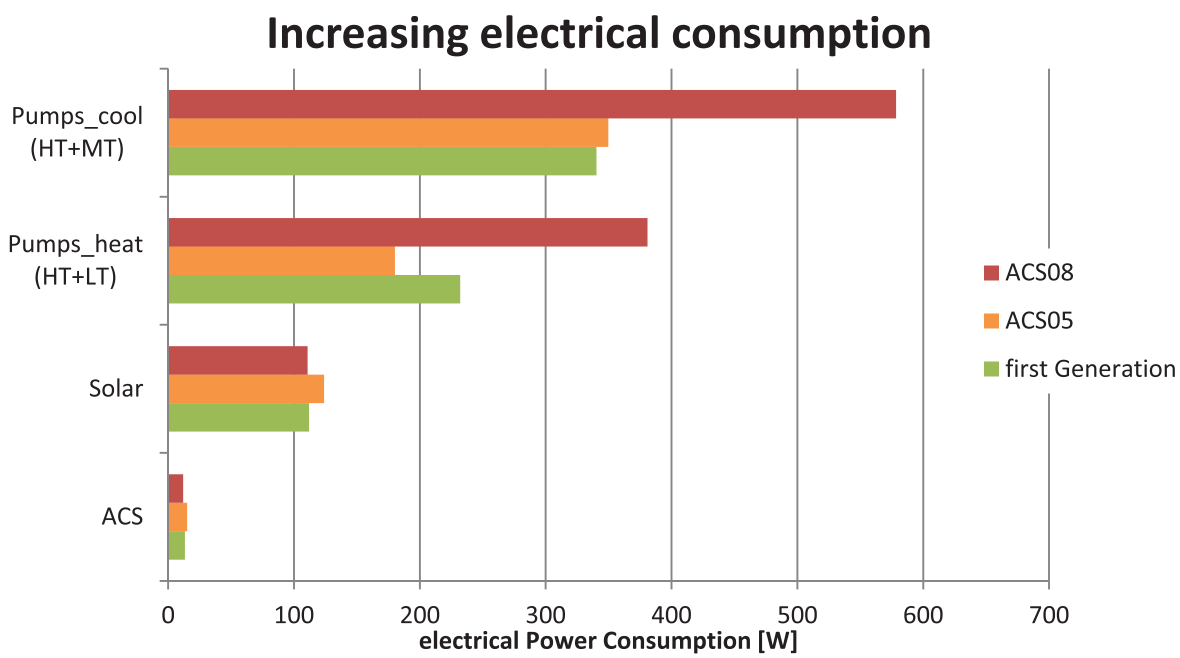
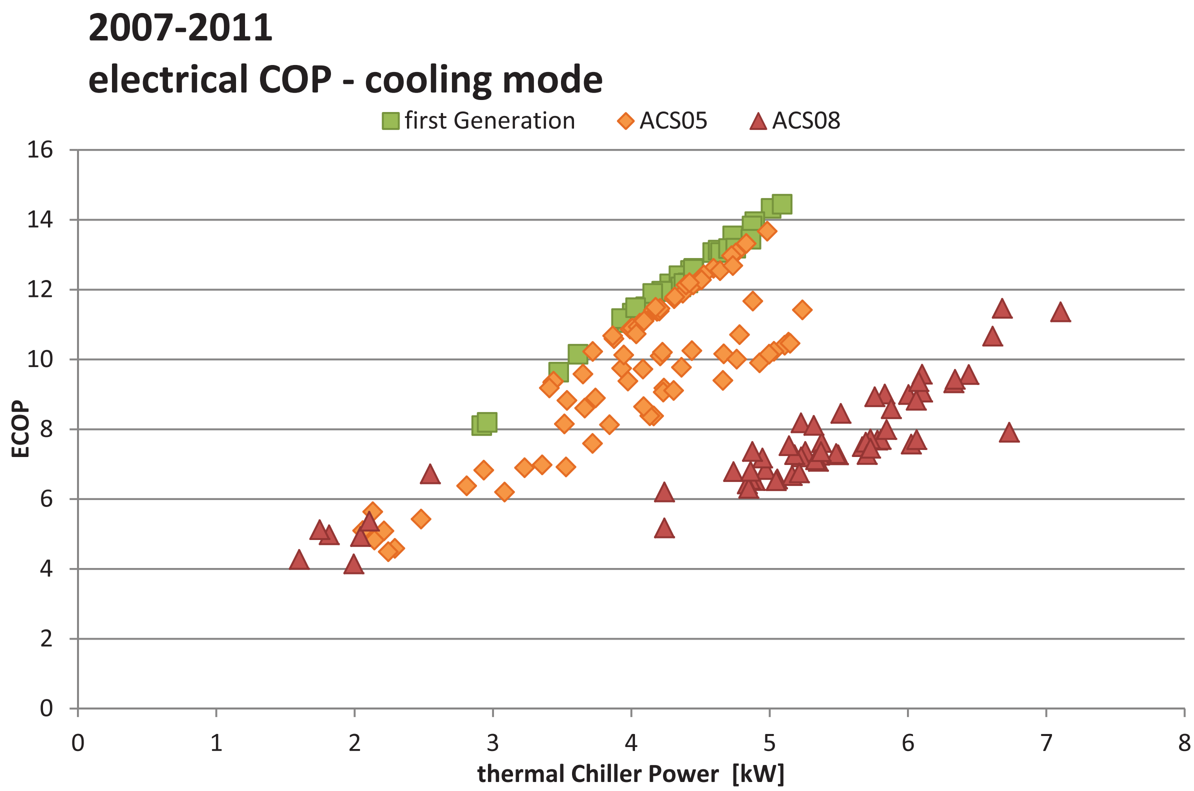


Figure 12: Electrical Power of the system components to calculate the ECOP. The power consumption of the distribution pump in Heating Mode (MT Pump) and in Cooling Mode (LT Pump) is excluded.

Therefore, the bars labelled ‘Pumps_cool’ contain the total electricity consumption of the system (ACS, Solar, HT+MT pumps) excluding the LT distribution pump (base for comparison with reference);
and the bars labelled ‘Pumps_heat’ contain the total electricity consumption of the system (ACS, Solar, HT+LT pumps) excluding the MT distribution pump (base for comparison with reference).



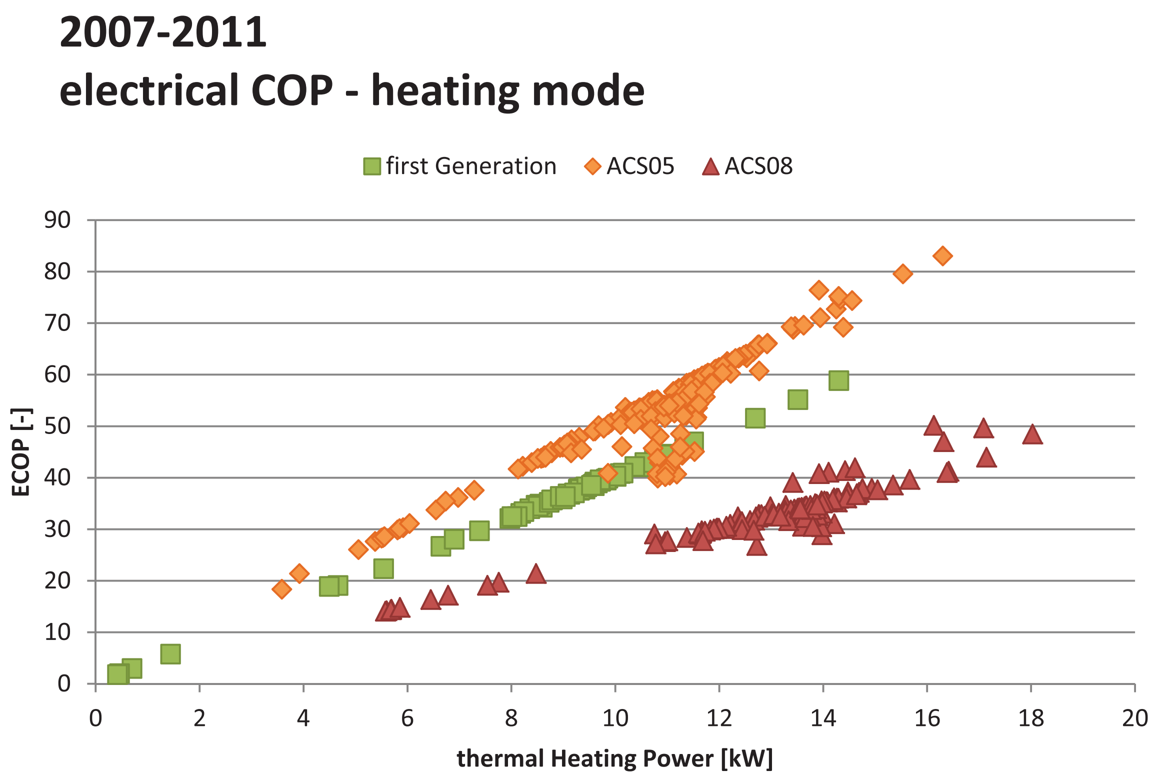


Figure 13: Hourly average values of chiller capacity versus electrical COP. Electricity consumption in cooling mode includes: solar circuit, ACS controller, pumps driving and heat rejection circuit;

Electricity consumption in heating mode includes: solar circuit, ACS controller, pumps driving and low temperature supply circuit

6 Experiences / Lessons Learned

Due to the fact that the existing collector system design (22m² Collector Area/2000 l Storage) is based on the hot water demand of the ISE canteen, it is undersized to for this solar cooling system. Because of the storage threshold temperature of 73°C minimum, before the stored thermal energy can be utilized by driving the adsorption chiller, often it takes several hours to reach the needed operating temperature. Lowering the threshold temperature as well as resize the collector area will improve the performance of the collector field.

In terms of the electrical COP, the replacement of the chiller by its latest generation, the actual hydraulic system design has become literally the bottle neck of the current system capacity of 8kW cooling capacity. Higher flow rates in all three circuits have increased the pressure drop of the pipe system followed by a higher electricity consumption of the pumps. The replacement of certain hydraulic valves as well as simplifying the hydraulic measurement equipment will make an improvement on the electricity consumption of the system. Also to change the control strategy of the MT-Pump by keeping outlet temperature of the heat rejection circuit at the chiller on an appropriate level will cut the electricity consumption by several Watts.

Driving temperature of the back up circuit (heating network of ISE) has not been stable as expected after commissioning the solar cooling system. Especially in the winter season the driving temperature is alternating from below 60°C to the required 70°C (see figure 11). The solar cooling system performance is following this unstable driving temperature behavior. An additional subroutine in the control program, which is monitoring the Back up temperature and stop the system, if the temperature drops, will cut inappropriate system operation.

7 Conclusions

The solar cooling system is operating since 5 years. During this long period valuable data about solar cooling combined with terrestrial heat probes has been recorded. The system design has proven a reliable operation in cooling as well in heat mode.

With the exception of changing the chillers by later models, the system design has not been modified and is still operating with a satisfying performance. Due to the fact that the control program / control strategy is still the same since the commission in 2006, some improvements could be implemented to increase the system performance.

The combination of a solar cooling system with terrestrial heat probes instead of a common cooling tower has shown an advantage in terms of stable heat rejection temperatures on a very low level of approx. 23 °C. Further more the overall system electricity consumption has been reduced due to the replacement of an active cooling device by those heat probes. At least for the first two generations of chillers an electrical COP in cooling mode from above 11 could be reached.

With the current hydraulic system design, based on the capacity of the first generation of the adsorption chiller in combination with the increased thermal capacity of the ACS08, the ECOP has decreased below 8 due to required higher flow rates, causing rise of the pipe resistance, followed by a higher pump capacity.

In relation to the increasing chiller capacity during the replacements, the heat load upon the terrestrial probes also went up. But as a result of the higher load the outlet temperature of the probes does not rise. This fact leads to the assumption that the capacity of the terrestrial heat probes has not reached its limit. The conclusion that the probes about three 80m deep boreholes are oversized is not approved yet, but could be expected.

A thermal storage effect at the boreholes could not be observed during the seasonal operation between winter and summer. There might be a slight shift of thermal energy during the alternating operation in spring and autumn, but is not proven yet.

8 Bibliography

Dr. Tomas Núñez; Björn Nienborg; York Tiedtke: Solare Kühlung kleiner Leistung mit Rückkühlung über Erdsonden. Contribution to 18. Symposium Thermische Solarenergie - Otti 2008, Bad Staffelstein, Germany

Dr. Tomas Núñez; Björn Nienborg; York Tiedtke: Heating and Cooling with a Small Scale Solar Driven Adsorption Chiller Combined with a Borehole System. Contribution to EuroSun 2008 Lisboa, Portugal

Dr. Tomas Núñez; Florian Mehling: Heating and Cooling with a Small Scale Solar Driven Adsorption Chiller Combined with a Borehole System - Recent Results - . Contribution to 3rd International Conference SAC, Otti 2009, Palermo, Italy