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Practical experience of two small scale solar cooling plants and cost comparison to PV driven compression chillers

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als Diplomarbeit der Kommission zur Abhaltung der dritten Diplomprüfung der Studienrichtung Wirtschaftsingenieurwesen-Maschinenbau an der Technischen Universität Graz über das Institut für Wärmetechnik vorgelegt.

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Graz. December 2009

STATUTORY DECLARATION

I declare that I, Thomas Weissensteiner, have authored this thesis independently, that I have not used other than the declared sources / resources, and that I have explicitly marked all material which has been quoted either literally or by content from the used sources

Graz, December 2009

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ABSTRACT

Title: Practical experience of two small scale solar cooling plants and cost comparison to PV driven compression chillers

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1st keyword: Solar cooling
2nd keyword: Monitoring
3rd keyword: Practical experience

Cooling and air conditioning is one of the major energy consuming services in buildings. Due to the ongoing worldwide increase of electricity consumed by conventional air conditioning units the renewable technology solar cooling gained much attention in the previous years.

In this work two small scale solar thermal cooling plants, located in Austria, are presented. Monitoring results measured on IEA SHC Task 38 level 3 basis are shown. Thus all practical experiences that were gained during summer 2009 are shared and described in this work. Most of the monitoring results are astonishing poor; especially the electrical COPs could not reach expected values. The highest total monthly electrical COP of the Solid plant in Graz could only reach a value of 1.87. At the Bachler plant in Gröbming average electrical COPs of 3.09 were measured between August and September 2009. Furthermore several suggestions of improvement are provided and some of them have already been implemented. Moreover a cost comparison between the two solar thermal cooling plants and photovoltaic (PV) driven compression chillers was carried out. The results show 4 to 7 times higher cold production costs (€/kWh) of the solar thermal plants compared to the PV driven systems and even 11 to 19 times higher production costs compared to the conventional compression chiller variant. These facts show that the economical performances of the two solar thermal systems in Austria are currently not competitive to other cooling systems.

KURZFASSUNG

Titel: Praktische Erfahrungen mit zwei solar thermischen Kühlsystemen und Kostenvergleich zu Photovoltaik basierenden Kühlsystemen im kleinen Leistungsbereich

Autor: Thomas Weissensteiner

1. Stichwort: Solare Kühlung
2. Stichwort: Monitoring
3. Stichwort: Praktische Erfahrungen

Kühlen und Klimatisieren zählen zu den größten Energieverbrauchern in Gebäuden. Durch den weltweit steigenden Stromverbrauch für Klimaanlagen richtet sich in den letzen Jahren verstärktes Interesse auf die erneuerbare solare Kühltechnologie.

In dieser Arbeit werden zwei in Österreich installierte solarthermische Kühlsysteme aus dem kleinen Leistungsbereich vorgestellt. Die nach dem standardisierten IEA SHC Task 38 Level 3 Monitoring durchgeführten Messungen werden ebenso wie praktische Erfahrungen. die im Sommer 2009 gewonnen werden konnten. beschrieben. Ein Großteil der Ergebnisse ist überaschenderweise schlecht ausgefallen. Insbesondere konnten die elektrischen Leistungszahlen die Erwartungen nicht erfüllen. Die höchste monatliche elektrische Leistungszahl. die in der Solid Anlage in Graz gemessen wurde. betrug 1,87 (August 2009). Die Bachler Anlage in Gröbming erreichte zwischen August und September 2009 durchschnittliche elektrische COPs von 3,09. Weiters wurden Verbesserungsvorschläge für die nächste Kühlperiode ausgearbeitet und bereits teilweise umgesetzt. Den letzen Teil der Arbeit bildet ein Kostenvergleich zwischen den bestehenden Anlagen und Photovoltaik (PV) basierenden Kühlsystemen. Die Ergebnisse zeigen 4- bis 7-mal höhere Kältegestehungskosten (kWh) der vermessenen solarthermischen Kühlsysteme gegenüber der PV Variante und sogar 11- bis 19-mal höhere Kosten gegenüber konventionellen elektrisch betriebenen Systemen. Dies zeigt. dass solarthermische Kühlsysteme im kleinen Leistungsbereich in Österreich derzeit ökonomisch nicht konkurrenzfähig sind.

# ACKNOWLEDGEMENT

I am particularly grateful to my supervisor Ao.Univ.-Prof. Dipl.-Ing. Dr.techn. Wolfgang Streicher for giving me the opportunity to work in the interesting field of renewable energies. He made a lot of efforts to support me in writing this thesis.

Furthermore I want to thank Dipl.-Ing. (FH) Daniel Neyer. Dipl.-Ing. (FH) Dr.techn. Andreas Heinz and all employees of the Institute of Thermal Engineering at the Graz University of Technology who helped and supported me in any way.

Also a great thanks to Werner Pink and Christian Halmdienst from the company Pink as well as Nicole Olsacher from the company Solid for their professional and friendly co-operation.

Finally I want to mention my parents, my sister and my girlfriend Sophie. Thank you for encouraging me in an incredible way during the whole five years of my studies.

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1 INTRODUCTION

1.1 Why cooling and air-conditioning?

An important factor for people to feel comfortable in buildings is a pleasant room climate. The two main parameters for a comfortable condition are the air humidity and the room temperature. To supply rooms with clean, fresh and well tempered air is the main goal of the HVAC (Heating, Ventilation & Air Conditioning) technology. Every kind of active HVAC-facility increases the energy consumption, the investment- and the operating costs of the building where it is installed. The reduction of the needed cooling load has to be a main goal of every building design. Cooling and air conditioning will play an important role in improving living standards especially in countries of the southern climates, but will also consume a significant part of the world's energy production. Therefore the finding of sustainable and renewable ways to reach the cooling and air conditioning demand of today and tomorrow is of great importance.

1.1.1 Cooling loads

In order to coordinate cooling supply and demand of a building the cooling loads are of great interest. Basically the cooling load consists of a sensible and a latent part. The sensible part itself consists of internal and external cooling loads; they are drawn simplified in Figure 1.

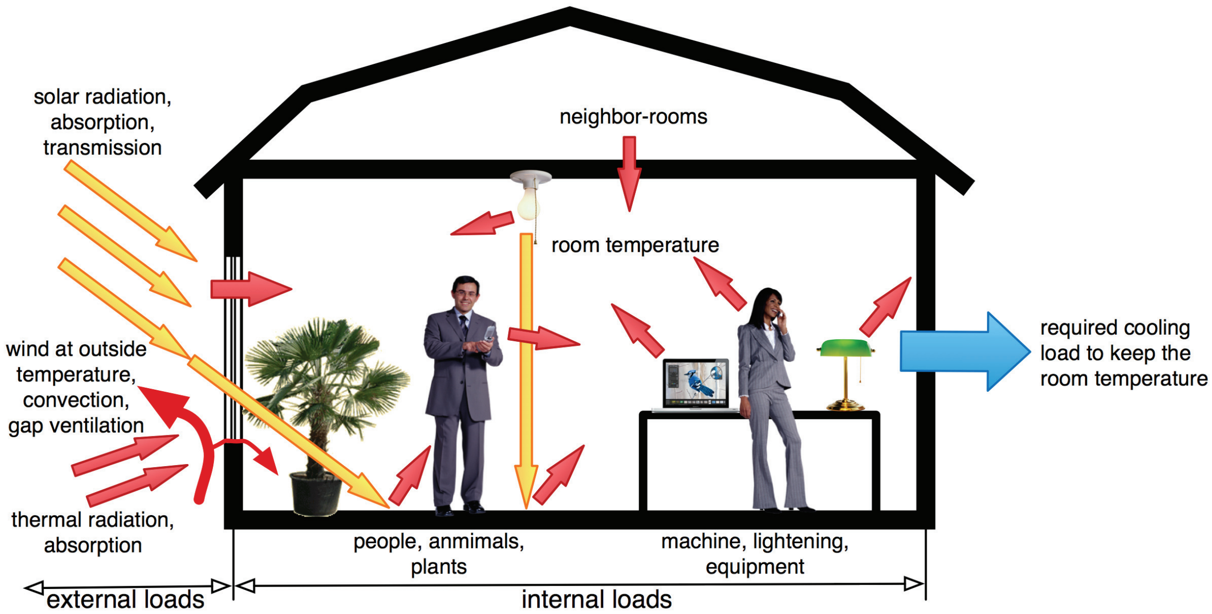


Figure 1: Simplified reasons for cooling loads in buildings (Henning, et al., 2009)

External loads are solar radiation through windows or other transparent objects, the thermal radiation absorbed by and transferred through the walls and the gap ventilation with warm outside air. Internal loads are heat sources like people, animals, plants as well as any kind of machines, equipment and lightening. Together with heat fluxes from neighboring rooms they cause an increase of the room temperature. To keep the room temperature at a comfortable level a certain cooling load has to be distributed to the room.

The latent part of the cooling load is caused through water evaporation from people or other moisture sources like plants or kitchen sinks. The total cooling load results of adding the sensible and the latent cooling load. The total cooling load defines the type, the size and also the operation conditions of the HVAC device for a specific building. To lower the external and internal loads a long list of measures exists, but just a few important such as intelligent solar building design (orientation. window area. building shading. thermal mass and night ventilation), active shadowing actions, thermal insulation and equipment with low electricity consumption shall be mentioned here. Especially in the temperate climate zone a high percentage of buildings can be cooled through an intelligent building design. By taking advantage of the day and night temperature variability passive cooling is possible (Mach, et al., 2008). In the following section the most common active cooling methods are introduced.

1.1.2 Methods

To cover the cooling demand of a building a big variety of methods are known. Principally there are two main ways to maintain a certain room climate, either to produce chilled water or to condition the air directly. To achieve this goal basically two possibilities exist:

- use of ambient heat sinks like outer air, ground/soil etc.
- use of machines or open thermodynamically processes

The use of ambient heat sinks can be also combined with methods that are able to use those heat sinks. In Table 1 a list of mechanical chilling processes is provided. Not all of these methods are used in practice respectively widespread on the market.

Table 1: List of methods used for mechanical cold production

Compression chiller	Mechanical compression with: -piston-compressor -screw-compressor -scroll-compressor -turbo-compressor
Sorption processes	-absorption -adsorption
Steam jet process	
Evaporative cooling	e.g. with open cooling tower
Special processes	-Stirling process -Linde method -Peltier effect etc.

Both of the two solar cooling plants that are monitored and investigated in this work use the absorption process in their chillers. Therefore in chapter 2 the main focus will be set to chillers working with the absorption process.

1.1.3 Cooling demand & conventional cooling technologies

Conventional cooling and air conditioning technologies use electrical driven compression chillers. These compression chillers have a higher energy demand the warmer the cooled air gets. In Figure 2 the domestic hot water (DHW), space heating (SH) and cooling demand of a typical Central European building is shown. The shaded area indicates a possible solar collector yield. The diagram demonstrates that there is a great potential to cover the lion's share of the cooling demand with solar energy.

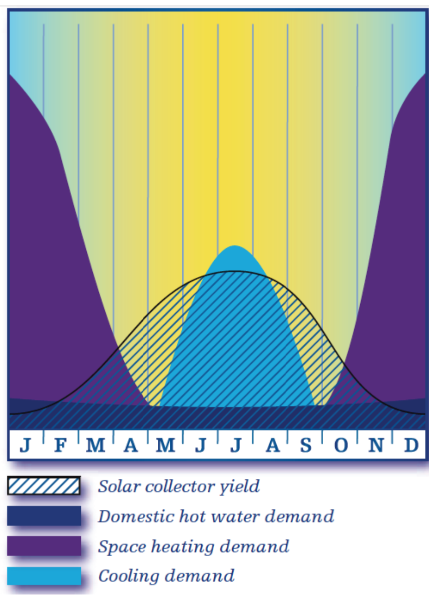


Figure 2: Cooling, space heating and domestic hot water demand of a typical central Europe building(ESTIF, 2007)

In the year 2002 in Germany around 79000 GWh/a were needed for technical cold production. Approximately 26% of the total was used for air-conditioning in buildings (DKV, 2002). That is to say around 21000 GWh/a. observing a strong upward trend. Main reasons for the upward trend are growing comfort standards. rising internal loads in buildings (electronically equipment), architectural trends to transparent facades and a higher number of operations hours due to longer shopping hours (Henning, et al., 2009). Conventional systems generally are divided into Room Air Conditioning (RAC) units and Central Air Conditioning (CAC) units. Figure 3 shows the situation of the conventional RAC and Portable Air Conditioning (PAC) unit market worldwide. The biggest cooling markets are China, the USA. Japan. South-East Asia and Europe and the world’s cooling demand is growing. Estimations within the EECCAC (2003) final report indicate a growth of the worldwide cooled area in the next 5 years of approximately 20%.

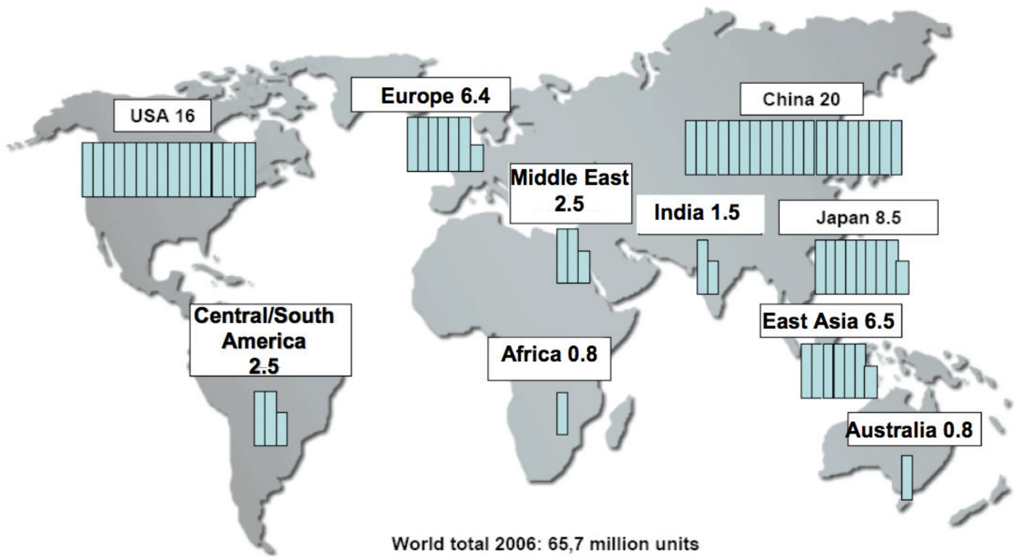


Figure 3: The conventional cooling (RAC/PAC) market worldwide in 2006 in millions (JARN, 2006)

1.2 Reasons for solar cooling

Stated in a special solar cooling report of the German *Bundesministerium für Umwelt. Naturschutz und Reaktorsicherheit* there are three main reasons for solar cooling. Primary energy savings compared to electrical air condition, reduction of the burden on the electrical grid at peak times and the avoidance of high temperatures and non-use phases of solar thermal collectors in summer time (Clausen, 2007). These three benefits are described in detail in the next paragraphs.

Primary energy savings

Primary energy savings for solar cooling plants compared to conventional compression chillers can be achieved through the low electrical energy consumption cased only by the auxiliary pumps and fans of a solar cooling plant. Henning (2007) estimates the primary energy saving for a properly designed chilled water system between 40 and 60% and of properly designed open cycles between 20 and 50%. Some of these estimations have been confirmed already through demonstration plants. For instance a well designed chilled water system is operating in Garching at the ZAE Bayern providing promising results (Helm, et al., 2009).

Unburden the electrical grid

Another important benefit that can be achieved by using solar energy for cooling is the possibility to unburden the electrical grid at peak times. The consumption of electricity can be reduced especially on hot and sunny summer days. In North America the peak electricity production was in the year 2007 in August (approximately 500 TWh) (IEA, 2009). The total peak load for an average weekday in August due to RAC in Europe is computed to about 27 GW (EERAC-Team, 1999). A remarkable part of this peak could be shaved with the solar cooling technology. To uncouple the cooling load from the electrical load should be a main goal for the future.

Unburden the solar thermal collectors

In Austria about 15% of the detached houses already use solar thermal (ESTIF, 2007). In summer time the collectors often suffer high temperatures and non-use phases which lead to higher mechanical stress and shorter lifetime. The high offer of solar energy of this period can't be used fully, due to a lack of demand. As shown in Figure 2 the cooling demand of the building fits perfectly with the solar yield. By using the same solar collectors for heating, cooling and DHW the usable solar yield can be optimized and the lifetime of those collectors can be extended. The concurrent demand and supply of the cooling loads is another big benefit of the solar cooling technology. Storage problems do not occur in summer and winter in a properly designed system because the cooling load is produced at nearly the same time as it is needed.

One final reason for solar cooling is today's political framework in Europe and the USA. The European Union including Austria has committed itself to reduce its overall greenhouse gas emissions to at least 20% below 1990 levels by 2020. Also a 20% share for renewable energy should be established as well as improving the energy efficiency by 20% *(European Parliament, 2008)*. Figure 4 shows the estimated growth of the CO_2-Emissions due to RAC from 1990 to 2020 in Austria. Solar cooling contributes to two of the three goals. It can help to reduce the greenhouse gas emissions by substituting conventional air conditioning systems and increases the renewable energy share.

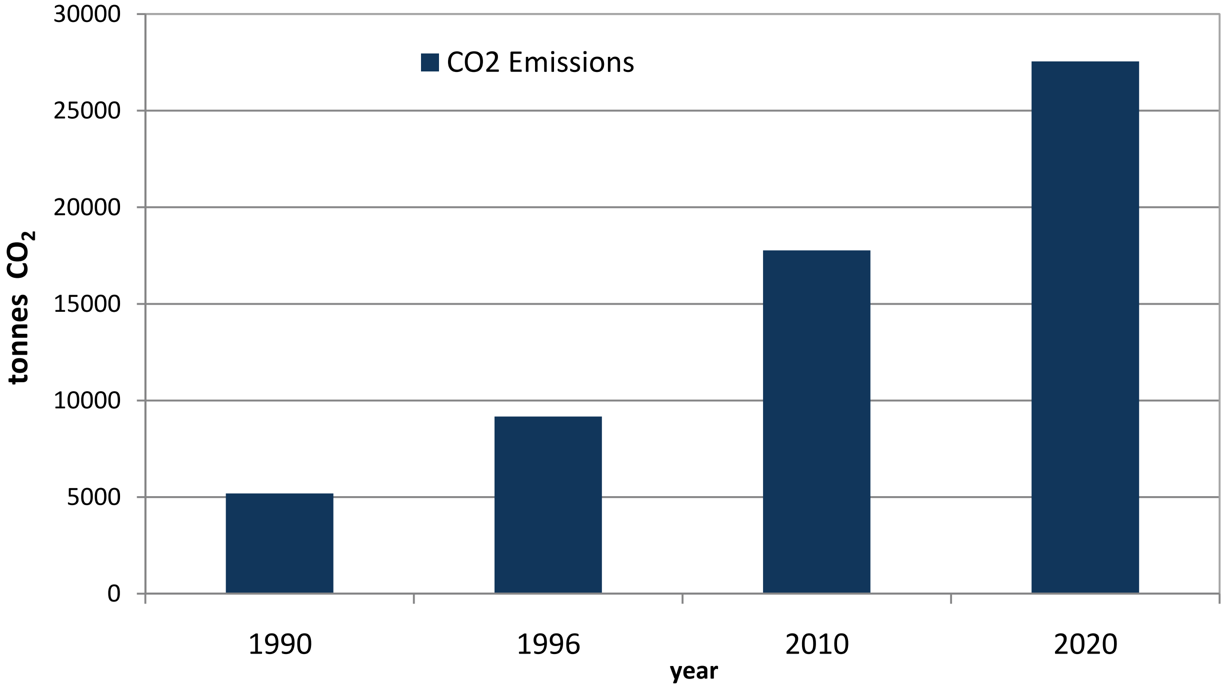


Figure 4: Emissions due to room air-conditioning (RAC) devices in Austria (EERAC-Team, 1999)

A sustainable market for solar cooling will be of great importance in the near future especially for countries with intense solar radiation. Summarized, the use of solar energy for cooling and air conditioning can save primary energy, shave peak loads of the electrical grid and extend the lifetime of existing collectors. In this way it can help countries to fulfill their energy and environmental goals which can be seen as one of the biggest advantages of the renewable technology solar cooling. Some disadvantages and reasons for problems of market penetration shall be discussed later in this work. Also proposals are provided to address these issues. Before introducing the basic technical principles of solar cooling in chapter 2, a short overview of the history of solar cooling as well as a state of the art market overview shall be provided in the following section.

1.3 History of solar cooling

Prior to the 1970s, hardly any research and entrepreneurship focus has been put on solar cooling. In the late 70s the first commercial, single-effect H_2O/LiBr ACM for solar cooling hit the market. About 100 demonstration plants were installed in the USA (Loewer, 1978). Due to the poor market success the production was stopped and the license was given to the Japanese company Yasaki. The market in the US flopped because of the high investment costs for solar cooling (Jakob, 2005). Driving heat temperatures were at this time in the range of 82-90°C for recooling water temperatures of 28-29°C and chilled water temperatures of 7 °C (Lamp, et al.,

1997). In the early 1990s Yasaki offered 5-10 kW cooling power chillers which were used for solar cooling projects. Production was stopped again as a consequence of missing demand.
In the last years the absorption technology in the small and medium size field gained increasing interest, especially from European companies. ClimateWell from Sweden had sales of 3.7 million Euro in the year 2008 and has already booked orders for more than 1000 machines (Climatewell, 2009). Also a number of German manufactures are active in the sector of small scale ab-/adsorption chillers. In the year 2008, 450-500 solar cooling systems have been realized in total worldwide (Mugnier, 2009).

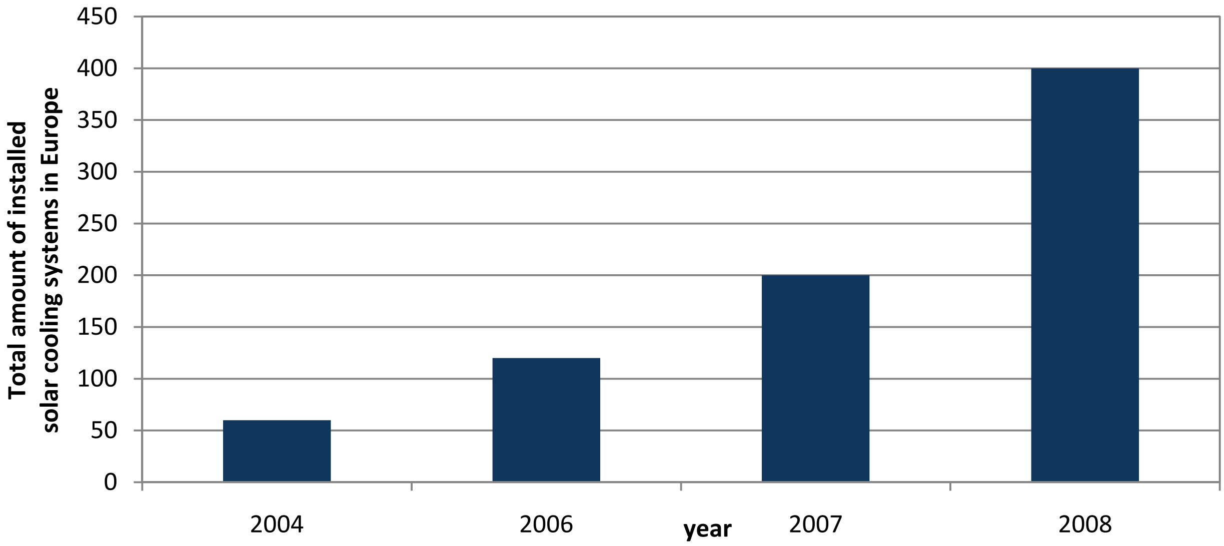


Figure 5: Market development of small and large-scale solar cooling systems in Europe (Jakob, 2009)

The market development of small and large-scale solar cooling systems in Europe is shown in Figure 5. It can be seen that the number of realized systems is growing fast. The practical experiences gained from those plants are of great value as they will play a crucial role to improve the technological maturity of the solar thermal cooling technology. Looking further into the future a market outlook for Austria is provided in Figure 6. A detailed market overview and outlook will be discussed in 1.4.

1.4 Solar cooling market overview and outlook

The most active countries in the solar cooling technology can be found in Europe. In the last year a lot of start-ups from Germany, Austria and Sweden have started to offer components for solar cooling systems or even complete systems. Some companies like Solarnext or ClimetWell are offering pre-designed systems that can be easily adapted to the particular requirement. Also big heating system companies such as Vaillant or Schüco showed some interest in this field by building demonstration plants on their own. In Figure 5 the development of the solar cooling market in Europe is presented. Most of the installed systems that are in operation are demonstration plants which are often governmental subsidized.

One trend in the last years leads towards solar cooling systems beyond the 500 kW cooling load level. For instance the Austrian company Solid commissioned 2009 a 545 kW cooling power plant in Lisbon/Portugal and has announced a 1.5 MW plant in Singapore (Solid, 2009). Large solar cooling systems can reduce costs per kW and can reach cost competiveness to conventional systems more easily (Eicker, et al., 2009).

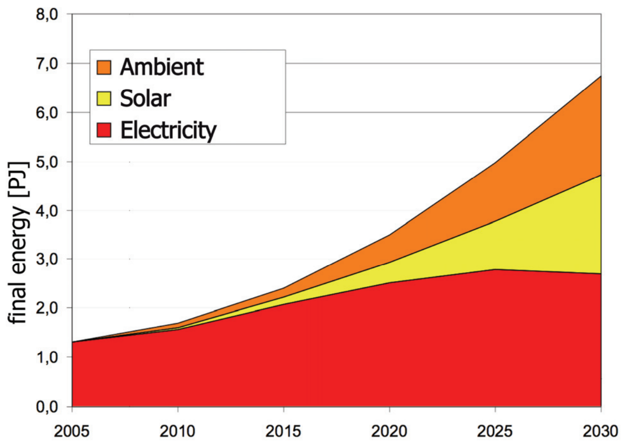


Figure 6: Outlook of cooling technologies in Austria (Haas, et al., 2007)

A market outlook is very difficult to predict and most manufactures maintain a low profile with future sales expectations. Never the less Clausen (2007) assessed the world market volume in 2020 between 4.5 and 18 billion Euro. These assumptions depend on the increase of total sold cooling systems (conventional all kind + solar cooling) and the market share of solar driven systems. If costs of small systems can be cut a broad market consisting of house owners which already own solar thermal collectors could be opened. Today 45 million households possess on-roof solar collectors for domestic hot water production. A lot of barriers to enter this market still exist today. Examples for these barriers are too high investment costs, too less standardized systems, not enough experienced system designers and installing plumbers and a low public awareness of solar cooling in general.

The scientific community is working in the International Energy Agency (IEA) Task 38 Solar Air-Conditioning and Refrigeration until 2010 together to approach those problems. They will deliver a handbook for designers and planers at the end of the task. In some countries interest groups have been formed such as Green Chillers in Germany and the Australian Solar Cooling Interest Group to bundle the forces of small companies, consultants and research institutions in fields like lobbying work or promotion(Jakob, 2009)(Kohlenbach, 2009). For further interest a detailed list of manufactures is attached in the appendix.

2 BASICS

In this chapter the theoretical and technical basics of the solar thermal cooling technology are outlined. All information needed to interpret the monitored results are described here in a condensed way.

2.1 Basics of solar cooling and air conditioning

The main aim of applying solar energy for cooling and air conditioning is to reduce the energy consumption and make cooling more environmental-friendly. There are many different ways to use solar energy for cooling and air conditioning in buildings. Table 2 shows an overview of possible solar cooling and air conditioning principles. One main differentiation is made between electrically and thermally driven systems. The electric process is using photovoltaic modules in combination with conventional compression chiller systems. Thermal processes are divided into two main categories, primary in the heat transformation with open and closed cycles and secondly the thermo-mechanical processes such as the Rankine or the steam jet cycle. All black boxes in Table 2 indicate market available processes and all grey marked processes are methods still in R&D status. The PV driven compression chiller system consists of market available technologies. The most common process for solar thermal cooling applications is the liquid sorption closed cycle. For instance 22 of the 35 investigated plants by the Rococo study (2008), have been systems including absorption chillers (Preisler, et al., 2008).

Table 2: Overview off physical ways to convert solar radiation into cooling or air conditioning (Henning, 2004). Meaning of the marks: Market available processes Processes at R&D status

<table>
<tr><td colspan="11">solar radiation</td></tr>
<tr><td>electric process (photo-voltaic)</td><td colspan="10">thermal process (solar collectors)</td></tr>
<tr><td rowspan="4">photovoltaic-compression</td><td colspan="7">heat transformation</td><td colspan="3">thermo-mechanical processes</td></tr>
<tr><td colspan="3">open cycles</td><td colspan="4">closed cycles</td><td rowspan="3">Rankine-cycle/com-pression</td><td rowspan="3">Vuilleu-mier cycle</td><td rowspan="3">low temperature steam jet cycle</td></tr>
<tr><td colspan="2">solid sorption</td><td rowspan="2">liquids (hygros-copic solution)</td><td colspan="2">liquid sorption</td><td colspan="2">solid sorption</td></tr>
<tr><td>dehumidifier rotors</td><td>fix bed process</td><td>water/ lithium-bromide</td><td>ammonia /water</td><td>adsorption (silica gel)</td><td>dry absorp-tion (e.g. ammonia-salt)</td></tr>
</table>

In this work a focus is set to the two liquid sorption closed cycles working either with water / lithium-bromide or ammonia / water, because the two monitored plants have such systems installed. Thermo-mechanical processes do not have a big relevance for practical solar cooling applications in buildings right now. Nevertheless some R&D effort is put into this field, for instance into the low temperature steam jet cycle (Pollerberg, et al., 2009). Furthermore in the last chapter of this work the economical performance of the built plants is compared to an electrically driven system.

The general scheme (Figure 7) of solar thermal cooling systems covered in this work includes the solar collectors, the thermal driven cooling process (covering a heat/cold-storage. an absorption chiller and the recooling unit) and a building which is cooled with chilled water.

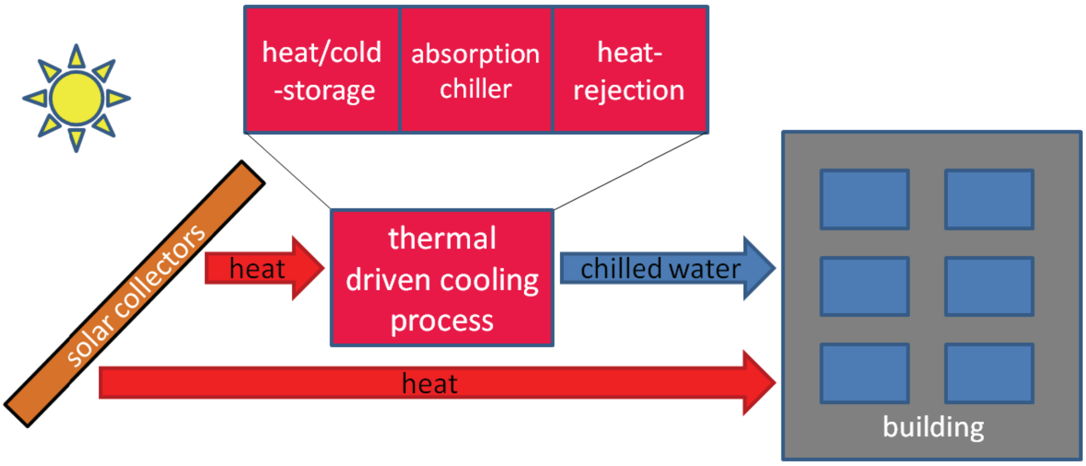


Figure 7: General solar thermal cooling process for systems covered in this work

The solar collectors convert solar radiation into heat, brought as an input to the thermal driven cooling process. It has to be stated that this figure just shows the system scheme of solar cooling systems covered in this work. In most cases water or water / antifreeze fluid mixtures are used as a heat transfer medium. Then it is possible to use the heat in times were no cooling is needed (winter times) to support the space heating and the domestic hot water production. This is especially the case if the thermal driven cooling process is combined with a solar thermal combisystem. Before the hot water flows into the absorption chiller it can be stored in a hot storage. The absorption chiller, driven by the hot water, produces a certain cooling load. In order to fulfill this task a heat rejection system has to be connected to the thermal driven chiller. Additionally the chilled water can be stored in a cold storage before it is distributed to the building.

Within the building there are several ways to use the chilled water for space cooling. In the two monitored systems, radiant ceilings and activated concrete cooling are applied in order to deliver the cooling load to the building. Storages that are installed in these systems are mainly to extend the working hours of the thermal driven cooling process, but basically the system is just running if the solar radiation is sufficient enough to drive the cooling process. All components in the overall system have a certain technical maturity. Therefore one major task in this technology is still the combination of those components to a joint system working with a well engineered control system.

Basic components of the plant such as the solar collectors, the storage system or the heat rejection unit are described in the following sections. General system aspects such as high flow or low flow solar hydraulic designs are discussed in point 2.6.

2.2 Solar collectors

The basic principle of active solar thermal use is the compilation of short wave solar radiation into heat through absorption of a great part of the solar radiation with a “radiation trap” (e.g. collectors). Solar thermal systems convert solar radiation partly into heat with the help of solar collectors. Afterwards this heat can be used directly through a heat transfer medium or can be stored for a posterior usage. The various collector types can be classified on the basis of the heat transfer medium (liquid or air) or on the basis of radiation absorption (concentrating and not concentrating). Figure 8 gives an overview over the most important existing collector types. In both monitored systems flat collectors were installed.

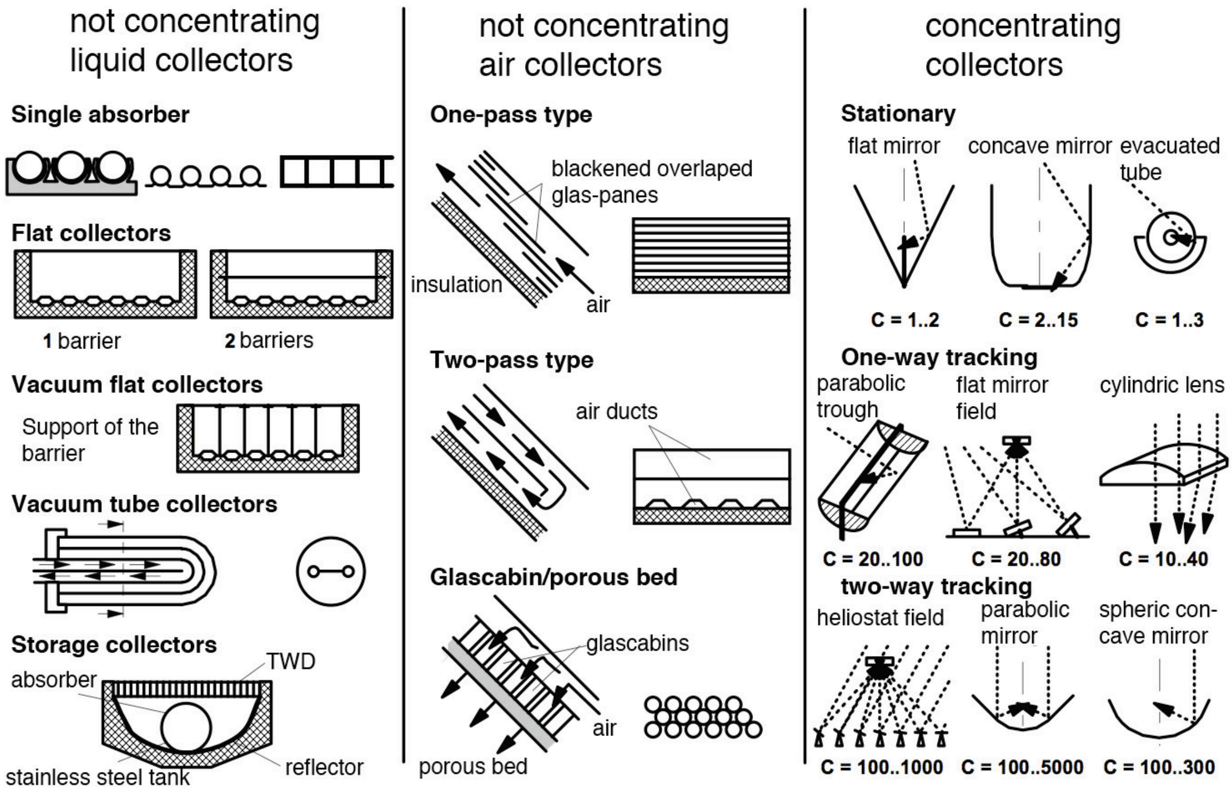


Figure 8: Overview of existing collector types (C concentrating ratio; defined as optical active collector area to radiated absorption area)(Kaltschmitt, et al., 2007)

Figure 9 illustrates a schematic build-up of a liquid not concentrating collector. Direct or diffuse (scattered) radiation is transmitted by the transparent barrier and incidents on the absorber. The absorber which is mostly made of metal or plastic coated with selective black surface absorbs the radiation and passes on the heat to the heat transfer medium. Further on the collector consists of an insulated frame equipped with fixings for the mounting. The thickness of the insulation depends on type and application of the collector. Existing convective heat losses are reduced by the integrated barrier. Often used materials are security glasses or plastic sheets and foils. Requirements for the barrier are transparency so that the solar radiation can enter the collector but imperviousness for the long wave radiation coming from the absorber. To reach higher medium temperatures in order to drive the thermal driven cooling process some collectors used for solar cooling have thicker insulations and sometimes a second transparent barrier.

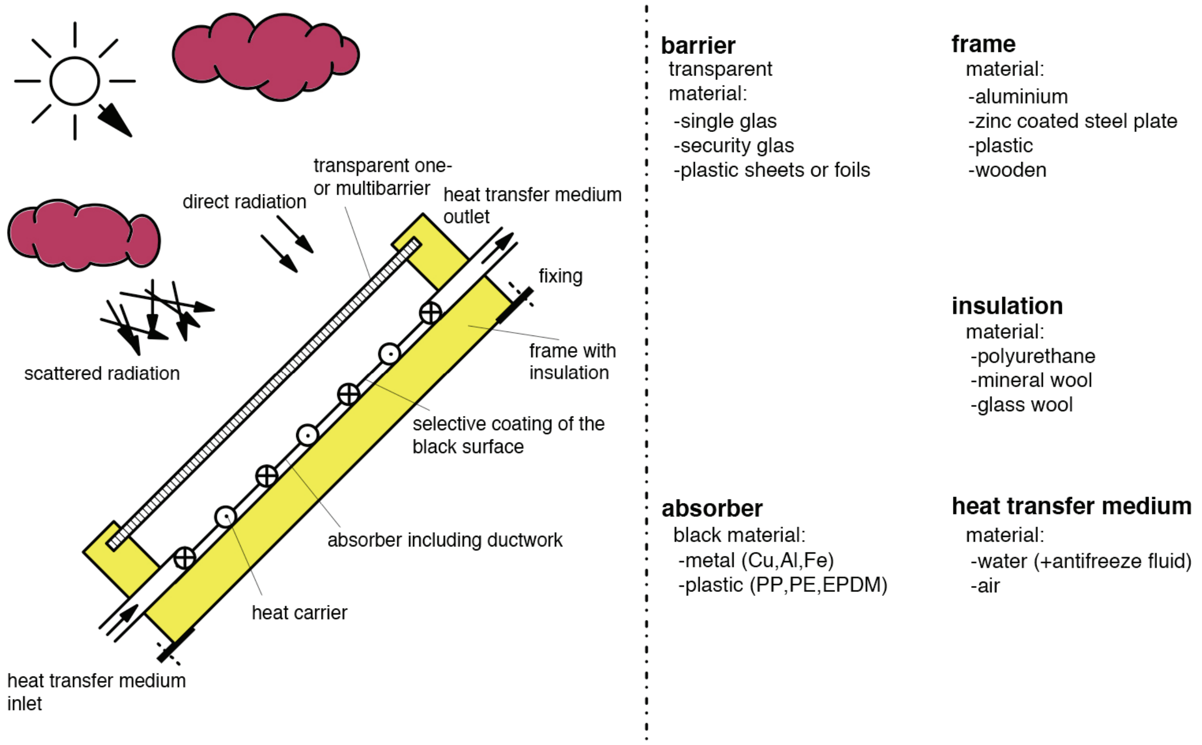


Figure 9: Schematic build-up of a not concentrating fluid collector (Kaltschmitt, et al., 2007)

Not all collector types are capable to work with any thermal driven cooling process. Figure 10 shows the approximate working areas of the different cooling processes. Following the diagram solar air collectors can work with open sorption cycles as well as with adsorption chillers. Flat plate collectors as they are built in the two monitored systems work also with single effect absorption chillers. Vacuum collectors can be additionally combined with double effect absorption chillers.

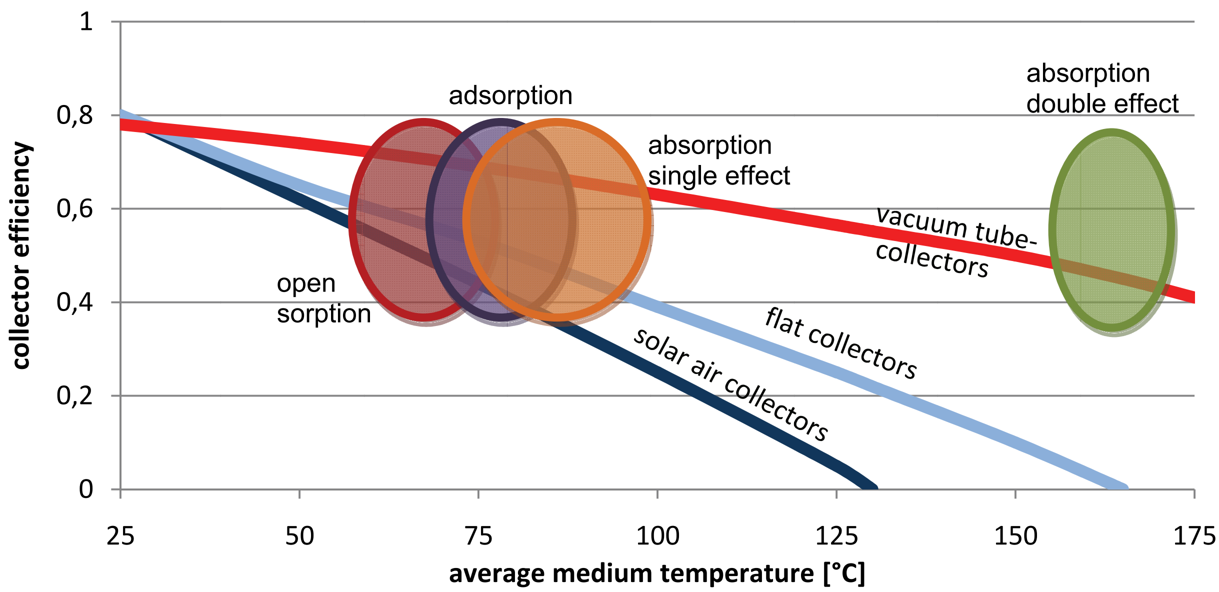


Figure 10: Typical efficiency characteristics for three stationary collector types. The curves are shown for a solar radiation of 800 W/m² and 25°C ambient temperature. Within the curves there is a dependency on the manufacturer. The diagram outlines working areas for cooling processes. (Henning, et al., 2009)

Under the next point a basic overview of possible storage technologies is provided.

2.3 Storages

To overcome mismatches between the cooling load and the solar gains most of the solar cooling plants use storage systems. Basically there are two storage possibilities as shown in Figure 11, either storing the hot water coming from the collectors in a hot water buffer storage or storing the produced chilled water. To integrate a heat storage in the heating cycle of the thermal driven cooling process is the most common way. Heat storages are possible in any solar assisted air conditioning system. Cold storages are only combinable with thermal driven chillers such as absorption or adsorption but not with open cycles such as a desiccant system.

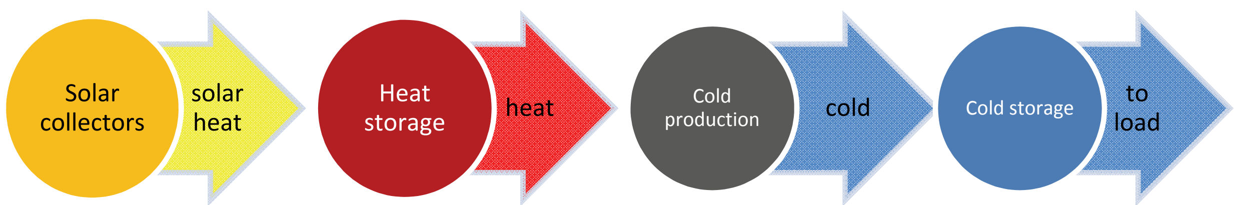


Figure 11: Places to store energy in solar thermal cooling plants (Henning, 2004)

The storages used in both monitored systems are heat storages integrated in the heating cycle. On the cold side only small (<500 liter) hydraulic switches are installed. Therefore the focus is set to hot water storages in this work.

The hot storage is one of the key components of a solar cooling plant and it has to fulfill a couple of tasks. Henning et al (2004) names five of the most important tasks:

- providing the heat sink with sufficient energy (mass flow and temperature)
- decoupling mass flows
- store heat for times where there is no heat source available
- storing the heat without mixing on a sufficient temperature level
- extending running times of auxiliary heating devices

Storage tanks have to resist the pressure of the storage medium; furthermore they have to be watertight and corrosion resistant. Used materials for such tanks are mainly steel, concrete or plastic. The storage density of water storages is given by the specific heat capacity of water. An appraisement of the storage density can be calculated following Equation 2.1.

$$q_{water} = \rho_{water} \times c_{pwater} \times \Delta T \approx 1.16 \left(\frac{kWh}{m^3 K}\right) \times \Delta T(\mathrm{K}) \qquad [\mathrm{kWh/m^3}]$$ *Equation 2.1*

For the Solid plant the dimensioned flow and return temperature to the hot storage is 90°C/75°C which evolves a storage density of 17.4 kWh/m³. The designed flow and return temperature to the cold hydraulic switch is 9°C/17°C and implicates therefore a cold storage density of approximately 9.3 kWh/m³. In most cases hot storages have higher storage densities due to higher temperature differences. On the other hand cold storages can help to reduce the chiller power in case the peak loads only occur within a few hours. Generally a decision on which storage is used depends strongly on the concrete local conditions and requirements.

2.4 Absorption chiller

The core component for thermal solar driven cooling plants is the thermal driven chiller. In this section the main mode of operation for absorption chillers is described. Absorption chillers are a widespread technique within the thermal driven chillers. Its main field of application is the cold production using waste heat from industrial processes, district heat or the waste heat of thermal power plants. The main manufacturers offer mostly chillers with a high range of performance. Table 3 lists typical characteristics of thermal driven chillers. In this table also adsorption chillers are listed in order to compare the both methods and get an overview over existing thermal driven chillers. For absorption chillers there are two main working pairs, described in 2.4.3.

Table 3: Important typical charactaristics of thermal driven chillers (list of manufacturers without a claim of completeness)(Henning, et al., 2009)

method	absorption chiller			adsorption chiller
number of effects	single effect	double effect	single effect	single effect
sorption material	lithium-bromide		water	silica gel. zeolite salt
working material	water		ammonia	water
driving temperatures	80°C-110°C	140°C-160°C	80°C-120°C	60°C-95°C
driven through	hot water (sometimes steam or direct heated)	hot water. steam. direct heated	hot water. steam. direct heated	hot water
COP	0.55-0.8	0.9-1.2	0.3-0.7	0.4-0.7
range of performance (on the market)	4.5 kW-several MW	>50 kW	10 kW-several MW	7.5-350 kW
manufacturer	York. Yasaki. EAW. Trane. Carrier. Broad .Ebara .LG Machinery. Sanyo-McQuany. Axima. Entropie. Century. SK Sonnenklima. rotartica. Thermax		*direct heated:* Robur. Colibri. AWT. Mattes *hot water or steam:* ABB. Colibri. Mattes. Pink	Mayekawa. Nishiyodo. Sortech. Invensor. ClimateWell

2.4.1 Coefficient of Performance

The key figure to characterize the energy performance of an absorption chiller is the Coefficient of Performance (COP). It shows for a produced unit of cold the required heat input (see Equation 2.2). This COP is also called thermal COP because it neglects the auxiliary electrical consumption. In order to implicate this as well the electrical COP is calculated following Equation 2.3. The values are only comparable if the same operation conditions are considered.

$$COP_{therm} = \frac{Q_O}{Q_H} = \frac{heat\ input\ to\ the\ evaporator}{heat\ input\ to\ the\ generator}$$ Equation 2.2

$$COP_{elec} = \frac{Q_O}{P_{el}} = \frac{heat\ input\ to\ the\ evaporator}{electricity\ input\ to\ the\ compressor}$$ Equation 2.3

The COPs shown in Table 3 are the thermal ones. The labeling of the heat and the electricity in in- and outputs for compression and absorption chiller cycles is shown in Figure 12.

2.4.2 Absorption cycle

Absorption systems do have the same working principles as mechanical compression systems as far as the core components evaporator and condenser are considered. The main components of a conventional compression cycle and an absorption chiller cycle are illustrated in Figure 12.

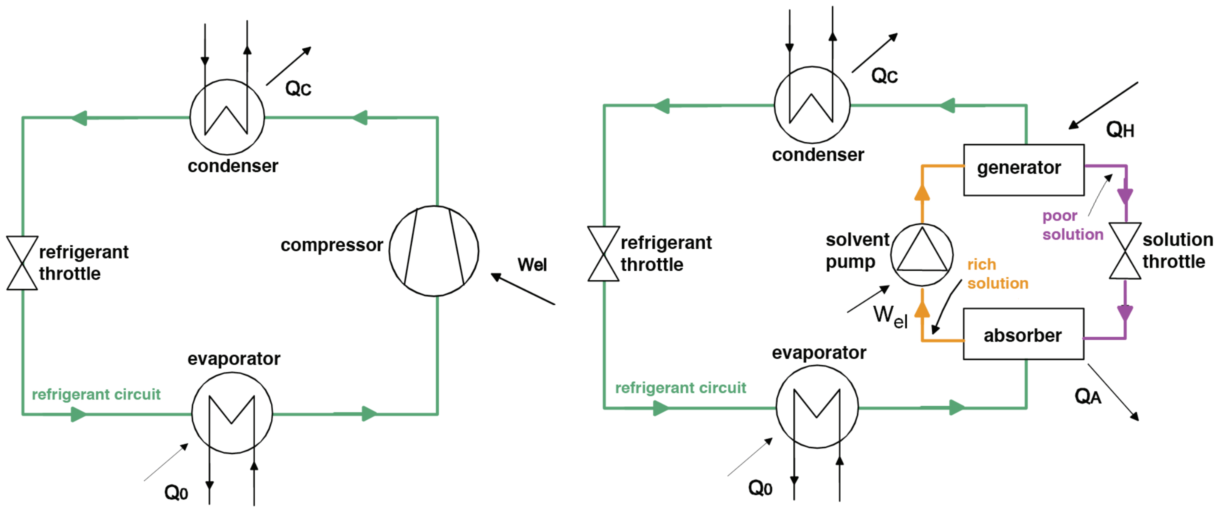


Figure 12: left: basic system scheme of a compression cooling cycle right: principle scheme of a absorption cooling process

A refrigerant (low-pressure vaporizing liquid) extracts heat from a low temperature level, which is considered as the gain of the process, the cooling. Afterwards the vapor is compressed to a higher pressure level in the compressor. In the condenser the vapor is condensed at a higher temperature level (heat dissipation). Further on the pressure of the refrigerant is reduced in the throttle and the cycle starts from the beginning. On the right side of Figure 12 the mechanical compressor is replaced by the absorption cycle which is running in clockwise direction. This sorption cycle is based on the fact that the boiling point of a pure liquid is lower than the corresponding boiling point of a mixture. In the same way as in the compression chiller, the refrigerant evaporates in the evaporator, thereby extracting heat and producing the useful cooling effect. The vapor flows into the absorber where it is absorbed in a concentrated solution.

Latent and mixing heat from the absorber have to be rejected to keep the process running. The solvent is pumped to the generator where it is heated above the boiling temperature of the refrigerant. The vaporized refrigerant is leaving the absorption cycle towards the condenser with high pressure and follows the same cycle as the compression chiller. The refrigerant poor solution flows back through the solution throttle into the absorber. The heat required for the generator can come from any source. If the driving heat is needed on a relative low temperature level then the heat from solar thermal collectors can be used. Three main temperature levels have an impact on the absorption cycle, the high temperature level (T_g) of the driving heat, the low temperature level (T_e) where the cooling load is delivered and the medium level (T_c, T_a) where the heat is rejected. These temperature levels are shown in Figure 13 in a log pressure and temperature diagram for an absorption and a compression chiller cycle.

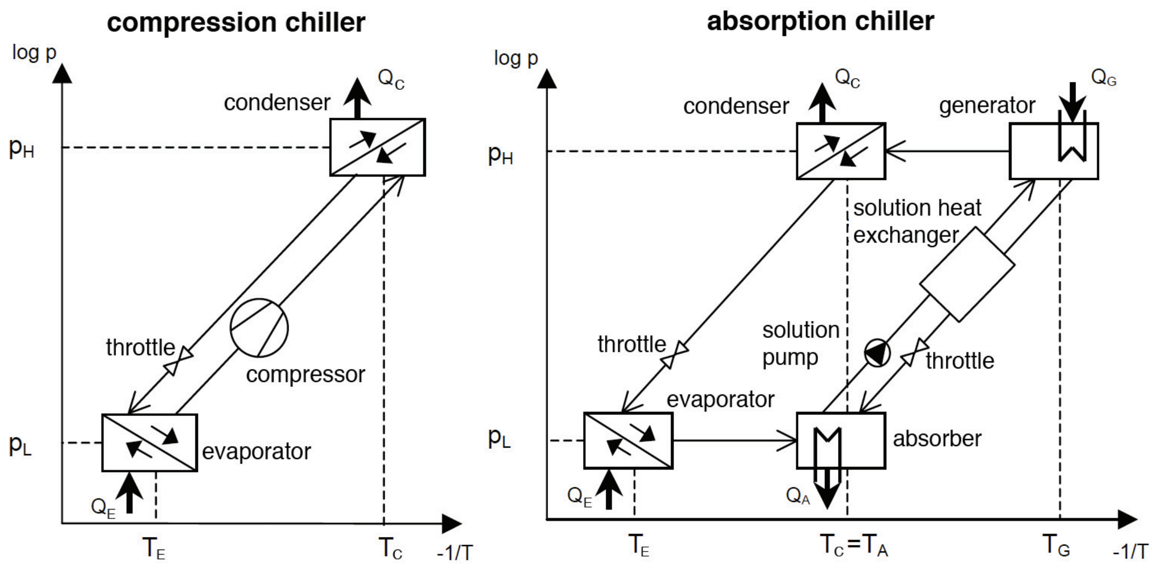


Figure 13: log pressure and temperature diagram of an absorption and a compression chiller cycle

Absorption chillers normally also need an electricity supply since a mechanical solution pump and a control system are employed. Nevertheless the general electricity consumption is normally in the range between 1% and 2.5 % of the chiller capacity and therefore negligible. Based on this fact the sorption based chiller is named thermal driven cooling process. In the following point the two most common working pairs for absorption chillers are described.

2.4.3 Working pairs

Two main working pairs for the absorption chiller cycle exist, water / lithium bromide and ammonia / water. Depending on the plant requirements one of the two can be chosen.

Water / lithium bromide:

In most cases water / lithium bromide machines are used if the cooling temperature is above 5°C. In this combination water is working as a refrigerant and the cooling process is based on evaporating water at low pressures (compare 2.4.2). The physical limit is therefore given by the freezing point of water below 0°C. Lithium bromide is soluble in water up to a mass fraction of it in the mixture of 70 %. If higher concentrations occur the lithium bromide crystallizes, which consequently requires a costly repairing of the chiller. Therefore the absorber temperature is restricted to avoid the crystallization.

Ammonia / water

Machines working with ammonia / water can be used for cooling temperatures below 5 °C. Due to the low freezing point of the refrigerant ammonia (-77.7 °C) it is possible to serve for refrigeration tasks. Ammonia is one of the most analyzed fluids in the world. Furthermore there are no crystallization problems compared to the water / lithium bromide chillers. One of the biggest disadvantages of the working pair is flammability, toxicity and the characteristic smell of ammonia which implicates higher security standard of the machine.

2.5 Heat rejection system

One main component of the whole solar cooling plant is the heat rejection system. Generally there are many possible ways to reject the heat of a thermal driven cooling process (e.g. air, ground or water). The use of ground or water for rejecting the heat has a strong dependency on local conditions. In contrast air is available for almost all applications. Furthermore in this work only ambient air heat rejection units are covered.

Compared to compression chillers, thermal driven cooling processes have to reject much more heat if they produce the same amount of cooling load. For the compression chiller the mechanical input work has an exergy ratio of 100 %. In comparison the exergy ratio of the input heat (driving heat) of the thermal driven chiller depends on the temperature. Figure 14 shows a simplified comparison of the drive between mechanical and thermal driven chillers. The illustration shows that the rejected heat for thermal driven processes is significantly higher compared to mechanical driven ones.

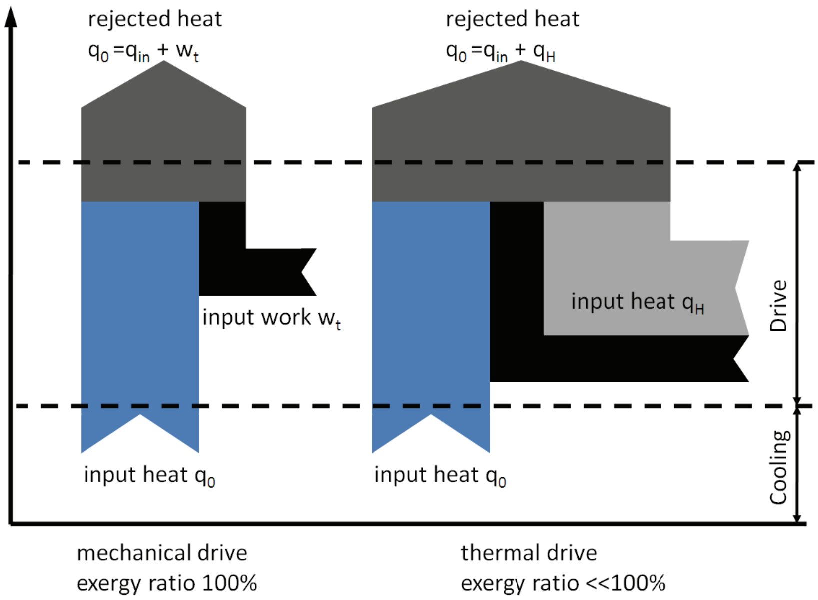


Figure 14: Simplified comparison of the drive between mechanical and thermal driven chillers

Therefore the recooling system is of utmost importance especially the focus is set to the auxiliary electricity consumption for instance by the cooling tower fan or the feeding pumps. If the heat is rejected to the ambient air two main possibilities exist, a closed cooling tower (dry cooler) and an open cooling tower (wet cooler). A combination of both is called hybrid cooling tower. The dry cooler uses a heat exchanger to reject the heat to the air. On the contrary the wet cooler sprays the cooling water into the air and a direct heat and mass transfer takes place. Therefore in wet cooling towers mainly latent heat is exchanged and in dry ones only sensible heat. The two recooling systems used in the two monitored plants are a wet and a hybrid cooling tower. Thus in Figure 15 a sketch of those cooling towers is shown. Furthermore a description of the various components is provided. Dry cooling towers are not further described in this work, because this method is not deployed in any of the two monitored systems.

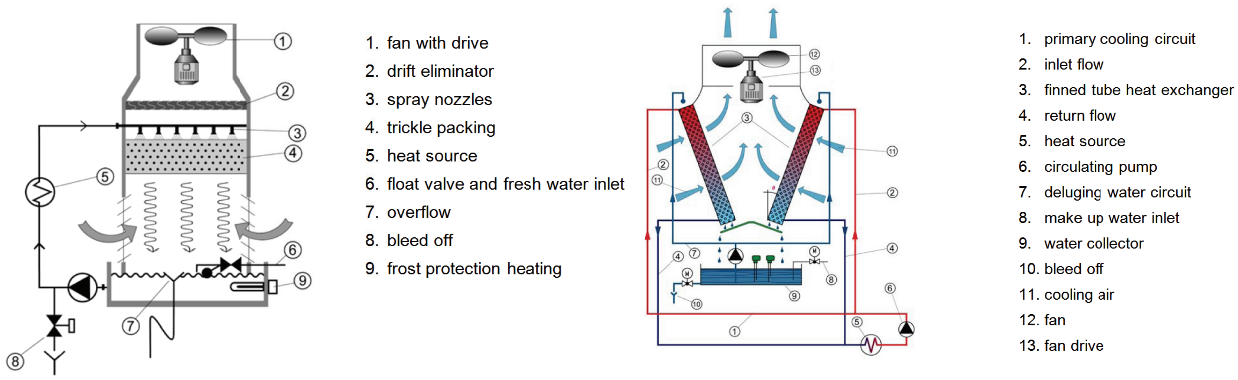


Figure 15: left: illustration of a wet cooling tower right: illustration of a hybrid cooling tower (Jaeggi, 2009)

Wet cooling towers have a large surface area trickle packing which is sprayed and provided with cooling water from the spray nozzles above. The cooling water flows through the packing into the basin and is pumped up from there again. The fan on top of the tower blows ambient air over the packing and is cooling the water. The air flow causes parts of the water to evaporate and is extracting heat from the cooling water. In the wet cooling tower the wet-bulb temperature indicates the degree of cooling, so cooling below the ambient dry-bulb temperature is possible. Characteristic approach temperatures (difference between water outlet temperature and the ambient wet-bulb temperature) are stated by the SWKI, (2005) between 4 and 8 K. Wet cooling towers are able to chill the water to cooler temperature levels, have lower investment costs and require less space compared to dry coolers. The hygienic problems, water consumption and the maintenance effort are basic disadvantages of wet cooling towers (Jaehning, et al., 2009).
Hybrid cooling towers as shown in Figure 15 on the left side combine the two methods of dry cooling and evaporative cooling. Depending on the requirements and the weather conditions they can switch between dry and wet mode. Figure 16 shows electricity consumptions of installed heat rejection systems from the Task 38 monitoring installations. The AXIMA EWK 036/06 and the Baltimore VXI 9-3X are installed in the two monitored plants (Bachler and Solid) of this work.

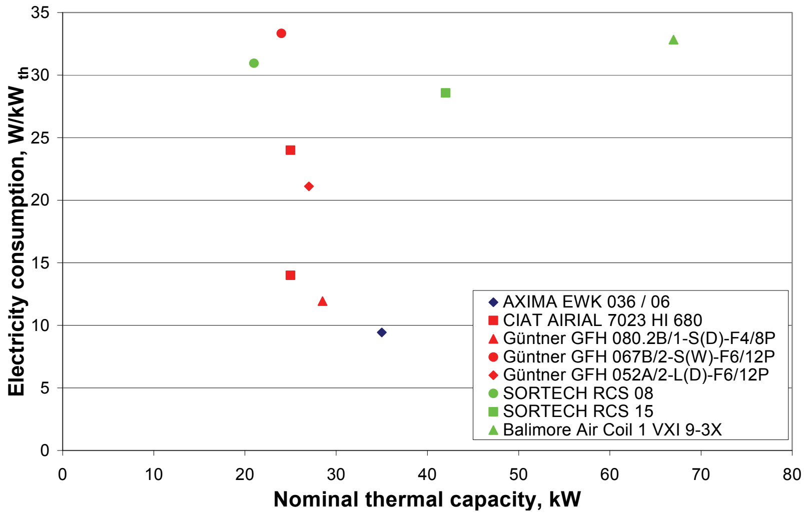


Figure 16: Electricity consumption of heat rejection systems installed in the Task 38 monitoring installations in Watt per kW rejected heat (Jaehning, et al., 2009)

2.6 System aspects

If a thermal driven cooling machine is connected to an existing solar thermal plant (such as a combisystem) a heat sink is added which has following characteristics and requirements:

- it needs little temperature differences (5-10 K) on a high temperature level (60-90 °C)
- there is a high volume flow towards the heat sink depending on the chiller (high flow)
- the storage tank size is relatively low in order to start the chiller early in the day without heating up a big size tank to high temperatures first

Due to these requirements it is not always possible to equip an existing solar thermal plant for space heating and domestic hot water with a thermal driven chiller. One aspect that is crucial for the decision, whether to apply a solar thermal cooling plant or not, is the hydraulic design of the collector field.

High-flow / Low-flow

To connect several single collectors there are two main possibilities, either in series or in parallel alignment. Often combinations of those two forms are used. Nevertheless a certain level of volume flow should not be undercut in the collector tubes in order to assure a turbulent flow and therefore a high heat transfer rate. Many existing solar combisystems (without cooling) work on the low-flow principle in the collector loop. Low-flow means a low volume flow and high temperature increase trough a serial collector flow. This is especially useful when the temperature lift for DHW and SH can be done in one step. If a thermal driven cooling process is connected only a small temperature increase is necessary for the chiller. Therefore a high-flow collector loop would be preferable. Summarized in summer high-flow operation and in winter low-flow operation would be favorable. There are some possibilities to achieve that goal such as switching the pump speed between summer and winter or to install a changeable collector design but none of them can serve both requirements in an optimal way. Thus plenty of optimization work is left in this field.

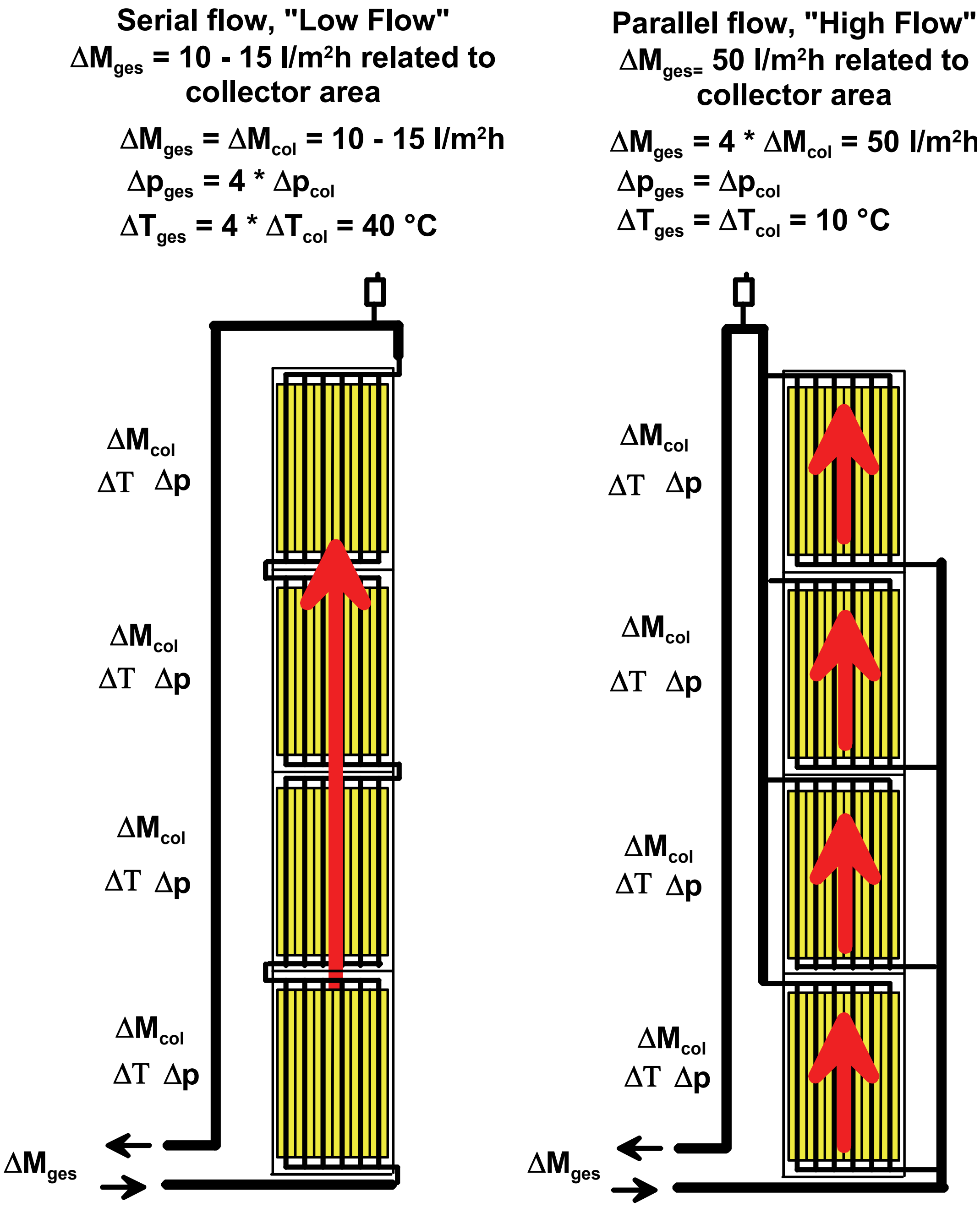


Figure 17: Hydraulics for the collector field Low-flow (left) and High-flow (right) systems. (Streicher, 2008)

3 MEASUREMENT RESULTS

In the following chapter the measurement and monitoring results of two solar cooling plants in Austria are presented. The monitoring results are part of the IEA Task 38 Subtask A.

3.1 Monitoring task for the IEA SHC Task38

Results of the International Energy Agency (IEA) Implementing Agreement on Solar Heating and Cooling (SHC) Task 25 showed that there is a great potential of the solar cooling technology for buildings especially in sunny regions (Henning, et al., 2006). In the year 2005 the IEA SHC Task 38 was started to bundle the research, design and planning processes. Within the IEA SHC Task 38 subtask A between 2008 and 2010. 23 small scale (<20 kW cooling capacity) and in subtask B 12 large scale systems are planned to be monitored. Therefore a unified monitoring procedure was introduced by Sparber et al in 2008. The two solar cooling plants which are presented in this work are monitored and analyzed following this unified monitoring procedure. In Figure 18 the basic monitoring scheme is shown. For the two monitored systems only the installed features are taken out of the proposed reference solar heating and cooling system. The respective schemes are drafted under the points 3.3 and 3.5.

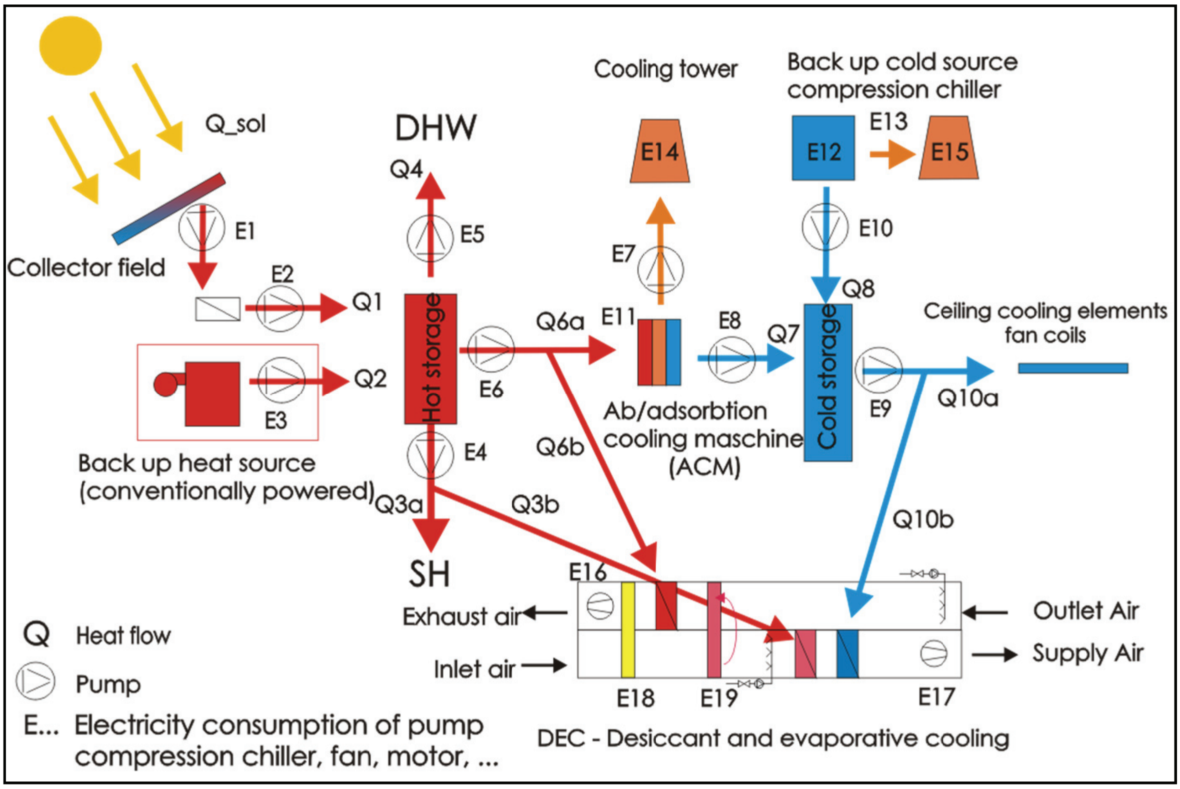


Figure 18: Proposed reference solar heating and cooling system from the Task 38 monitoring procedure (Sparber, et al., 2009)

In the monitoring procedure also a conventional reference system is described. This system is not further shown here, because the monitored systems do not have any kind of backup for the cooling mode. In order to allow a clear comparison between the different systems and monitoring results a few characteristic figures are introduced.

Primary Energy Ratio (PER)

The PER of the solar cooling system can be calculated as shown in Equation 3.1:

$$PER = \frac{(Q_3 + Q_4 + Q_{10})}{\frac{Q_2}{\varepsilon_{fossil} \cdot \eta_{boiler}} + \frac{E_{elec.tot}}{\varepsilon_{elec}}}$$ *Equation 3.1*

All the heat and electricity fluxes are measured. The primary energy conversion factors have been set to the values shown in Table 4. The assumptions are based on European Directives (Parliament, 2006), IEA SHC Task 25 and IEA SHC Task 32.(http://www.iea-shc.org, 2009)

Table 4: Values of primary energy conversion factors for heat and electricity from fossil fuels

$\varepsilon_{elec.}$=0.4	kWh of electricity per kWh of primary energy
ε_{fossil}=0.9	kWh of heat per kWh of primary energy
η_{boiler} =0.95	boiler efficiency

Total electricity consumption ($E_{elec.tot}$)

The total electricity consumption is calculated as the sum of all measured electrical consumers except E4, E5 and E9. Equation 3.2 shows the calculation.

$$E_{elec.tot} = \sum E$$ except E4. E5 and E9 *Equation 3.2*

$E_{elec.tot}$ is useful to calculate other important key figures such as PER and the electrical Coefficient of Performance.

Coefficient of Performance (COP)

The thermal COP is defined as the ratio of usable cold produced to the heat input, as seen in Equation 3.3. State of the art single effect absorption chillers that can be used for solar cooling vary in the thermal COP from 0.55-0.75.(Eiker, 2009)

$$COP_{therm.} = \frac{cold\ produced}{heat\ input} \quad \text{Equation 3.3}$$

$$COP_{elec.tot} = \frac{Q_{10}}{E_{elec.tot}} \quad \text{Equation 3.4}$$

In order to be able to compare solar driven cooling plants with conventional compression chillers the electrical COP for solar cooling plants is introduced. The electrical COP can be calculated as shown in Equation 3.4.

Costs per Kilowatt (CPK)

From the financial point of view a key figure is the overall cost per installed cooling capacity. The costs per kilowatt can be calculated as seen in Equation 3.5.

$$Specific\ solar\ assisted\ cooling\ installation\ cost = \frac{Cost\ (€)}{kW_{cold}} \quad \text{Equation 3.5}$$

In the following sections the two monitored plants are described and monitoring results for the summer 2009 are presented.

3.2 Coolcabin Solid

Name:	COOLCABIN
Type:	absorption technology
Capacity:	17.6 kW
Location:	Graz, Austria
Application:	office building

The “Coolcabin” was built in autumn 2008 as a demonstration plant for the company Solid GmbH. Solid realized in cooperation with the company Energy Cabin a compact cooling system for small offices. In the course of the IEA SHC Task 38 the plant was monitored in summer 2009 and will be monitored in summer 2010.

3.2.1 Overview

The whole cooling system (including a 17.6 kW absorption cooling machine running with lithium-bromide. a 2000 liter hot storage and all hydraulic auxiliaries) is placed in a cabin built by the company Energy Cabin and Solid at the entrance of the Solid office building. No backup for the cooling task is installed. The cold water is distributed through radiant ceilings in order to cool the 573.5 m^2 big office. There are 57.6 m^2 flat collectors installed on the Coolcabin. Via a heat exchanger the heat is delivered to the 2000 liter hot water storage. From there the hot water flows to the cooling machine. if there is a request for cooling, or to the heating distribution system, if there is space heating demand. The heat dissipation is carried out by a hybrid cooling tower which has a power of 67 kW and is placed on the roof of the office building. One special feature of the Coolcabin is the free cooling mode. Free cooling means the cooling machine is turned off; the cooling tower processes the whole cooling load. This is only possible if the outside conditions are sufficient, e.g. if the outside temperature is cold enough. In Table 5 the design data of the cool cabin is shown.

Table 5: Design data of the Coolcabin

absorber area of solar flat plate collectors	57.6	m^2
azimuth of solar field	south	
slope of solar field	11	°
heat storage capacity	2000	liter
power of the cooling tower	67	kW
type of cooling tower	hybrid BA VXI 9-3X	
to cool area of office building	573.5	m^2
maximum cooling capacity of the cooling machine	17.6	kW
rated COP of cooling machine	0.65	

3.2.2 Hydraulic scheme

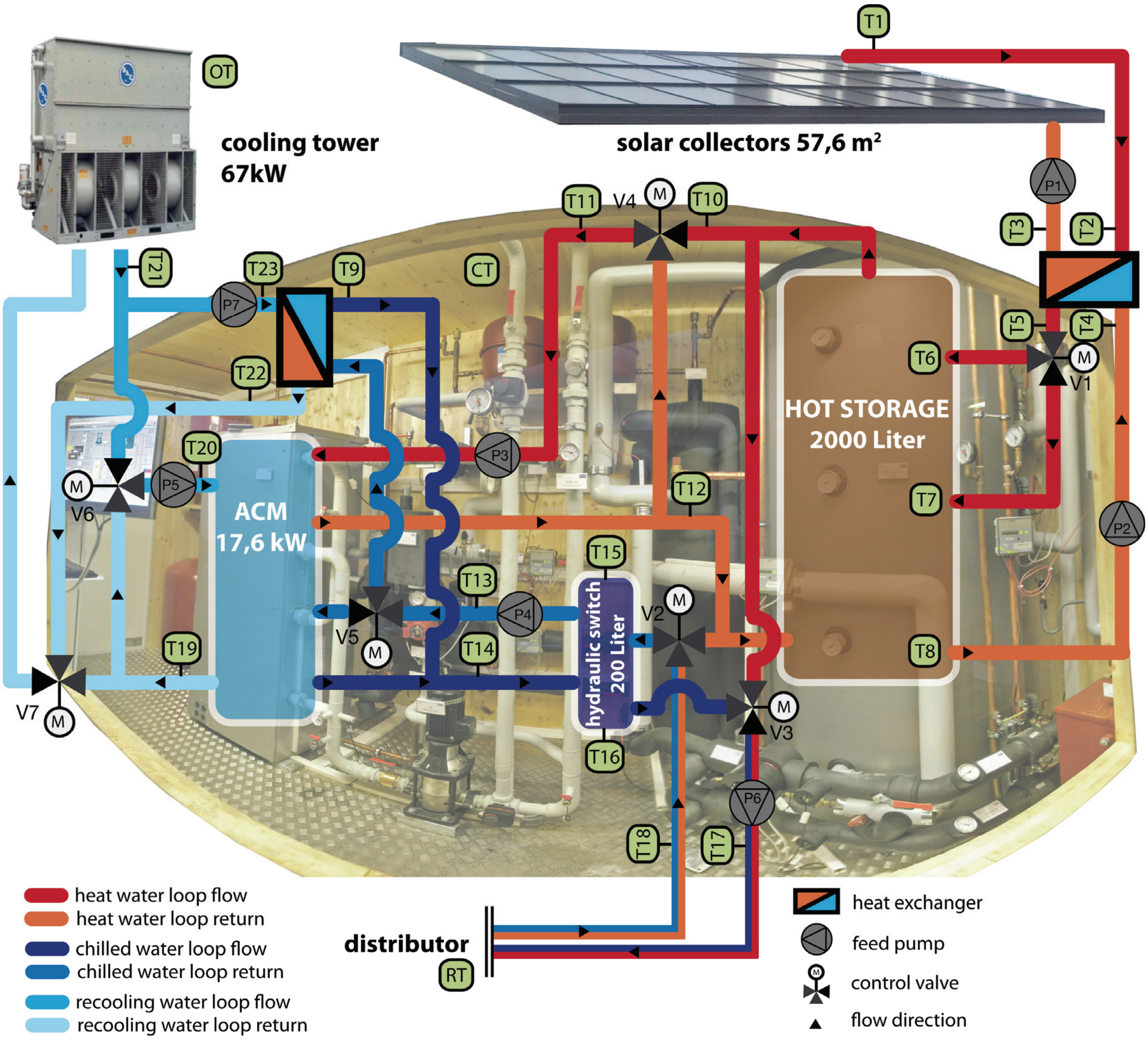


Figure 19: Hydraulic scheme of the Coolcabin including temperature measurement points

Figure 19 shows the functional scheme of the whole system. The hot water is produced in the 57.6 m^2 flat plate collectors, which are also used as roof for the Coolcabin. The collector field is aligned south with a slope of 11°. From the collector pump P1 pumps the hot water to the heat exchanger. Pump P2 delivers the medium in the secondary solar cycle to the hot storage tank, which has a volume of 2000 liter. Valve V1 controls the position (top. medium. down) where the hot medium is brought 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 (at certain standard temperature levels; see appendix), a rated COP of 0.65 and it is 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. Connecting the cooling tower and the ACM the recooling water loop can work in two different operation conditions. 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.

3.2.3 Control strategy

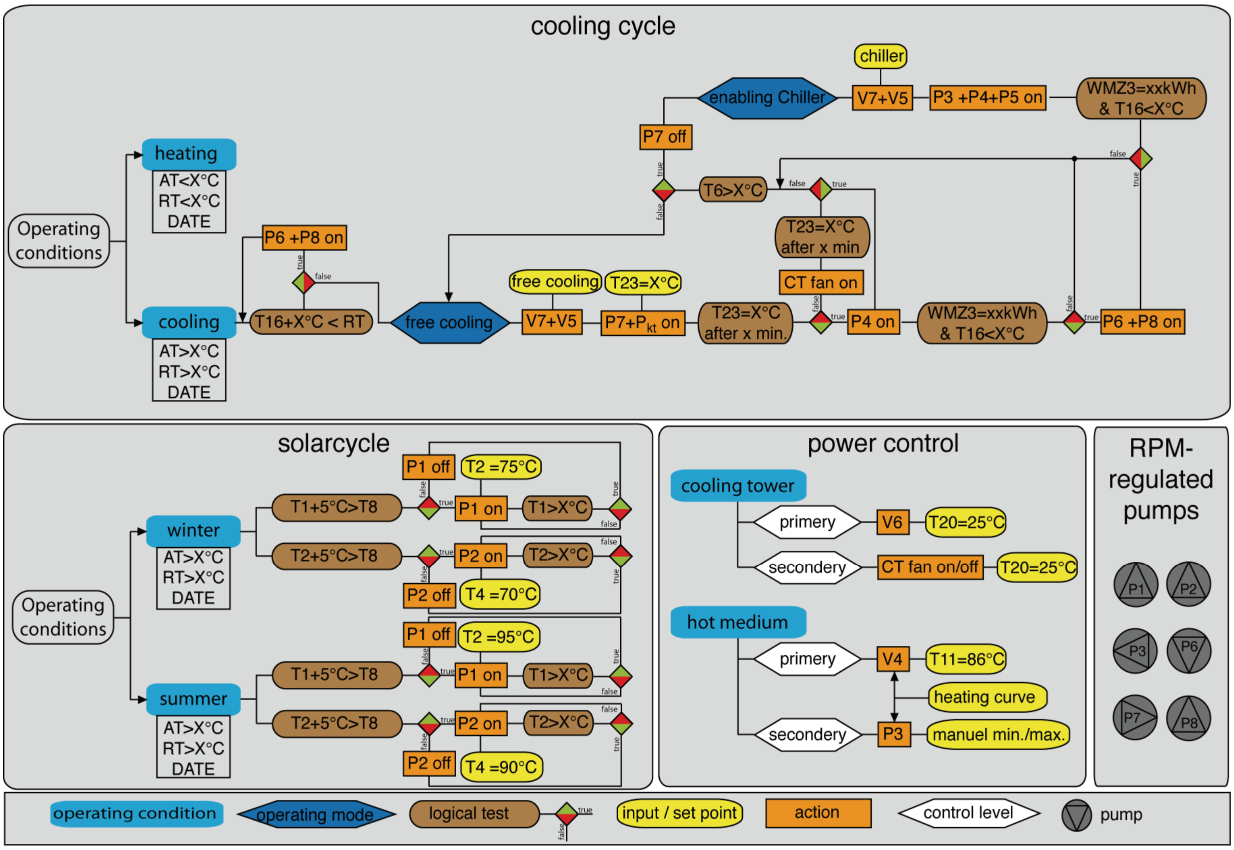


Figure 20: Logical control strategy for cooling conditions and the solar cycle control

In Figure 20 the logical control strategy is drafted for the cooling cycle, the solar cycle and the power control of the cooling tower and the hot medium. The first step considered in the control strategy is the determination of the operation conditions out of the outside temperature (OT), the room temperature (RT) and the date. Further on in this work the focus is set on the cooling conditions where two modes can be chosen, the free cooling and the enabling chiller mode. For the solar cycle there is a differentiation between winter and summer measured with the same parameters. In summer mode the primary solar cycle checks if T1+5°C is bigger than

T8, if the logical test is true, pump P1 is switched on and controlled with temperature T2. The same checks are done in the secondary solar cycle only with T2, pump P2 and T4. For winter conditions only the set points for the control temperatures differ. If the OT falls below 0°C an anti-freeze protection is turned on to protect the heat exchanger. The hot storage is protected with a maximum temperature restriction, shown with the logical test T1>X°C.
After determined cooling operation conditions a preliminary logical test is carried out. If T16+X°C is lower than the room temperature. P6 and P8 are switched on to bring the cold water to the radiant ceiling. This works especially in the morning time after a cold night before starting the free cooling mode.

Mode 1: Free Cooling
Valves V5 and V7 are switched to free cooling position. P7 and the pump of the cooling tower are turned on and regulated with T23. Ongoing a logical test is carried out if T23 has reached a certain temperature after a certain time. If the test turns out true pump 4 is started. if false the fan of the cooling tower is turned on and the test is carried out again. So it can be stated that there are two types of free cooling, one mode with and one without the fan. When T23 reaches x°C after x minutes also pump P4 is started. If T16 falls under a certain temperature and WMZ3 (heat flow meter positioned at the chilled water loop of the cooling machine) measures a certain amount of cold the distribution pumps P6 and P8 are starting. Now the cool cabin is working on free cooling mode. If T23 is not reached or T16 is not under x°C the chiller is enabled.

Mode 2: Enabling Chiller
If the cooling tower cannot satisfy the cooling demand the cooling machine has to be started. To turn on the ACM the temperature T6 has to be over x°C, then the valves V5 and V7 are switched to chiller position. Pumps P3. P4 and P5 are turned on. There is again a test if T16 is under x°C and if WMZ3 is making x kWh, then the distribution pumps P6 and P8 are started. Now the Coolcabin is working with the absorption chiller.

Power control
For controlling the power of the cooling tower (CT) and the hot medium, two control levels (primary and secondary) are used. In case of the cooling tower the valve V6 controls T20 to the set point of x°C, as a second way the CT fan can be switched on or off to reach this temperature. In order to control the hot medium the valve V4 regulates T11 to x°C. In the secondary control level the pump P3 can be switched manually to minimum or maximum flow rate for regulating the power with the volume flow. There is also the possibility to set a heating curve for the valve V4 and the pump P3 in winter operation conditions.

3.3 Measurement results Solid

In the following pages the measurement results of the summer 2009 are presented. At first the monitoring and measurement configuration is described in detail. The results are discussed under point 3.3.6. Some of these monitoring results were also presented at the IEA Task 38-Meeting 2009 in Palermo.

3.3.1 Monitoring configuration

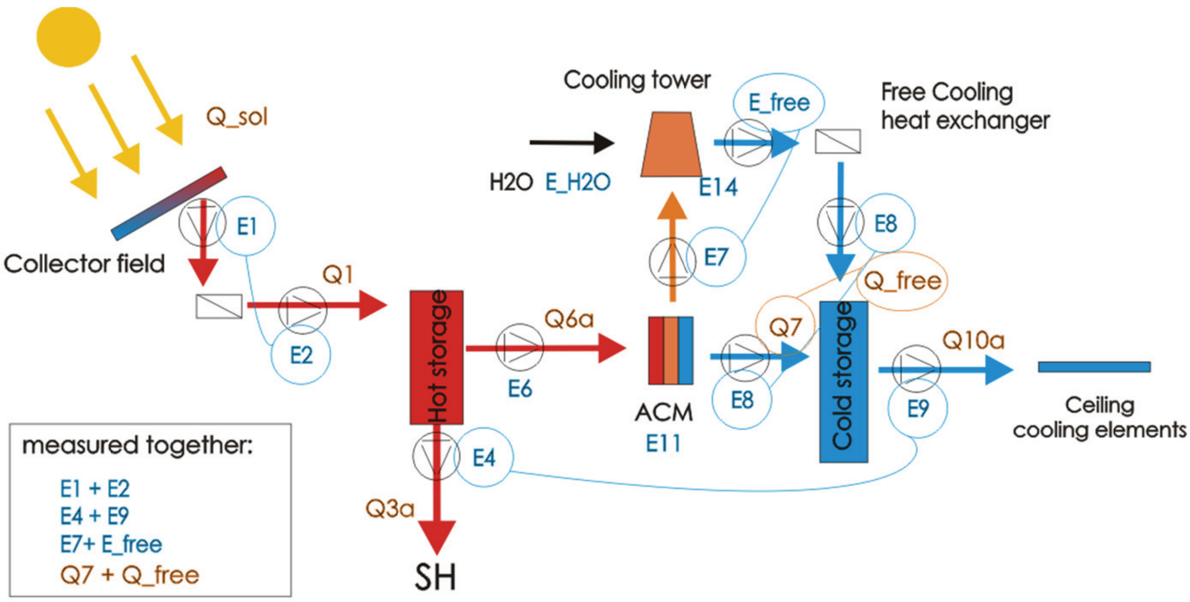


Figure 21: Monitoring scheme of the Coolcabin including electricity and heat measurement points

In Figure 21 the reduced reference system out of Figure 18 for the Coolcabin is shown. All electrical loads that are monitored are named with E and a running number. The heat loads are indicated with Q. Furthermore the combinations respectively the combined measured loads are shown with linked lines. The two electrical loads E8 in this scheme are in fact just one pump used in two ways and that is why it is indicated this way.

Most of the monitoring equipment is positioned directly in the Coolcabin. The company Schneid was responsible for designing the whole measurement and control system as well as the installation on site. Three main controllers of type Schneid where installed and a desktop computer makes an online access via internet possible. All in all four heat flow meters, six electricity counters and one pyranometer were installed to reach the level 3 monitoring requirements of the IEA SHC Task 38 monitoring procedure.

The electricity counters used are Berg DCMi-461 WP Class B and all heat flow meters are of the type Landis+Gyr Ultraheat 50 Class 3. Temperature measurement sensors are calibrated Pt-1000. All the monitored temperatures can be seen detailed in Figure 19. In the office rooms the room temperature is measured as well as the outside temperature. To measure the solar radiation (Q_sol) a Kipp & Zonen SPlite pyranometer was mounted in the same plane as the solar collectors.

3.3.2 Monthly results

In Figure 22 and Figure 23 the monthly results for July and August 2009 are shown. In this work only these two month are presented because of the low cooling demand in June and September in this summer. Three main parameters are taken to give a broad overview of the performance of the plant. The grey line indicates the outside temperature. The up- and downturns that can be seen indicate the day and night temperature variation. Furthermore the more stable red line shows the room temperature of the office building. An internal objective target of the design was to reach a temperature bandwidth in the office between 20 and 26°C. This objective was reached most of the time. In the days between the 6^{th} and the 10^{th} of July an offset problem did occur due to a manual offset setting of a maintenance worker. In the basis of the diagram the blue line indicates the cooling power that was delivered to the radiant ceiling. Two levels can be observed, the first one at about 15 kW when the chiller is enabled and the second when the free cooling mode was running at approximately 5 kW. The free cooling mode is mostly running during night times as well as during days with little solar radiation and relatively cold outside temperatures, for example on the days between the 10^{th} and the 11^{th} of July. Following average temperatures are linked to the cooling power in July and August (flow/return). For the chiller mode the chilled water temperature is 9°C/12°C. at an average recooling temperature of 26°C/28°C and a driving heat temperature of 76°C/73°C. If the free cooling mode is in operation the average chilled water temperature is 17°C/18.25°C at an average outside temperature of 19.5°C.

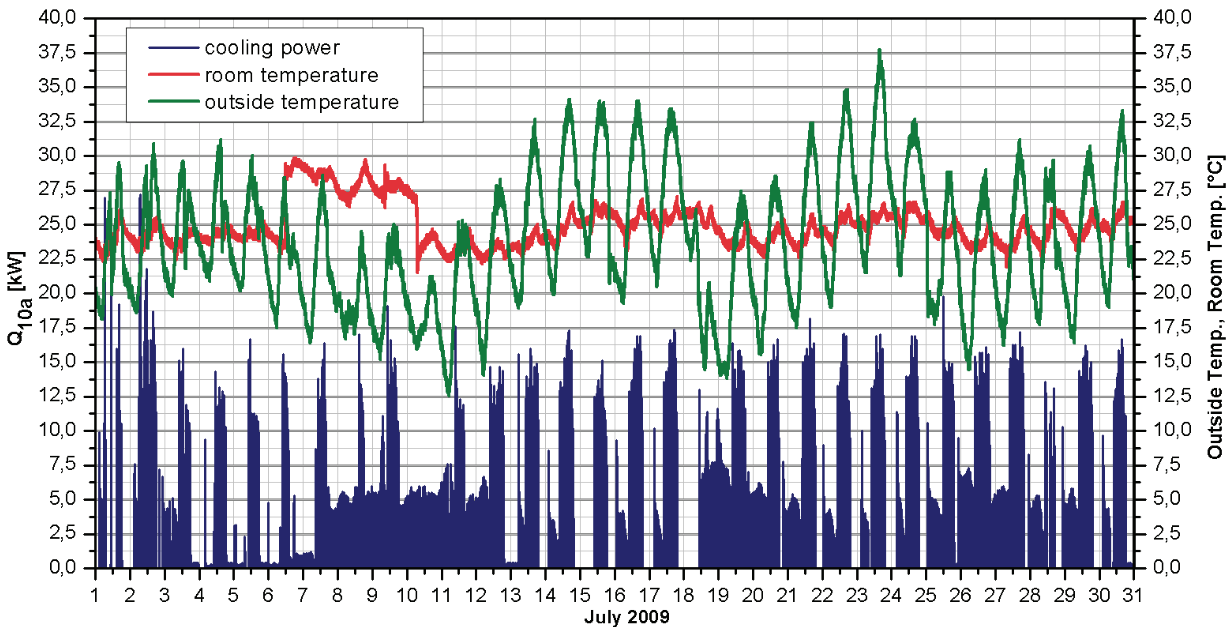


Figure 22: July 2009 monitoring results including the outside and room temperature

Furthermore it can be stated that the cooling season 2009 in Graz started in June and ended in September. Monitoring results for June and September are following these results quite well. All the monthly average values can be seen under 3.3.5.

Between the 2nd and the 3rd of August the Yasaki absorption chiller was not in operation due to an electrical problem. In this time no cooling load was delivered to the office because the outside temperature was too high for free cooling. It can be observed that the room temperature was soaring to a peak of 29 °C at an outside temperature of around 36°C. On the 11th of August a complete monitoring blackout happened due to a heavy thunderstorm.

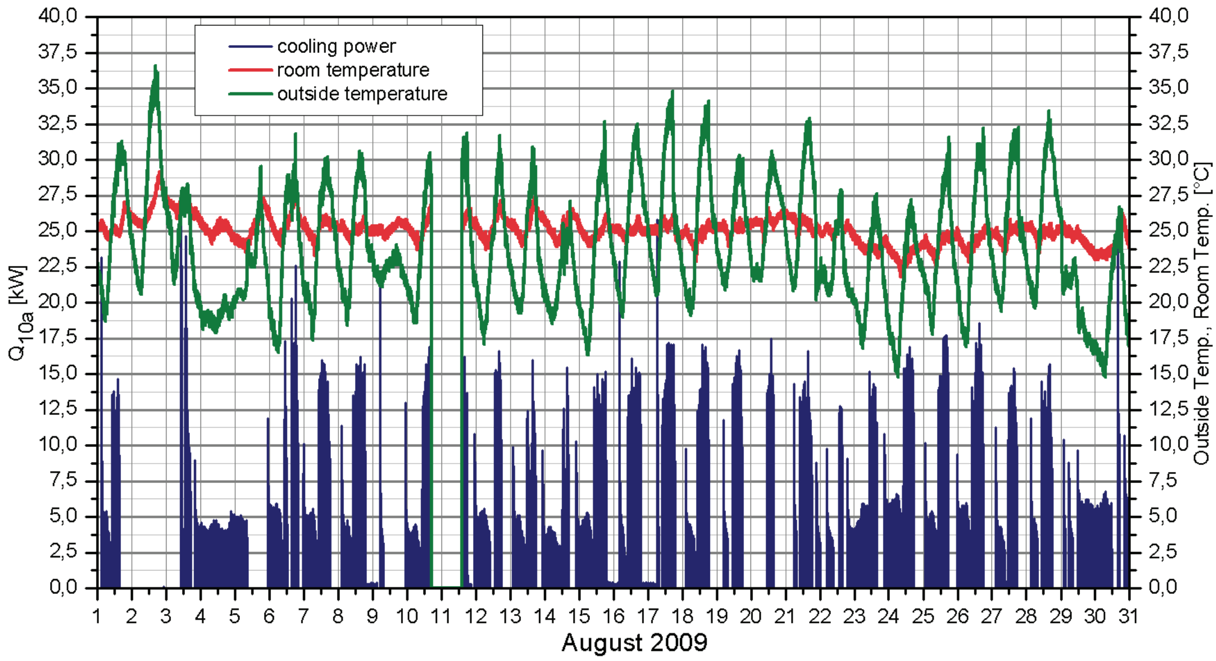


Figure 23: August 2009 monitoring results of the Coolcabin including the outside and room temperature

Furthermore the chiller started nearly every day in July and August, which indicates that the solar collectors are sufficient dimensioned. The practical operation behavior with detailed on/off switching points cannot be seen in these monthly diagrams. Only the overall performance as well as correlation between hotter or colder periods with the room temperature can be observed. More detailed results are presented in the following pages treating the absorption chiller and the free cooling mode.

3.3.3 Absorption chiller

One of the most important components of the whole solar cooling plant is the absorption chiller. The following section focuses on the water / lithium-bromide absorption chiller used in the Coolcabin. In Figure 24 a random day in August was picked and drafted in a diagram in order to evaluate the absorption chiller. Following Figure 23 the 25th of August is an average sunny summer day which represents a normal cooling day for the chiller. The cooling power, driving heat and solar yield are drafted together with the linked respective temperatures. Additionally the thermal COP is shown in Figure 26 for that day, to compare the practical experience with the manufacturing information of Yasaki.

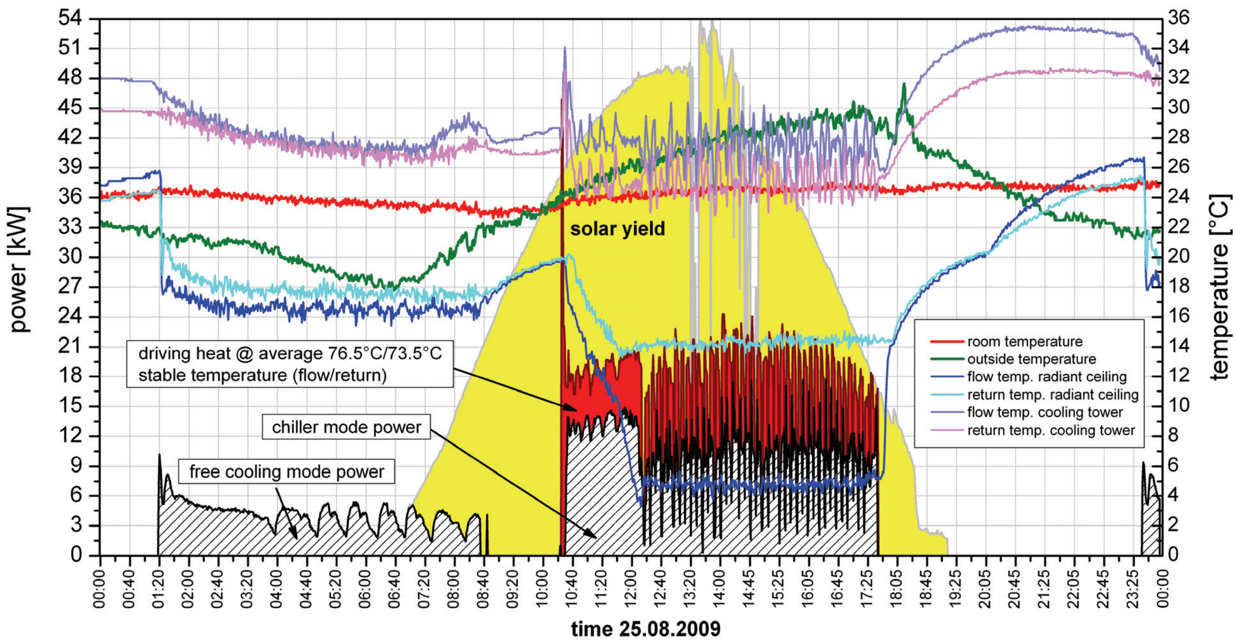


Figure 24: Power characteristics of the Coolcabin on the 25.08.2009

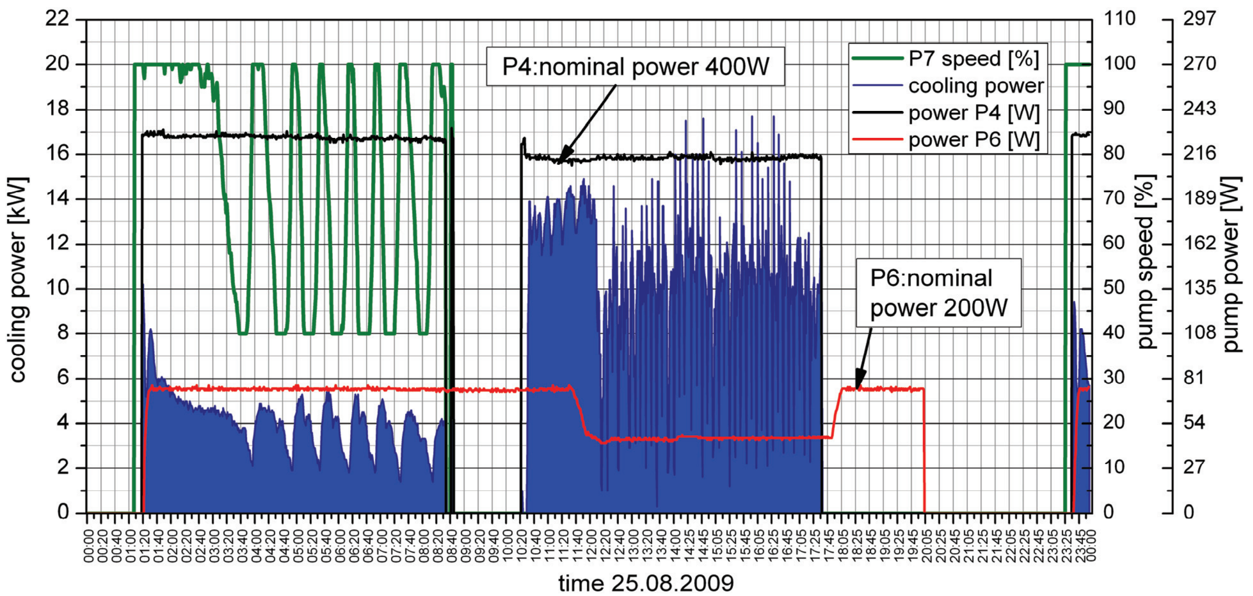


Figure 25: Different pump speeds and powers for the 25.8.2009

At 10:16 o'clock 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 24. 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 flow and return, the regulating valve of the distributor controls the flow temperature to 14 °C. Furthermore the flow temperature of the cooling loop is dropping to 4°C. 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.

In Figure 25 pump speeds of the pumps P4. P6 and P7 are illustrated. The chiller responds mainly to three volume flows. These volume flows are the hot water, the recooling and the chilled water loop. Pumps P5 and P3 (recooling of the chiller mode and hot water loop) are not speed controlled and run therefore constantly on a certain speed. Following Figure 25 pump P4 is running on a constant speed level as well. This pump delivers the chilled water to the small hydraulic switch. Furthermore pump P6 delivers the chilled water to the distributor towards the radiant ceiling. The pump is running below 50 % of its nominal power and reduces its speed right before the time where the machine starts to clock. Due to the on and off switching operation of the machine the chilled water outlet temperature stays stable at approximately 5 °C. During 12:20 and 17:45 the chiller is showing this power cycling behavior. The purpose of the volume flow reduction is a lower needed mass flow if the flow temperature falls below a certain value.

Ongoing the pump is still running even if the chiller or the free cooling mode is already switched off in order to deliver the last stored chilled water to the radiant ceiling. This is happening in the above diagram for instance between 8:40 and 10:20 o'clock. In case the free cooling mode is running pump P7 circulates the water loop from the cooling tower to the heat exchanger. As seen in the diagram the pump regulates the flow and return temperature in a certain way. Therefore the cooling power revealed a saw tooth tread design which can also be seen in the developing of the radiant ceiling flow temperature. The control bandwidth of the pump is limited between 40 and 100 %. In Figure 25 also the nominal electrical powers of P4 and P6 are listed in order to compare the actual working power levels with their full power levels. 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.

The standard operation conditions stated by Yasaki are at 88°C heat medium inlet temperature, a recooling water temperature of 31°C and a chilled water outlet temperature of 7°C (Yasaki, 2009). Due to the lower heat medium inlet temperature (approximately 76°C) the nominal output of 17.6 kW cannot be reached. Even though the higher average chilled water outlet temperature and the lower recooling temperatures are compensating some of the output losses compared to nominal conditions. In the night time the free cooling mode is running on a very low power level. The flow/return temperatures of the chilled water loop are at a level of 17°C/18.25°C.The free cooling mode is analyzed in detail under point 3.3.4.

Out of the manufacturing information of the Yasaki chiller WFC-SC5 the thermal COP reaches a value of 0.65 under standard conditions. In Figure 26 the thermal coefficient of performance is aggregated in 5 minute values for the operating period of the 25th of August. There are some values that reach high COPs. This can be mainly explained by the fluctuation of the driving power and the unsteady conditions related to the clocking. Furthermore the diagram shows the steady conditions in the beginning, where the 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.

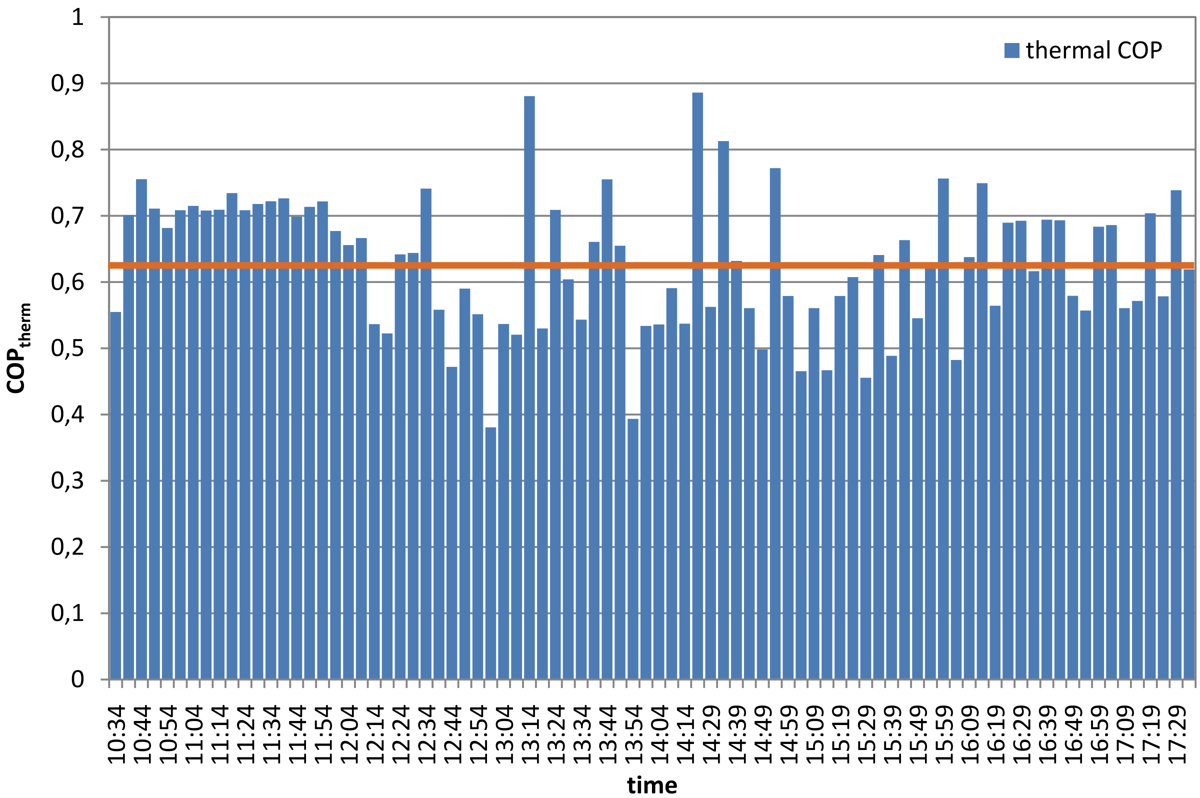


Figure 26: Thermal COP of the Yasaki absorption chiller of the 25.08.2009

The chiller was running very reliable throughout the whole summer. It also brought the expected power level and reached a feasible COP as well. 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.

3.3.4 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 under 21.5°C and the office still has a cooling demand. The company Solid never built a solar cooling plant with this option before. In order to evaluate the usefulness of further adoptions these measurement results show the practical experience with the free cooling mode in the summer 2009.

Similar to the absorption chiller under 3.5.3 a day in August was picked in order to evaluate the operation behavior of the free cooling mode. In Figure 27 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 in Graz the solar radiation of the 25.08.2009 is added. The whole day the solar radiation stays on a very low level compared to the 25th of August. It can be seen that the free cooling mode started three times that day, always switching on if the outside temperature falls under 21.5°C and stopping if the outside temperature exceeds the 22°C level. The room temperature stays quite stable throughout the day showing a slight downtrend in bandwidth between 25°C-23°C. Furthermore the flow and return temperature to the radiant ceiling of the office shows values between 18-16°C/19-17°C (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.

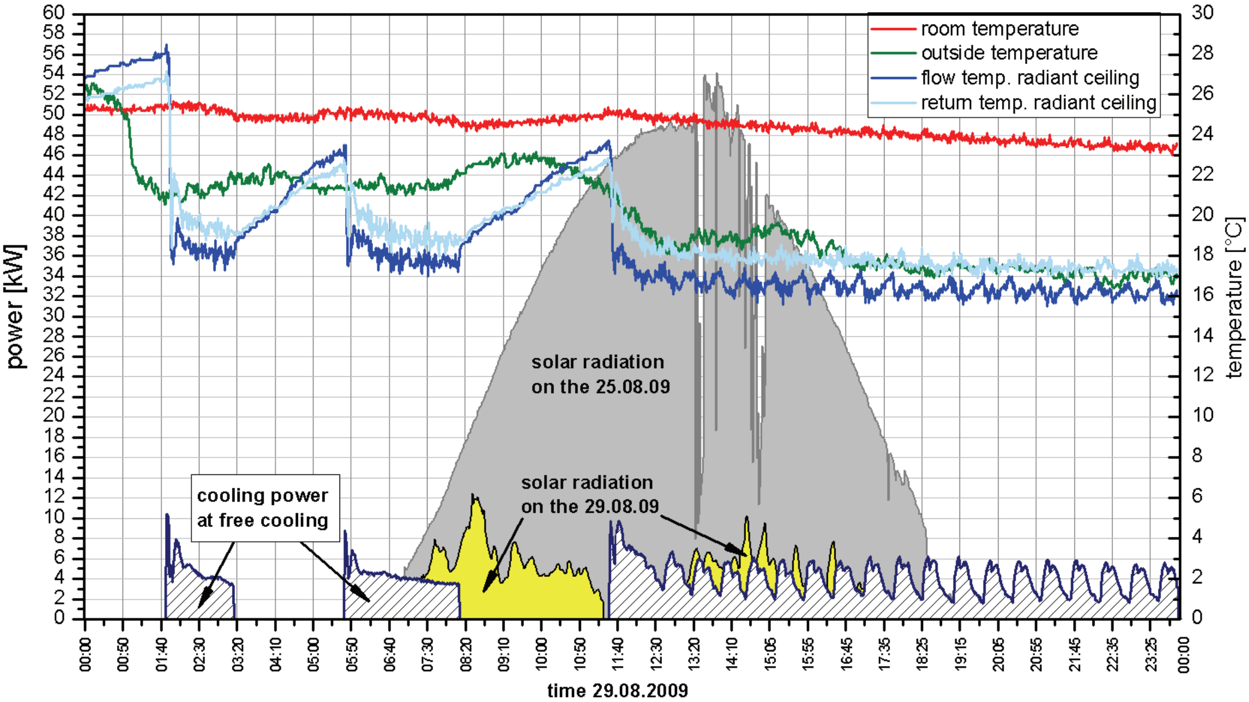


Figure 27: Free cooling operation performance of the Coolcabin on the 29.08.2009

From 12:30 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 28 the practical operation experience is marked in a diagram. The room temperature stays very stable between 25°C and 23°C. 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.

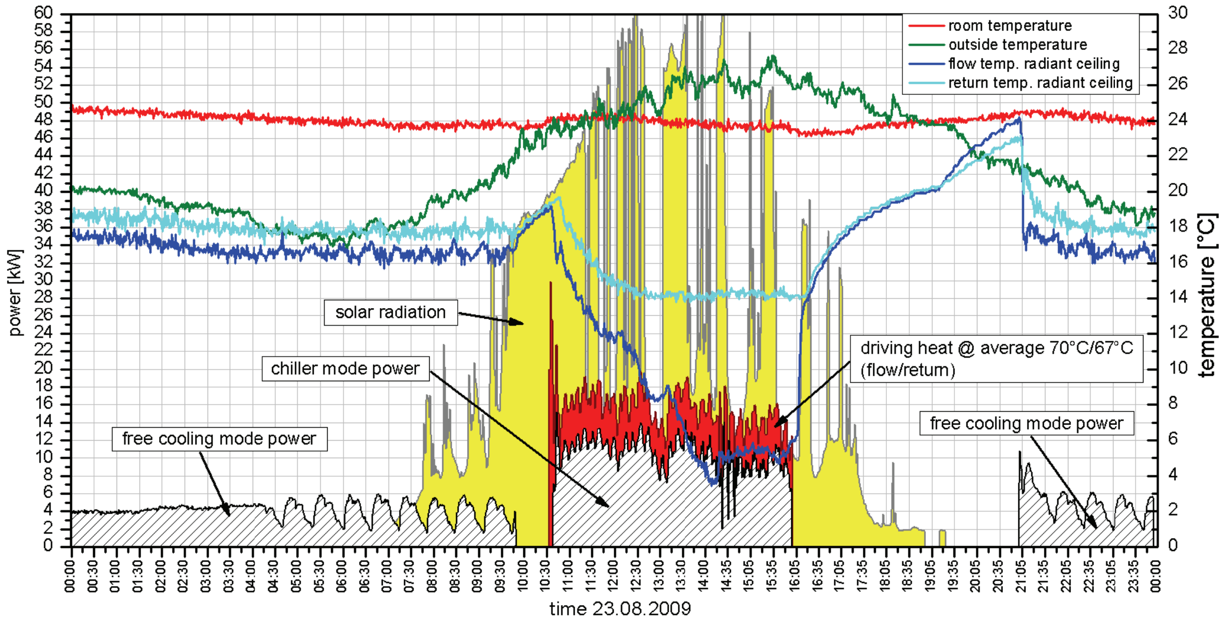
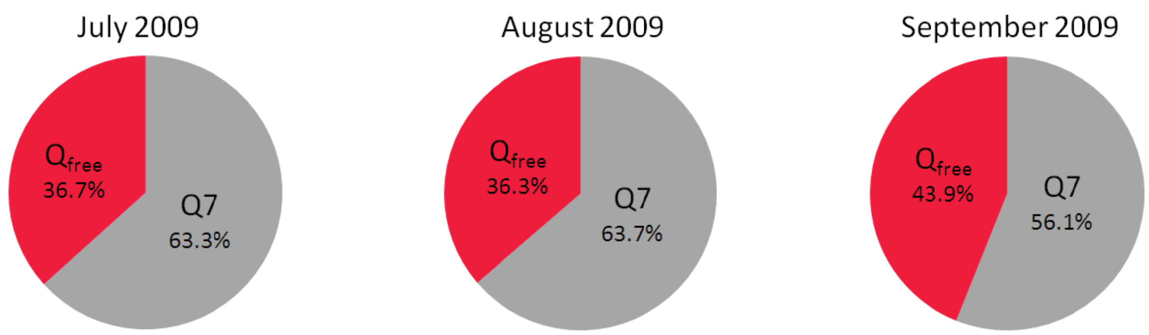


Figure 28: Practical operation performance of the Coolcabin on the 23.08.2009

Out of Figure 29 the free cooling share as a slice of the total delivered cooling load for the month July. August and September is shown, 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 29: Free cooling share of the total delivered cooling load for July, August and September

Approximately more than one third of the total cooling load was done with free cooling. The free cooling mode is especially running in night times and on rainy days. For further interest a detailed list of the starting and stopping conditions in August is attached in the appendix.

Under 3.3.6 the monitoring result of this section will be discussed in detail and suggestions of improvements will be provided, especially regarding the free cooling mode and the cooling tower.

3.3.5 Level 3 monitoring results

Level 3 monitoring results make it possible to compare different existing solar cooling plants around the world. In Table 6 the monitoring results for summer 2009 are summarized.

Table 6: IEA SHC Task 38 monitoring results for the Coolcabin in the summer 2009

Name	June	July	August	September	Unit
Q_sol	8294.0	9298.3	8822.0	6458.3	kWh
Q1	2347.0	3673.0	2958.0	1753.0	kWh
Q3a	0	0	0	0	kWh
Q6a	1981.0	3397.0	2679.0	1267.0	kWh
Q7	1667.5	2013.1	1702.0	670.0	kWh
Q_free	252.5	984.9	971.0	525.0	kWh
Q10a	1920.0	2998.0	2673.0	1195.0	kWh
E1+E2	33.0	54.0	45.0	35.0	kWh
E4+E9	10.0	37.0	35.0	25.0	kWh
E6+E11	94.0	199.0	151.0	137.0	kWh
E7+Efree	212.0	379.0	293.0	260.0	kWh
E8	128.0	131.0	103.0	78.0	kWh
E14	558.0	921.0	835.0	363.0	kWh
E_tot	1025.0	1684.0	1429.0	873.0	kWh
PER	74.9	71.2	74.8	54.8	%
CPK	10466.0				€/kW

The PER is calculated with Equation 3.1 using the primary energy conversion factors from Table 4. Figure 30 illustrates the PER together with the total electrical consumption and the delivered cooling load per month. The PER only reaches very poor values due to the high electricity consumption compared to the cooling load. The high $E_{elec.tot}$ is mainly due to the high electrical consumption of the cooling tower. For the 17.6 kW nominal cooling power the cooling tower is overdesigned with 67 kW maximal heat rejection power. With approximately 33 W/kW_{th} the Baltimore hybrid cooling tower performed very poor compared to other cooling towers within the Task compare *(Jaehning, et al., 2009)*.

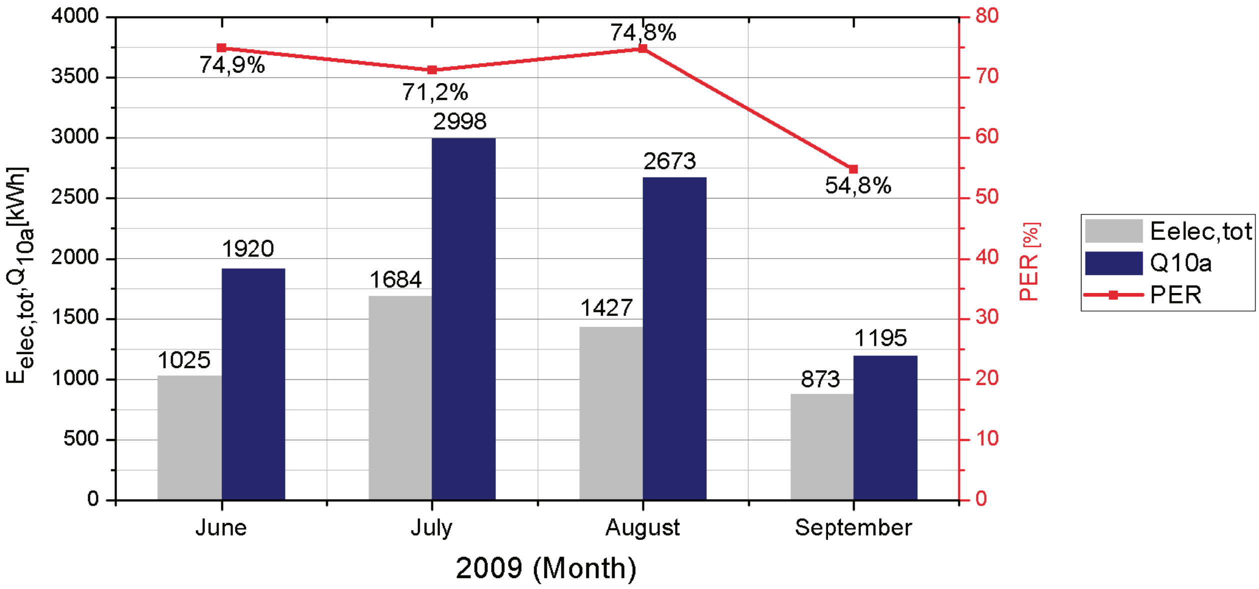


Figure 30: Values of the Primary Energy Ratio (PER) for summer 2009

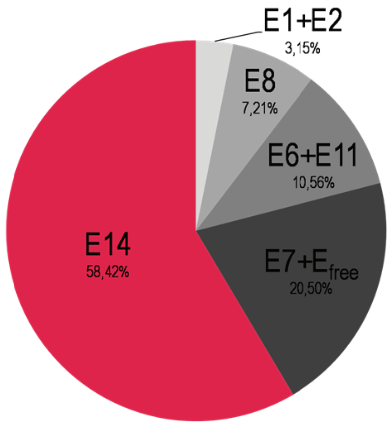


Figure 31: Split up total electrical consumption in August 2009

Figure 31 takes a closer look at the total electrical consumption in August and splits up the different contributors. E14 (cooling tower) has the main share of the total consumption. The reason for the high electrical consumption of the cooling tower is its high fan power which is not speed-regulated. The second largest share is taken by the pump towards the cooling tower (E7 + E_{free}). As described before, the cooling tower is placed on the roof of the office building. Therefore the Grundfos CRI 10-3 pump with 1200 W nominal electrical power was installed to deliver the volume flow up to the roof. All the other pumps like the solar pumps (E1+E2) or the other circulating pumps together are only responsible for 20% of the total electricity consumption in August. These results direct the focus for further improvements of the plant to the heat rejection system including the hybrid cooling tower and the pumps P5 and P7. Using the position of valves V5 and V7 to divide the different heat flows and electrical consumptions into the two modes (free cooling and chiller mode) the split up coefficient of performance (COP) can be calculated. Figure 32 shows the fragmented average electrical COPs for the month July, August and September. Also the average thermal COP of the chiller is drafted in this diagram. The average results for the COPs indicate very bad values especially for the free cooling mode. For the nominal power of 67 kW (at 35°C/29.5°C inlet temperature/outlet temperature) the cooling tower only delivers around 5 kW cooling power (at 18°C/16°C inlet/outlet temperature) at the free cooling mode. The average thermal COP of the absorption chiller matches quite well with the expectations. Suggestions for improvement after the first cooling summer will be given in the next section.

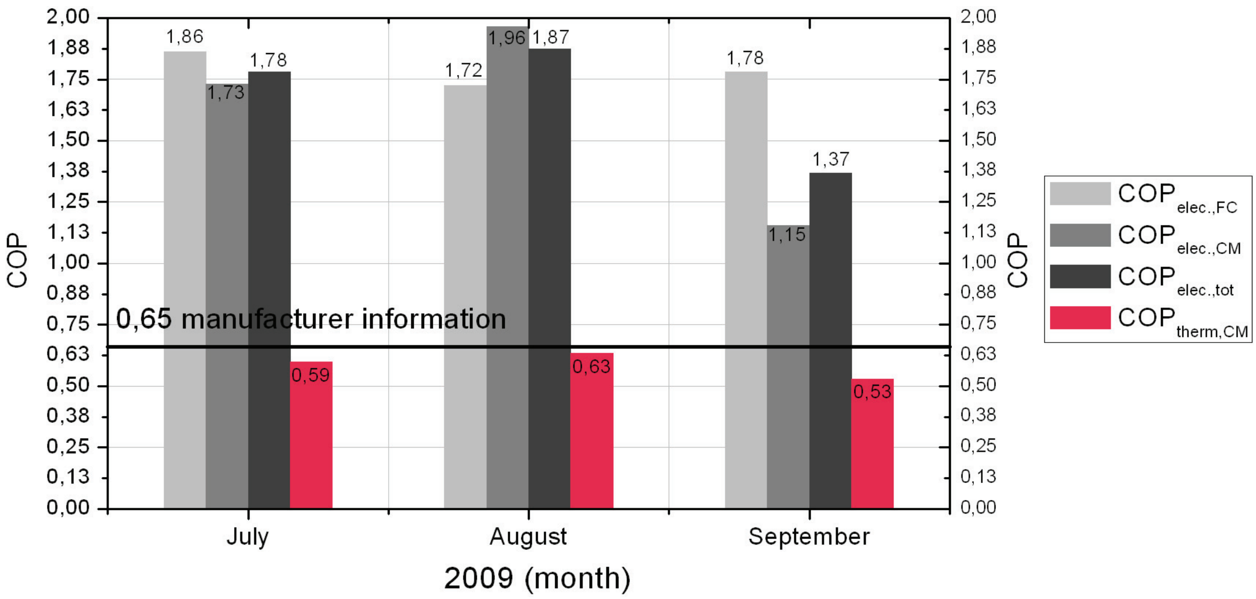


Figure 32: Fragmented average electrical and thermal COPs for the month July, August and September

Table 7: Subdivided investment costs of the solar cooling plant Coolcabin in detail

name	without VAT	with 20% VAT	unit
absorption chiller Yasaki	20.000.00	24.000.00	€
solar flat collectors ökoTech HT	24.000.00	28.800.00	€
Coolcabin including auxiliary hydraulics	51.000.00	61.200.00	€
anti-freeze protection + electro installations	3.000.00	3.600.00	€
hybrid cooling tower	17.000.00	20.400.00	€
control system	1.500.00	1.800.00	€
water treatment system	4.000.00	4.800.00	€
planning + designing	18.000.00	21.600.00	€
installation of the whole plant	15.000.00	18.000.00	€
total investment costs I	153.500.00	184.200.00	€
CPK	8.721.59	10.465.91	€/kW_{cold}

In Table 7 the broken up investment costs for the plant in Graz are listed. The total costs are shown with and without 20% VAT as well as the costs per kilowatt cold. These costs indicate total investment costs for a demonstration plant.

3.3.6 Analysis of the result & suggestions for improvement

One of the main problems of the plant is the high electrical consumption leading to bad electrical COPs. A reason for this is for sure the oversized cooling tower with its high energy demand. When the system was designed the electrical COP was not of special interest because it was expected to be low. Therefore the whole design was not focused on electrical energy efficiency. Changing an operating system is usually difficult and linked with high costs. Nevertheless some suggestions of improvement concerning the electrical performance of the plant will be provided at the end of this section.

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 (see Figure 23. first few days). At the end of the summer some problems with lime scale were discovered in the recooling loop, the consequences of which cannot be estimated yet.

Furthermore the daily operation performance diagrams indicate that the radiant ceiling is dimensioned to small for the system or vice versa. One lesson that can be learned out of the Coolcabin-project is, either the cooling system is designed for a specific cold distribution system that is known in the beginning, or the distribution system is dimensioned to reach the cooling power demand of the cooling plant. 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. Especially the absorption chiller has to clock as it can be seen in Figure 24. Interviews with people working in the office showed that the room temperature is in the higher range of their personal satisfaction. So the radiant ceiling is not only too small for the solar cooling plant but also for the office itself. Increasing the cooling power of the radiant ceiling in general is one suggestion of improvement. Another big issue is matching the temperatures between the two systems. The solar cooling system was designed for running on a 9°C/17°C nominal flow/return temperature, but the cold distribution (radiant ceiling) requires only 16°C/19°C. 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.

As Figure 29 shows the free cooling mode delivers over one third of the total cooling load to the building. The electrical COPs of the free cooling mode are not very satisfying in times where the office is not occupied. Due to the low heat storage capability of the building the cooling effect is very limited. Simply opening the window in times where the outside temperature is falling below 21°C (when the free cooling mode is running) would lead to the same or even better chilling effect in the office. Taking the current boundary conditions of the Coolcabin into account it can be concluded that it is not reasonable to run on a free cooling mode. 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 there are people in the Solid office. Free cooling of an empty office with low heat storage capability just raises electricity costs.

Out of the measurement results (compare Figure 31 and Table 6) it can be seen that the heat rejection system including the cooling tower and the recooling water loop pumps plays a key role in reducing the electrical consumption of the plant. Replacing the existing cooling tower to a smaller and more efficient one would be one solution for solving the problem. With a more efficient cooling tower (compare Axima in Figure 16) and better pumps the electrical COP could easily be doubled. Therefore a pure wet cooling tower is preferable. Due to the fact that the fan of the existing hybrid cooling tower is not speed controlled the electrical power stays stable at 80% of the nominal power. So a speed-control for the existing fan in combination with the correct control strategy could reduce the electrical consumption of the heat rejection system as well.

Summarizing four practical suggestions for improvement have been found:

- reducing the radiant ceiling inlet temperature to 15°C in order to increase the cooling power of the distributor without making structural alterations
- including an outside and inside humidity measurement and monitoring for the next summer
- adopting the free cooling time control to office hours or stop free cooling completely
- replacing the cooling tower or controlling the speed of the existing cooling tower fan

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 further computer simulations will be carried out.

3.4 Bachler/Gröbming Pink

Name: Bachler Pink
Type: absorption technology
Capacity: 12 kW
Location: Gröbming, Austria
Application: office building

The solar cooling plant at the Bachler GmbH training and office building was installed in spring 2007. It is delivering its cooling load to an office trough a radiant ceiling. In the course of the IEA SHC Task 38 the plant was monitored in summer 2009 and will be monitored in summer 2010.

3.4.1 Overview

This solar cooling plant was built as an attachment of an existing solar heating system. Commissioned in the year 2007 it delivers its cold load to the office of the Bachler GmbH. Steiner GmbH in cooperation with the Pink GmbH planned and built a solar cooling system 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 facade and are 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 and for warming the water of a swimming pool as well. In the heating season the solar plant is also used to provide space heating combined with local district heating. In Table 8 the most important design data is listed.

Table 1: Design data of the solar cooling plant in Gröbming/Austria

absorber area of solar flat plate collectors	46	m^2
azimuth of solar field	152° south-east	
slope of solar field	45	°
heat storage capacity	4500	liter
power of the cooling tower	34	kW
type of cooling tower	Axima EWK 036/06	
to-cool area of office building	700	m^2
max. cooling capacity of the cooling machine	12	kW
rated COP of cooling machine	0.7	

3.4.2 Hydraulic scheme

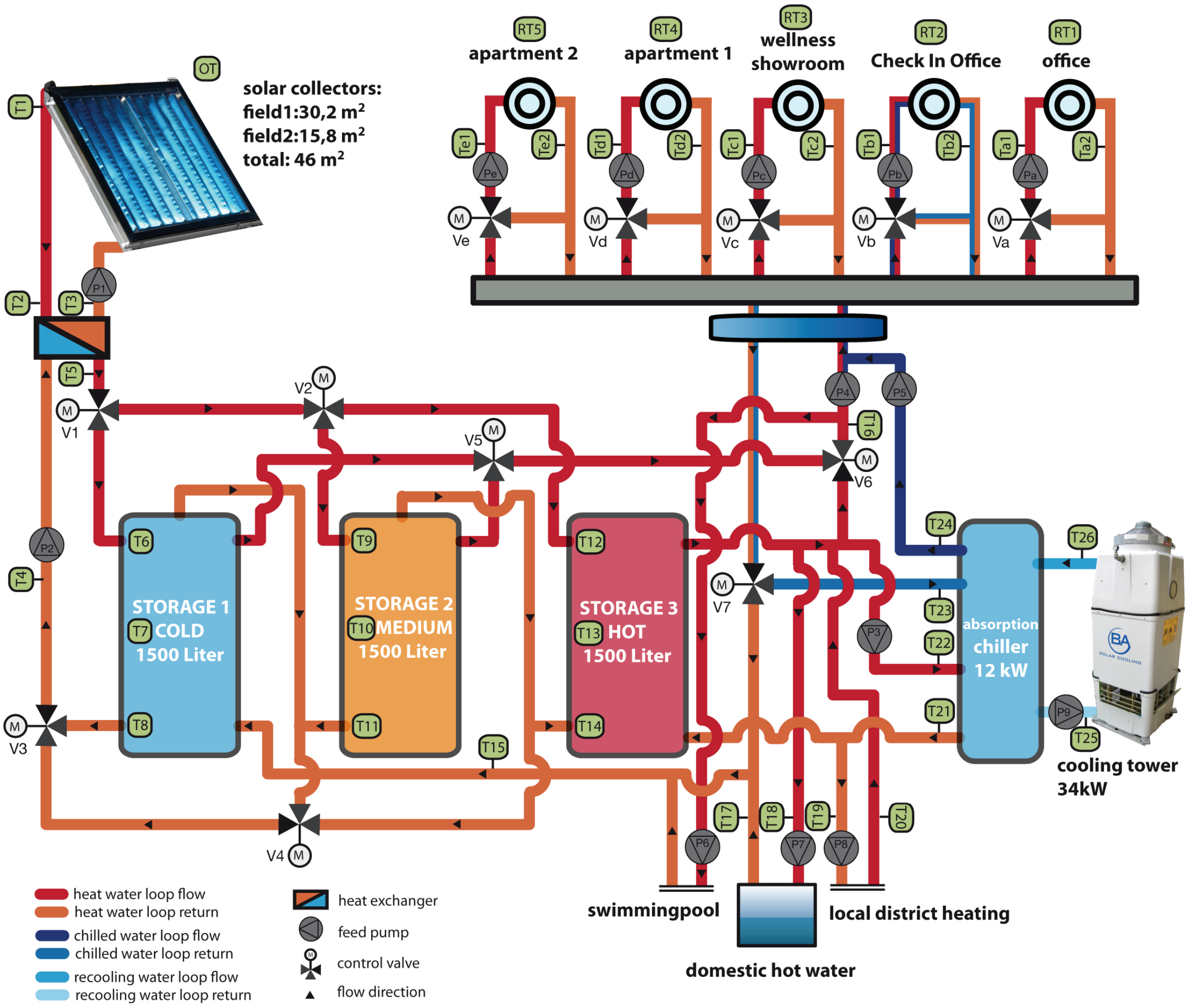


Figure 33: Hydraulic scheme of the solar cooling system in Gröbming at Bachler GmbH

In the beginning the system was designed as a solar assisted heating system for two office rooms, two apartments and a wellness showroom. Furthermore the swimming pool is heated with the solar energy. As a backup for the heating system the domestic 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. Also radiant ceilings were installed in the office rooms of the Bachler GmbH.

In Figure 33 the hydraulic scheme of the installed system in Gröbming at the Bachler GmbH is shown. There are two solar collector fields, one on the ground with 30.2 m^2 and the other one integrated in the facade with 15.8 m^2. From the solar collectors the heat flows through a heat exchanger directly to the hot storage. Three 1500 liter water storages are used to store the solar energy. Depending on the temperature the heat water flow is stratified to the hot, medium or cold storage. In winter the return flow 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 be also done alternatively from storage 1 or 2 if the temperature is sufficient. From the central distribution station the hot water is distributed to the different recipients. In summer time when the cooling mode is enabled the chilled water for the radiant ceiling of the office room is also distributed through the same central distributor station. The recooling water loop connects the absorption chiller directly to the wet cooling tower. All in all nine auxiliary pumps are installed in the system as well as five distribution pumps. To control the plant two Technische Alternative (TA) UVR1611 controllers are included and also an online monitoring infrastructure installed by the company RKG. In the following section the control strategy shall be described.

3.4.3 Control strategy

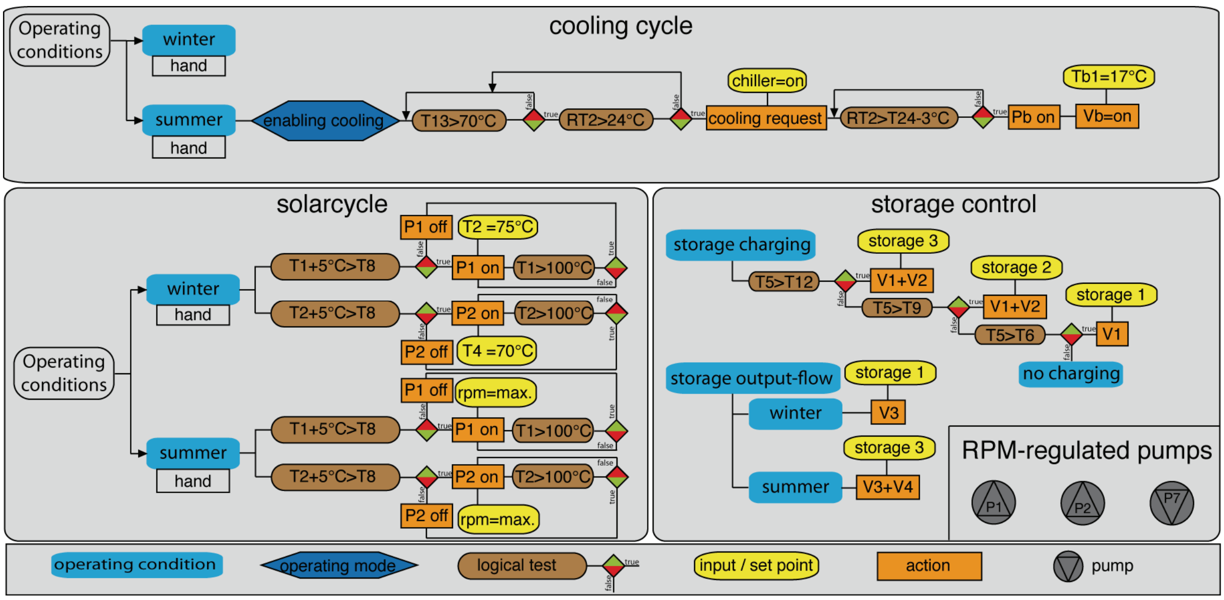


Figure 34: Control strategy of the solar cooling plant at Bachler GmbH in Gröbming

In Figure 34 the control strategy of the solar cooling plant in Gröbming is illustrated. All the temperatures and nomenclatures are linked to the hydraulic scheme (Figure 33). Only the cooling cycle as well as the management of the solar cycle is described in this work. Of course, 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. Under point 3.5.5 some examples for problems regarding the interaction of the different components will be described.

Cooling cycle

The absorption chiller PSC-12 as mentioned under 3.4.1 is located outside the building. As a consequence of this location, twice a year a maintenance worker has to discharge and charge the absorption chiller before and after the winter period to avoid freezing damages. To eliminate any control failure regarding winter and summer specific 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°C. 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°C and the chiller is switched on. In the next stage the chilled water flow temperature minus 3°C has to be less than the room temperature, then the distributor pump is started and the control valve Vb is enabled to regulate the flow temperature at 17°C.

Solar cycle

The control of the solar cycle is very similar to the control strategy of the solar cooling plant in Graz. Switching between summer and winter is done manually. Temperature T1 plus a margin of 5°C has to be above the lowest temperature of storage 1 (T8) to start the primary solar pump. In summer this rpm-regulated pump (P1) is not controlled to any set temperature it is just running on maximum speed. In winter the primary solar pump is regulated to a set temperature of 75°C. Because there are two solar collector fields T1 is averaged out of two collector temperatures measured in those two fields. Analogical the secondary solar cycle is turned on with T2. It is not regulated in summer but regulated to a set temperature of 70°C in winter. To avoid too high pressures in the hot storage the solar cycle is switched off if either T1 or T2 is above 100°C.

Storage control

As shown in Figure 33 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 34. 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 as much as possible solar energy should be stored. 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°C to 80°C is recommended. Nevertheless the machine is running down to 65°C driving temperature, but with poor thermal COPs. Detailed diagrams and information regarding the Pink chiller PSC-12 can be found in the appendix. 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 be the case. 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. How good the whole plant is working in reality and which problems did occur during summer 2009 will be discussed in the following section.

3.5 Measurement results Pink

Analogical to 3.3 measurement results of the summer 2009 are presented in the following section for the solar cooling plant at Bachler/Gröming. Primarily the monitoring and measurement configuration is described and specifications of the measurement equipment are provided. The results will be discussed under point 3.3.6.

3.5.1 Monitoring configuration

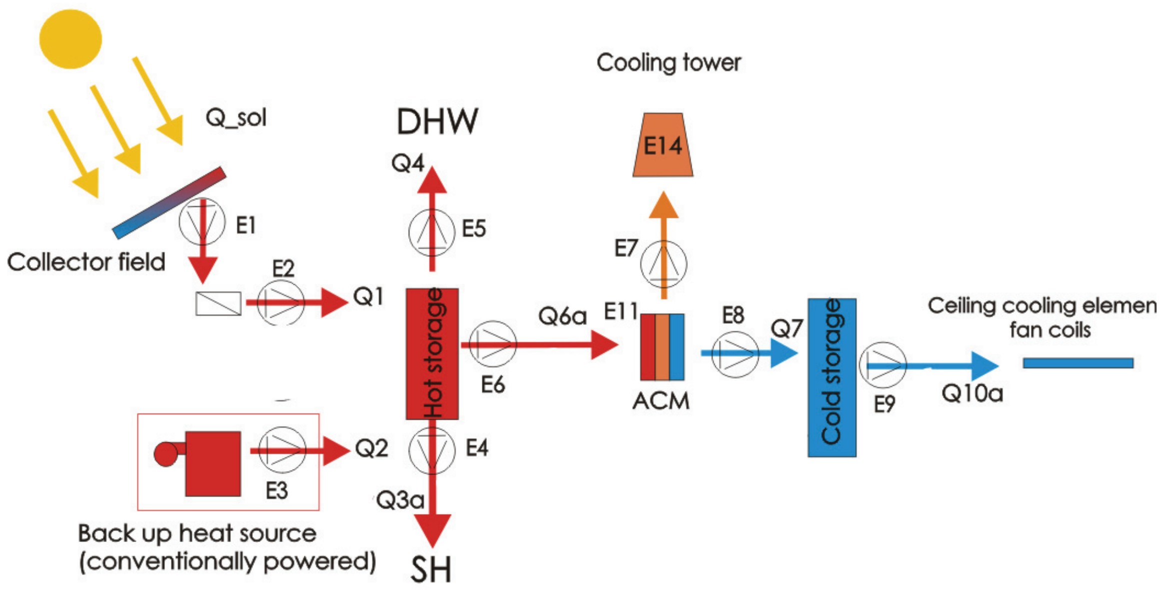


Figure 35: Monitoring scheme of the Pink plant including electricity and heat measurement points

Figure 35 shows the symbolic scheme of the system indicating all monitored energy fluxes. The heat flow Q2 is drawn to back up the space heating but is also a backup for the domestic hot water. Further on the cold storage of the plant is only a small hydraulic switch linked to the central distribution station. Cold losses due to the cold storage are expected quite low. The backup heat source from the local district heat is additionally monitored from the district heat supplier.

Responsible for the planning and mounting of the monitoring and control system was the company RKG. All in all seven heat flow meters (Kamstrup Multical 601 Class E1) and seven electricity meters (Kamstrup 162B Class A) have been installed. The same Kipp & Zonen SPlite pyranometer as mounted for the Coolcabin in Graz is used in Gröbming to measure the solar radiation. Moreover the monitored data is sent by a modem to a computer where the data can be read out of a database. The database computer is located at the company Pink in Langenwang.

Due to a combination of monitoring problems during the summer 2009 not enough valuable and comparable monthly data could be collected. Therefore the monitoring data is not included into the IEA SHC Task 38, but still useful system operation experiences were made and shared with the Task 38 community.

3.5.2 Monthly results

Figure 36 and Figure 37 illustrate the monitoring results for August and September 2009. The black line indicates the space cooling power for the Check In Office. The bars show the solar power, the received district heat power, the domestic hot water- and the space heating power.

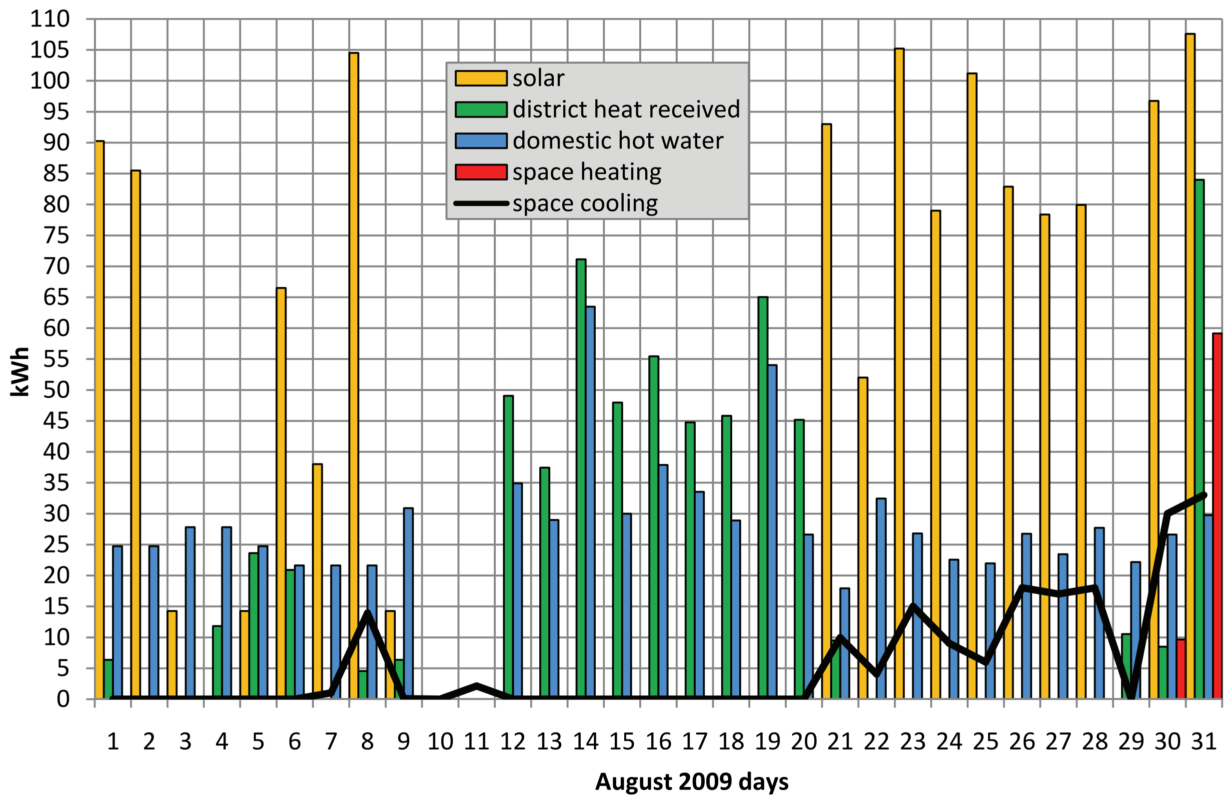


Figure 36: Daily results for the Bachler plant in Gröbming for the month August

Before August 2009 there were no cooling activities of the solar cooling plant. Reasons for not starting the chiller were on the one hand the cold temperatures in May and June especially during night times and on the other hand a not working logical test in the control system. As shown in Figure 34 the old control system included a logical test before enabling the cooling mode. An average value was formed out of the outside temperature of three days. Due to the low night time temperatures in Gröbming this value never cut across the limiting temperature to enable the cooling mode. In the beginning of August this 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 Technische Alternative controllers broke. Moreover the broken controller did control the solar- and the cooling cycle. It took about 10 days to replace the controller and to bring the plant back to fully functionality. From there on 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 situated in 776 m altitude difference where night temperatures fall easily below 10-15°C 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.

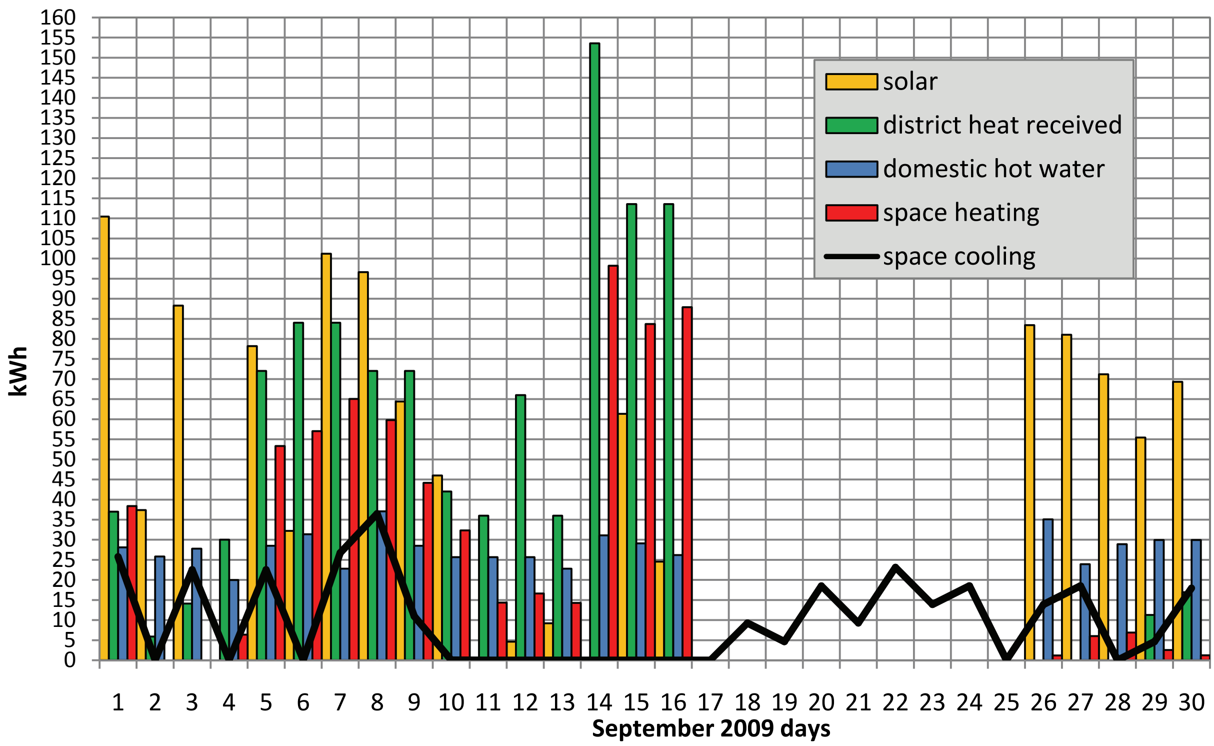


Figure 37: Daily results for the Bachler plant in Gröbming for the month September

The domestic hot water demand (blue bars) has a stable value of around 25 kWh a 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. As shown in Figure 33 the inlet of the district heat flow is not flowing into any storage but instead joining the flow of the DHW or the SH directly. The energy balance of all incoming and outgoing heat flows outlines high overall system losses as shown in Figure 38. An average loss of 28.6 kWh per day is calculated for the period between the 21st of August and the 16th of September 2009.

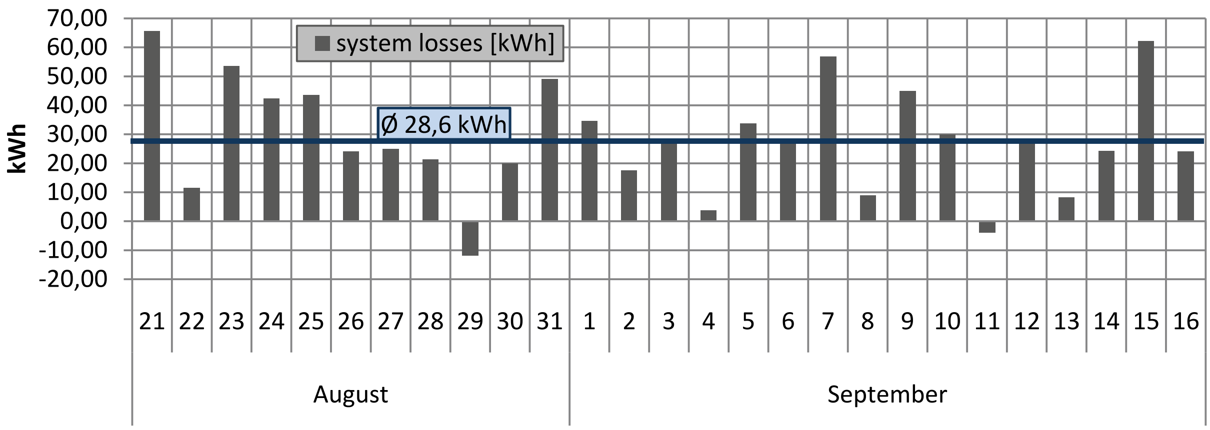


Figure 38: System losses of the solar heating and cooling plant in Gröbming

Further detailed reflections regarding the absorption chiller are made in the next section.

3.5.3 Absorption chiller & daily results

One of the key components in this solar cooling plant is the ammonia water absorption chiller of the company Pink. Figure 39 shows data regarding the chiller at the 5th of September 2009.

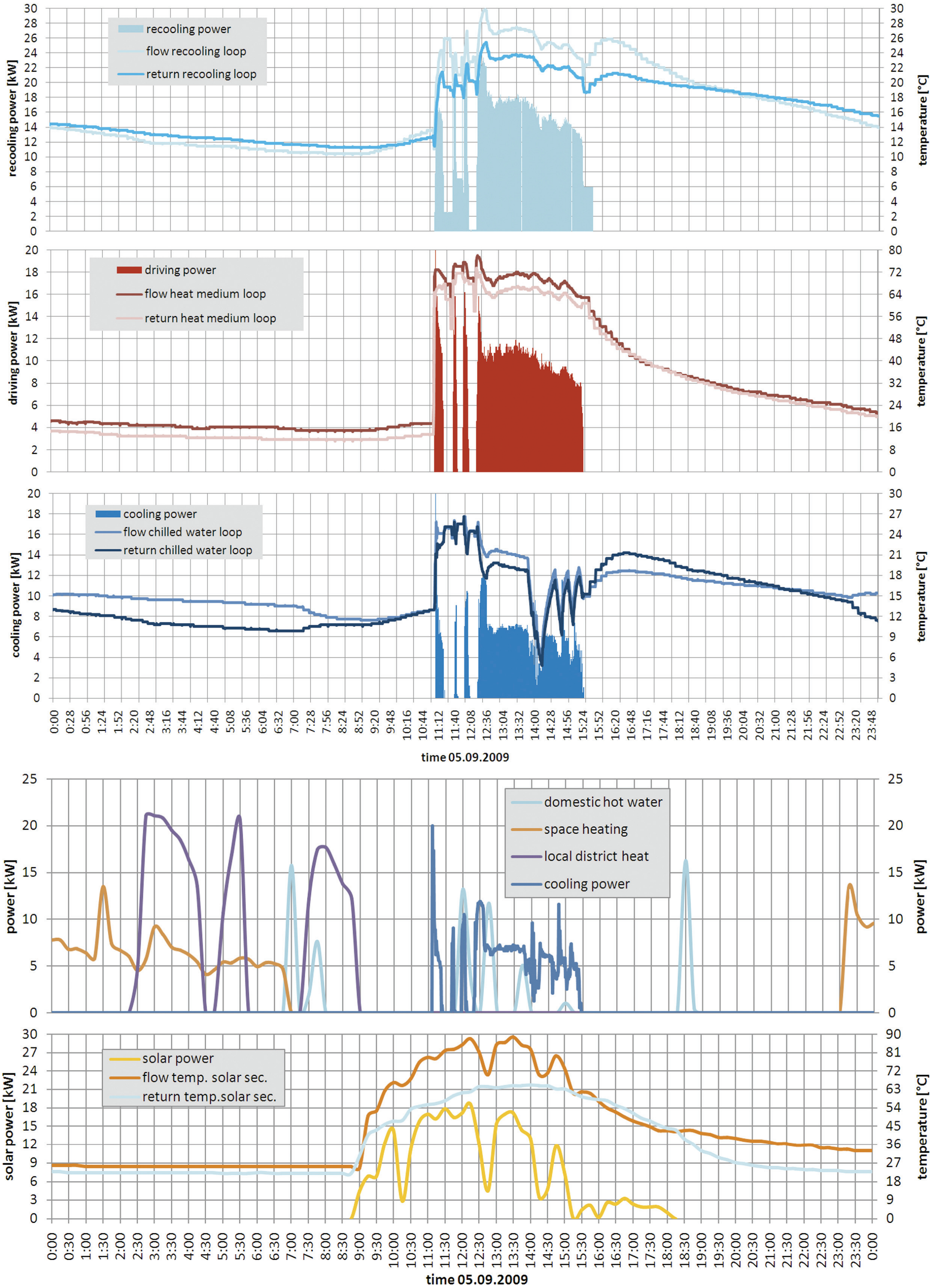


Figure 39: Absorption chiller and system analysis of the Bachler GmbH plant on the 05.09.2009

In the first three diagrams the cooling, driving and recooling power is drafted together with the linked flow and return temperatures. In the beginning the chiller shows a clocking behavior, starting three times before running in a stable condition. The reason for the on and off switching is the insufficient useable heat quantity (with a temperature base over 63°C) in the hot storage. At approximately 12:30 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 (compare 2.6). On this day the chiller has an average power share of around 6/10/16 kW (cooling/driving/recooling). For the recooling chiller inlet temperature stable values of approximately 22°C 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°C for the day.

In the beginning the cooling outlet temperature towards the radiant ceiling shows stable values of approximately 19°C. At around 14:00 the flow and return temperature drops rapidly down to a value of 5°C. After a fast temperature rising up to 18°C 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 a short closed cycle as far as possible until it opens again. These low cooling temperatures affect the chiller performance and are also critical for the costumer due to the danger of condensation at the radiant ceiling. In Figure 40 the thermal COP during the 5th of September 2009 is shown. Three indentations 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.

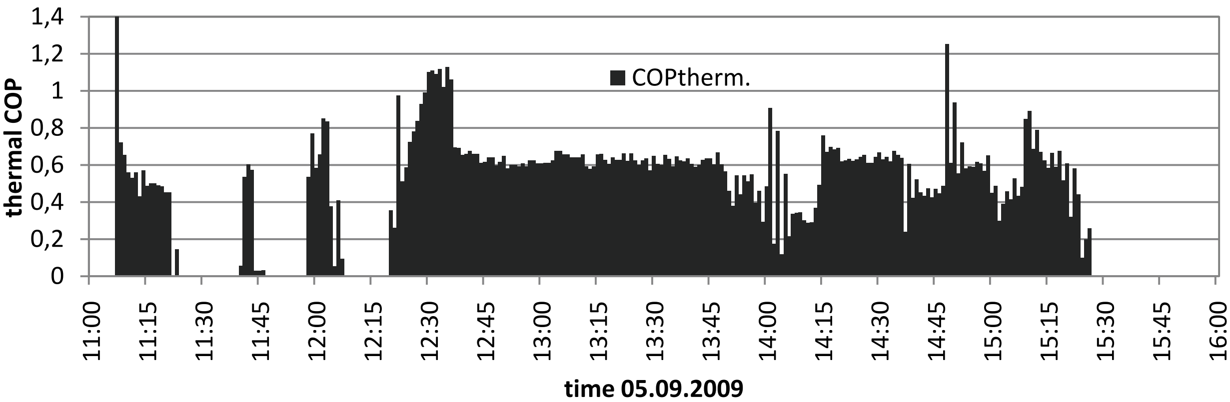


Figure 40: Thermal COP of the Pink chiller on the 5th of September 2009

Thermal and electrical COPs for the period between the 21st of August and the end of September are shown in Figure 41. An average thermal COP for that period of 0.565 could be reached. Electrical COPs range in daily values from 0.5 to 5 and achieve an average value of 3.094. In the period between the 11th and 18th of September no results were monitored due to a data processing problem of the computer system at the Pink GmbH in Langenwang.

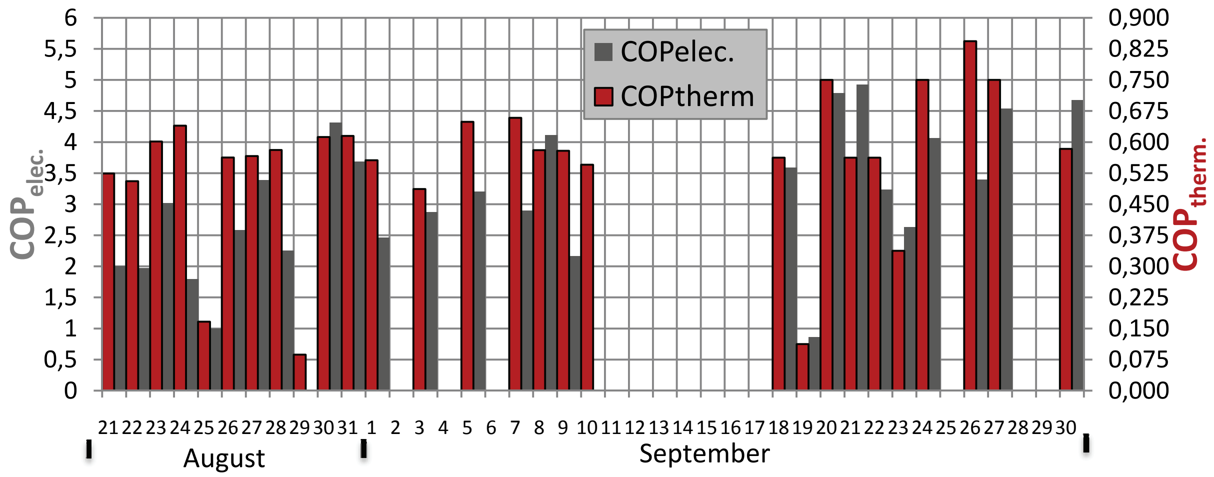


Figure 41: Average daily thermal and electrical COPs for August and September at the Bachler plant

In the following diagrams in Figure 39 the space heating, the domestic hot water and the district heat power are shown. Moreover the solar power of the secondary side as well as the flow and return temperatures are displayed. The solar power is measured after the heat exchanger, which can be seen as input into the hot storage. It rises up to 18 kW at midday with a peak temperature at approximately 90°C. The total solar energy input on that day was 78 kWh. The solar pumps are working in summer times without rpm-regulation. 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. More information to the troubleshooting of this summer can be found under point 3.5.5.

On that same day the space heating was enabled with an average power of 8.6 kW during the night time. The heat can be provided either to the office rooms of the Bachler GmbH. the wellness showroom or to the two apartments in the house. A direct allocation is not possible with the actual monitoring system. The domestic hot water was working in peaks during the morning, midday and evening. In the morning hours the local district heat distributed 72 kWh of heat. The room temperature was not monitored.

3.5.4 Level 3 monitoring result

Due to a combination of monitoring problems during the summer 2009 not enough valuable and comparable monthly data could have been collected. Therefore only the investment costs and further the Cost per Kilowatt (CPK) are shown in Table 9.

Table 9: Investment costs of the solar cooling plant at the Bachler GmbH

name	without VAT	with 20% VAT	unit
solar cooling kit PSC12	30.000.00	36.000.00	€
solar flat collectors	30.833.33	37.000.00	€
installation of cooling kit + material	5.000.00	6.000.00	€
total investment costs I	65.866.33	79.000.00	€
CPK	5.486.00	6.583.00	€/kW_{cold}

3.5.5 Trouble-shooting

During the monitoring period in summer 2009 for the IEA SHC Task 38 some problems did occur including monitoring and control system problems. The main problems shall be described in this work which should enable an experience process for further projects. Most of the problems could be avoided easily in advance if the fields of special sensitiveness are known.

Control problems

The control system as the central regulating unit of the plant is especially delicate to mistakes made in the design phase. It has to be adapted to the special conditions on site. After commissioning a special focus should be set to the control strategy in order to optimize the whole plant. Some problems with the control strategy have been discussed already. such as the broken TA controller and the unsuccessful average value of the outside temperature due to the low night time temperatures in Gröbming.

One problem that did occur during the end of August was linked to low inlet driving temperatures of the chiller coming from the hot storage. High values were shown by the solar heat flow meter, but the absorption machine was switching on and off due to too low temperatures in the hot storage. The failure was connected to the inlet temperature towards the solar collectors. With the speed of the solar pumps P1 and P2 the control system regulated the output temperature to 90°C. The return flow was taken out of the coldest storage, as in winter conditions in order to get a maximum solar yield. So the volume flow was reduced to values of around 200 l/h to reach the 90°C. The power stayed high due to the big temperature difference between T8 and the set point of 90°C. Changing the inlet flow position of the collectors to the lowest outlet of storage 3 (T14) was one part of the solution for this problem. Furthermore the speed-regulation of the solar pumps was changed to maximum speed. These provisions led to a significant higher driving temperature of the chiller and also to fewer starts and stops of the absorption cooling machine.

Another problem that appeared was the interaction between the cooling cycle and the domestic hot water priority rule in the control system. In normal space heating systems including DHW, the domestic hot water always has the highest priority. This means that all distribution pumps are stopped and valves closed in times where domestic hot water is taken. In the case of the plant in Gröbming exactly that happened but the chiller was not stopped and the distribution cycle towards the office was not excluded in the control strategy. As a result of this issue the cooling temperature dropped with the consequence of a low thermal COP. The distribution cycle towards the radiant ceiling has been excluded of the DHW priority rule.
Most important issues of the control system are clear turn on and off rules fitting to the climatically situation as well as to special adoptions of the existing system. Moreover a focused look to the different volume flows is highly recommended.

Monitoring problems

One goal for the summer 2009 was to monitor the plant at the Bachler GmbH in course of the IEA SHC Task 38. Due to different problems in the monitoring procedure this goal could not be fulfilled. Nevertheless some important data of the operation behavior was collected during this summer. Experiences with the plant were made and some improvements were achieved.
The biggest problem of the monitoring configuration is that room temperatures are not monitored. To control the plant the temperatures are measured but not monitored. Therefore it is very difficult to reconstruct the different switching operations because one of the most important parameters is missing. In addition the monitored outside temperature and humidity shows big variations during the day. Figure 42 shows the measured outside temperature and humidity for the 25th of September. The temperature/humidity sensor was exposed to sunlight in the afternoon. Therefore the sensor temperature rises in the late afternoon over 35°C and the humidity drops down to zero. For the next monitoring period the sensor will be moved to a sun protected location and a second sensor will be included as a backup.

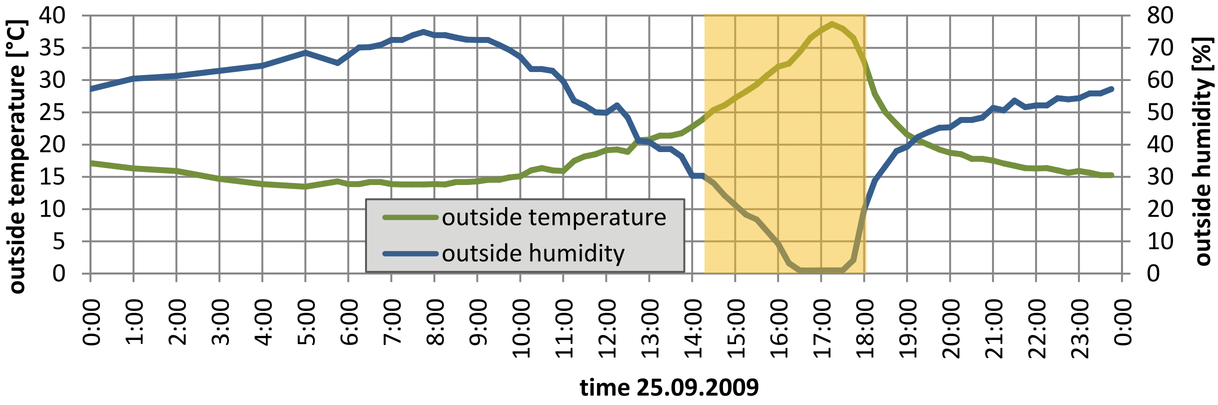


Figure 42: Sunlit temperature and humidity measurement at the solar cooling plant in Gröbming

Another big problem that occurred during this summer is related to varying time intervals of the taken measurements. Due to an unknown reason the different heat flow meters and electricity counters show alternating time intervals during this summer. Some measurements are monitored per minute some in 15 minutes interval and again others per hour. Therefore a comparison is difficult. A clearance with the responsible company RKG will hopefully solve this problem for the next monitoring season.

One last weak point of the monitoring configuration at the Bachler plant is mentioned here. The computer which is responsible for the readout of the monitoring data is not protected against electricity blackouts. In September a blackout in Langenwang was responsible for a 9 day monitoring data loss of nearly all values (compare Figure 37 and Figure 41). As a solution for this critical problem the readout function will be switched to a computer directly at the plant which will be equipped with an UPS (Uninterruptible Power Supply).
As a result of all these problems the monitoring task could not be fulfilled sufficiently for this season.

3.5.6 Analysis of the result & suggestions for improvement

The solar cooling plant in Gröbming only worked between the 21st of August and the end of September. On this account only this period can be analyzed. However the reasons for the standstill have been named under the point 3.5.5.

In the time where the plant was running, it worked quite well. Figure 41 shows daily COPs where it can be seen that on some days the ratios reach sufficient values. Nevertheless the average values over a longer time period are still improvable. From the design point of view the different solar cooling components fit well together even though the solar thermal collectors could have more power. 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. Rising the driving temperature over 75°C and reducing the system losses is a big challenge for the next summer. 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.
The biggest issue of the solar cooling plant in Gröbming is the complicated interaction between the cooling system and all the other parts. For instance Figure 38 illustrates system losses over a certain period of time. The losses include storage losses, ambient losses in the pipeline and interaction between the different components such as heating and cooling on the same day. These losses underline the importance to adopt preinstalled heating control systems when a solar cooling plant is included. Due to the low outside temperatures in night times the heating system started several times during this summer. Therefore a clear division between summer and winter, cooling and heating conditions is necessary. One other interaction between the cooling system and the other components is the above described DHW priority rule which caused low cooling temperatures in the absorption chiller and therefore poor thermal COPs (Figure 40). For this reason the cooling cycle has to be included of this priority rule.

Most of the problems that occurred during this summer 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 discussed previously.
Reengineering the monitoring system includes following five points:

- including the room temperature and several other data points into the monitoring system
- switching the monitoring period of all data points towards minute intervals
- changing the readout function to a UPS protected computer
- repositioning the outside temperature and humidity measurement sensor
- linking the data of the TA controller with the monitoring infrastructure of RKG if possible

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 for the next summer in order to minimize the measurement errors.
Within the last pages practical experience have been described and changes of the system that have been suggested, specified and explained. One goal for the next summer is to reach a stable and reliable operation of the cooling and monitoring system. Furthermore a sustainable improvement of the thermal and electrical COPs is projected.

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 RT2>24°C 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

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 Solar Cool Monitoring project further computer simulations will be carried out.

3.6 General discussion and outlook

During summer 2009 a lot of practical experience has been made with two small scale solar thermal cooling plants in Austria. Generally it can be stated that all specific components worked on their own as expected. For instance both absorption chillers worked fine, following the promises by the manufactures. As well the solar thermal system worked as expected. The overall systems though had significant problems and fulfilled only partly the performance expectations of the planners. One lesson that has been learned from analyzing the practical operation of both plants is that a solid and systematic design process saves a lot of money and time. Machines and plants that have been built already and are running in operation are difficult to change. One thing that can be changed comparably easy is the control system. Therefore a lot of suggestions for changes are related to the control strategy of the plants. It is particularly important to adapt the strategy to the local boundary conditions and make specific adjustments. Fewer failures occur and are easier to find if the physical plant and the control strategy are designed straight forward and in an easy way. More complex systems including seldom used features lead to higher error-proneness and higher customer dissatisfaction.
One major target for future projects in this field has to be the auxiliary electricity consumption. Measured electrical COPs beyond 10 are compulsory in order to outweigh the higher investment costs. Especially the recooling system and all the system supply pumps form a big area of interest. A lot of improvements can be expected when focusing on high electrical COPs already in the design stage. These two plants are expected to reach average electrical COPs between 3 and 7 in the coming summer if further optimizations and improvements are done. Some of the suggested improvements are already implemented.

Another lesson has been learned in the field of monitoring. Especially during long monitoring periods (month and years) an online data access to the plant is recommended. Even if there is a geographical distance between the user of the monitoring system and the plant, errors and measurement black-outs can be identified easier and counteractive measures can be done much faster. Furthermore the operation behavior can be observed and analyzed much easier on an online visualization compared to a table based evaluation. Crucial parameters of the plant (as identified above for the two plants) could be visualized in default diagrams or even in monthly and yearly reports.

In the following chapter a cost comparison between the two described systems and PV driven compression chiller systems is carried out, in order to analyze the market competitiveness of these two plants.

4 COMPARISON OF COSTS

In this chapter a cost comparison of the two main solar cooling methods is done. These two methods are solar thermal cooling and photovoltaic (PV) solar cooling using the electricity produced by a local PV plant. Both methods use solar-energy to fulfill their cooling task. However the comparison only considers economical aspects.

4.1 Introduction & reason for comparison

How much would it cost to cool the two previously described offices with PV-driven compression chillers? This question should be answered within this chapter. To make a fair comparison boundary conditions have to be set in the beginning. Furthermore some assumptions are necessary to be made. PV solar cooling plants are compared to the two existing plants. where most of the data is known such as the investment- and operation costs. Therefore the PV solar cooling plant is dimensioned in a way, to produce exactly the same cooling load as the plant in Graz and Gröbming produced this summer. Ongoing a sensitivity analysis will be done in order to find crucial assumed parameters. Variations of the most sensitive parameters are carried out in 4.3. One reason for the comparison is the lately dropping photovoltaic investment price in Europe. Figure 43 illustrates the average consumer price point between 2006 and now. The price decreased approximately 37% from the beginning of 2006 until the fourth quarter of 2009 with a strong downward trend in the last four quarters. Kaltschmitt and Streicher (2009) show similar numbers for Austria. The prices include the PV-modules, the inverter, miscellaneous components as well as the planning and installation on site.

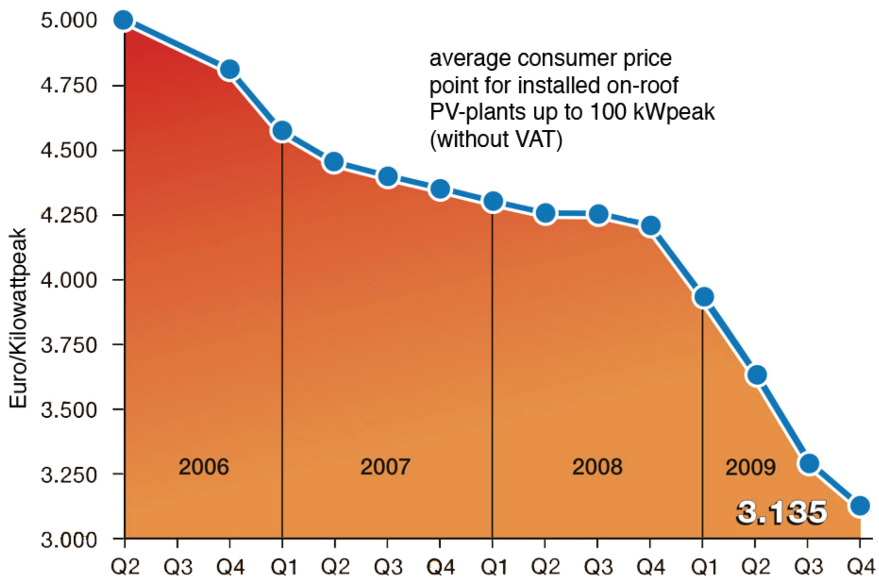


Figure 43: Average consumer price point for installed on-roof PV-plants up to 100 kWp (without VAT) (Bundesverband Solarwirtschaft, 2009)

Another reason for the comparison is an ongoing discussion within the IEA Task 38 community, which puts a special interest to the topic. In order to make a fair comparison, the taken assumptions and boundary conditions are structured and pointed out in the following section.

4.2 Structured boundary conditions and assumptions

The system border for the cost comparison is shown in Figure 44. Inside the system border all components which are related to the cold production are taken into account. The distribution system such as the radiant ceiling or the distribution pump is not included in the comparison. Neither the monitoring systems costs as installed in the current plants are included. All shown prices are overall gross prices including all taxes, dues, insurance, planning and installation costs.

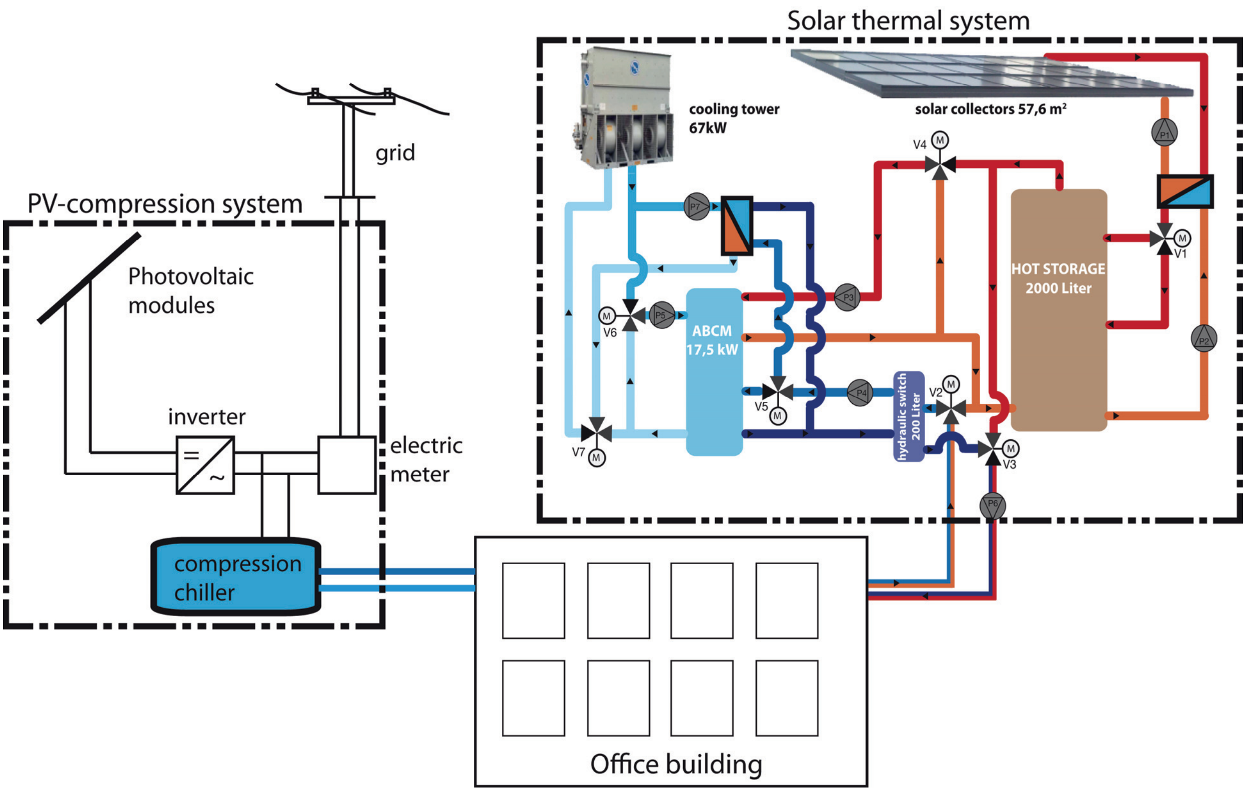


Figure 44: System definition of solar thermal and PV-compression cooling systems in the cost comparison

The analysis presented here is based on the annuity method assuming that the plant is financed through a private loan. All the plant specific gaining such as extra produced electricity in the case of the PV driven compression chiller or the useful solar heat for space heating or domestic hot water at the solar thermal plant are subtracted from the operating costs.

In the whole cost comparison no governmental subsidizes are included. No subsidized feed-in tariffs and no investment subsidies are considered. Neither, needed roof area for the collectors nor installation spaces for the plants are taken into account but are compared and discussed in chapter 4.4. The recycling costs of the plants are neglected. Most of the used numbers related to the solar thermal plants are measured values out of the monitoring summer 2009 (compare chapter 3). For the PV plant up-to-date costs and yield values were collected. A detailed calculation schematic including all steps leading to the value of comparison (annual cold production cost (CPC) in €/kWh) is demonstrated in 4.3. Following points show the boundary conditions and assumptions for the two compared systems and plants.

4.2.1 Solar thermal cooling plants

Table 10 summarizes the assumptions of the two solar thermal plants. The assumed thermal COPs are average expected values for the next summer after optimization of the system. Also the electrical COPs are expected reachable values for the coming summer. These numbers are linked to the existing plants and result out of the practically experiences made.

Table 10: Boundary conditions and assumptions for the solar thermal systems

Solar thermal cooling	Solid Coolcabin	Pink Bachler	units
Absorption chiller power	17.6	12	kW_{cold}
Solar collectors area	58	46	m²
Thermal COP	0.63	0.6	-
Electrical COP	5	5	-
Rate of interest	5	5	%
Term of the loan	15	15	a
Ø district heat prize	0.0918	0.0918	€
Maintenance costs	30	30	€/month
Electricity tariff	0.1507	0.1507	€

The rate of interest is assumed out of typical market rates for the year 2009 in Austria. The lifetime of the total plant was set to 15 years. Counting the heat gains of the solar plant as avoided district heat costs, the average district heat prices are shown for the 4th quarter of 2009 (IWO Austria, 2009). The real average electricity price for households of the last three years (2008-2006) (Austrian Energy Agency, 2009) is calculated for the auxiliary power supply of the plant. Maintenance costs (change of equipment and regular inspections) including the water supply of the cooling tower are assumed at 30 €/month, compare (Streicher, et al., 2009). Values that are assumed in monthly basis are presented in 4.3. Investment costs of the two plants are shown in Table 7 and Table 9.

4.2.2 Photovoltaic solar cooling plants

Table 11 shows the assumptions and boundary conditions for the photovoltaic based system. The PV area and power values are already calculated results out of chapter 4.3 in order to reach at least the same cooling load as the solar thermal cooling plant. The rated COP of the compression chiller is assumed with 2.8 following the IEA Task 38 (Sparber, et al., 2009) taking part loads and practical conditions into consideration. The rated COP of the chosen chiller is 3.14 following the manufacturing information.

Table 11: Boundary conditions and assumptions of the PV-compression chiller systems

PV-compression cooling	Solid Coolcabin	Pink Bachler	units
Compression chiller power	17	12	kW_{cold}
PV-panels area	58.5	40.5	m²
Power PV-panels	6.5	5	kW_{peak}
Chiller COP	2.8	2.8	-
Rate of interest	5	5	%
Term of the loan	15	15	a
Maintenance costs	22	17	€/month
Electricity tariff	0.1507	0.1507	€
Investment costs PV	3915.6		€/kW_{peak}
Investment costs Chiller	300		€/kW_{cold}

Rate of interest, term of the loan as well as the electricity tariff are assumed in a similar way as the solar thermal systems. The maintenance costs are assumed to be 22 €/month for the Solid plant and 17 €/month for the Gröbming plant including insurance and the rental fee for the electrical meter. Kaltschmitt and Streicher (2009) describe annual operational cost of photovoltaic plants depending on size and type between 25 and 40 €/kW_{peak}.

Specific investment costs for the PV installation are shown in Figure 43. The value in Table 11 includes 20% VAT. Similar average consumer prices have been collected for Austria by Kaltschmitt and Streicher (2009). The prices include the inverter, the elevation as well as the installation on site. Henning (2009) shows average specific investment cost of compression chillers including the heat rejection system for small cooling loads at 300 €/kW_{cold}. Specific investment costs have also been determined through an interview with Mr. Keckeis. an experienced proprietor of a HVAC business in Bludenz confirming the value of 300 €/kW_{cold} including VAT and installation for Austria. Therefore the specific price for compression chillers was set to this value.

4.3 Cost comparison

Measurement results collected from the two small-scale solar cooling plants are implicated into the cost comparison presented here. The long-term loan was calculated with a lifetime (N) of 15 years and interest rate (d) of 5% based on the annuity method for both cases. Therefore the initial investments (I), see Table 7 and Table 9, connected with the cooling system (compare Figure 44) are converted into a series of annual payments of equal amounts. The calculation of the cost is done on a monthly basis. Therefore the annual investment payments are converted into 12 equal parts. Equation 4.1 shows the calculation of the monthly investment costs in euro.

$$ic_m = \frac{I \times \frac{d \times (1+d)^N}{(1+d)^N - 1}}{12} \qquad \text{€/month}$$

Equation 4.1

By definition the cooling system is working between May and September. so only in these months a cold load is produced. In the heating season the plants are used either to support the SH and DHW production or to produce electricity in case of the photovoltaic panels. The monthly usable solar heat which is not used for cooling is multiplied by 0.0918 € and calculated as heat revenue. Equally additionally produced electricity is multiplied with the electricity tariff of 0.1507 € and calculated as electricity revenue. The auxiliary electricity costs of the solar thermal cooling plants are calculated through the electrical COP of the plant and the electricity tariff as shown in Equation 4.2. In times were the cooling plant is not working an auxiliary electricity consumption of 5% of the produced usable heat was assumed.

$$ac_m = \left(\frac{cold\ load}{COP_{elec.}}\right) \times Electricity\ tariff \qquad \text{€/month}$$

Equation 4.2

Table 12 and Table 13 show the monthly tabular cost calculation of the two monitored solar cooling plants. All listed prices are overall gross prizes including taxes.

Table 12: Monthly cost calculation of the solar thermal cooling plant Coolcabin

Solid solar thermal	specific usable solar heat	usable solar heat *ush*	effective produced cold load *cl*	heat revenues hr_m	Auxiliary electricity costs ac_m	Operating costs oc_m	Investment costs ic_m	Total costs tc_m
unit	kWh/m^2	kWh	kWh	€	€	€	€	€
January	4.5	259.2	0	23.79	-1.94	-8.15	-1478.85	-1487.0
February	2.8	161.3	0	14.81	-1.21	-16.40	-1478.85	-1495.3
March	8.5	489.6	0	44.95	-3.67	-11.27	-1478.85	-1467.6
April	20.7	1192.3	0	109.45	-8.94	-70.51	-1478.85	-1408.3
May	33.0	1900.8	1197.5	0	-35.93	-65.93	-1478.85	-1544.8
June	34.2	1969.9	1241.0	0	-37.23	-67.23	-1478.85	-1546.1
July	58.6	3375.4	2126.5	0	-63.79	-93.79	-1478.85	-1572.7
August	46.2	2661.1	1676.5	0	-50.30	-80.30	-1478.85	-1559.2
September	21.8	1258.3	792.7	0	-23.78	-53.78	-1478.85	-1532.6
October	10.1	580.0	0	53.25	-4.35	-18.90	-1478.85	-1460.0
November	1.7	97.9	0	8.99	-0.73	-21.75	-1478.85	-1500.6
December	1.0	57.6	0	5.29	-0.43	-25.14	-1478.85	-1504.0
total	243.1	14003.4	7034.2	260.52	-232.31	-331.79	-17746.3	-18078

Table 13: Monthly cost calculation of the solar thermal cooling plant in Gröbming

Pink solar thermal	specific usable solar heat	usable solar heat *ush*	effective produced cold load *cl*	heat revenues hr_m	Auxiliary electricity costs ac_m	Operating costs oc_m	Investment costs ic_m	Total costs tc_m
unit	kWh/m^2	kWh	kWh	€	€	€	€	€
January	4.5	207.0	0	19.00	-1.55	-12.55	-634.25	-646.8
February	2.8	128.8	0	11.82	-0.96	-19.14	-634.25	-653.4
March	8.5	391.0	0	35.89	-2.93	2.96	-634.25	-631.3
April	20.7	952.2	0	87.41	-7.14	50.27	-634.25	-584.0
May	33.0	1518.0	910.8	0	-27.32	-57.32	-634.25	-691.6
June	34.2	1573.2	943.9	0	-28.32	-58.32	-634.25	-692.6
July	58.6	2695.6	1617.4	0	-48.52	-78.52	-634.25	-712.8
August	46.2	2125.2	1275.1	0	-38.25	-68.25	-634.25	-702.5
September	21.8	1004.9	602.9	0	-18.09	-48.09	-634.25	-682.3
October	10.1	463.2	0	42.52	-3.47	9.05	-634.25	-625.2
November	1.7	78.2	0	7.17	-0.59	-23.40	-634.25	-657.7
December	1.0	46	0	4.22	-0.35	-26.12	-634.25	-660.4
total	243.1	11183.3	5350.1	208.06	-177.50	-329.44	-7611.00	-7940.5

Using the results of the calculation above the production cost of one kWh_{cold} can be calculated following Equation 4.3. This value represents the basis of the cost comparison presented here.

$$cold\ production\ costs = \frac{total\ annual\ cost}{eff.produced\ cold\ load} \quad €/kWh$$

Equation 4.3

For the Coolcabin plant in Graz at the Solid office cold production costs of 2.57 €/kWh_{cold} were calculated. The Pink plant in Gröbming at the Bachler GmbH has cold production costs of 1.48

€/kWh_{cold}. The big difference comes about the huge difference in the initial investment costs. It can be said that the Coolcabin was built as a demonstration and test plant directly at the company's office contrary to the Pink plant which was sold to the costumer Bachler GmbH. All results are compared in Figure 45. In Table 14 and Table 15 the similar tabular calculation was done for the PV-driven compression variant.

Table 14: Tabular cost calculation of the PV-compression chiller cooling system for the Solid office

Solid PV-driven	Ø PV-yield 2004-2009	Produced electricity	effective produced cold load	not used produced electricity	electricity revenues er_m	Operating rev./costs oc_m	Investment costs ic_m	Total costs tc_m
unit	kWh/kW_{peak}	kWh	kWh	kWh	€	€	€	€
January	26.8	174.4	0	174.4	26.16	4.16	-245.28	-241.12
February	40.7	264.3	0	264.3	39.65	17.65	-245.28	-227.63
March	74.7	485.3	0	485.3	72.80	50.80	-245.28	-194.48
April	111.7	726.9	0	726.9	109.04	87.04	-245.28	-158.25
May	126.0	819.0	1197.5	391.3	58.70	36.70	-245.28	-208.58
June	128.0	832.0	1241.0	388.8	58.32	36.32	-245.28	-208.97
July	126.3	821.0	2126.5	61.7	9.26	-12.74	-245.28	-258.03
August	111.3	723.7	1676.5	124.9	18.74	-3.26	-245.28	-248.55
September	93.3	606.7	792.7	323.6	48.53	26.53	-245.28	-218.75
October	63.8	414.9	0	414.9	62.24	40.24	-245.28	-205.05
November	25.0	162.5	0	162.5	24.38	2.38	-245.28	-242.91
December	19.2	124.6	0	124.6	18.69	-3.31	-245.28	-248.60
total	947.0	6155.5	7034.2	3643.3	546.49	282.49	-2943.39	-2660.9

Table 15: Tabular cost calculation of the PV-compression chiller cooling system for the Bachler office

Bachler PV-driven	Ø PV-yield 2004-2009	Produced electricity	effective produced cold load	not used produced electricity	electricity revenues er_m	Operating rev./costs oc_m	Investment costs ic_m	Total costs tc_m
unit	kWh/kW_{peak}	kWh	kWh	kWh	€	€	€	€
January	26.8	134.2	0	134.2	20.13	3.13	-186.09	-182.96
February	40.7	203.3	0	203.3	30.50	13.50	-186.09	-172.59
March	74.7	373.3	0	373.3	56.00	39.00	-186.09	-147.09
April	111.7	559.2	0	559.2	83.88	66.88	-186.09	-119.21
May	126.0	630.0	910.8	304.7	45.71	28.71	-186.09	-157.38
June	128.0	640.0	943.9	302.9	45.43	28.43	-186.09	-157.65
July	126.3	631.7	1617.4	54.0	8.11	-8.89	-186.09	-194.98
August	111.3	556.7	1275.1	101.3	15.19	-1.81	-186.09	-187.90
September	93.3	466.7	602.9	251.3	37.70	20.70	-186.09	-165.38
October	63.8	319.2	0	319.2	47.88	30.88	-186.09	-155.21
November	25.0	125.0	0	125.0	18.75	1.75	-186.09	-184.34
December	19.2	95.8	0	95.8	14.38	-2.63	-186.09	-188.71
total	947.0	4735.0	5350.1	2824.2	423.64	219.64	-2233.02	-2013.4

The specific average photovoltaic yield for Germany between 2004 and 2009 *(www.pv-ertraege.de, 2009)* was assumed as a pessimistic yield for the PV panels in Graz and Gröbming. The power of the PV plant was calculated in a way that the exactly same cooling load as the solar thermal plant can be produced. The compression chiller was dimensioned for the same power as the absorption chiller. For the PV driven compression cooling system in Graz cold production costs of 0.38 €/kWh were calculated. The PV variant in Gröbming reaches nearly the same price with the smaller plant. Figure 45 illustrates the annual cold production costs for the two monitored systems compared with PV driven and conventional cooling systems. The results show huge cost differences between the solar thermal and the other two cooling systems. In case of the Bachler plant the solar thermal produced kWh_{cold} costs nearly 4 times more than the kWh_{cold} produced with the PV driven solar cooling system.

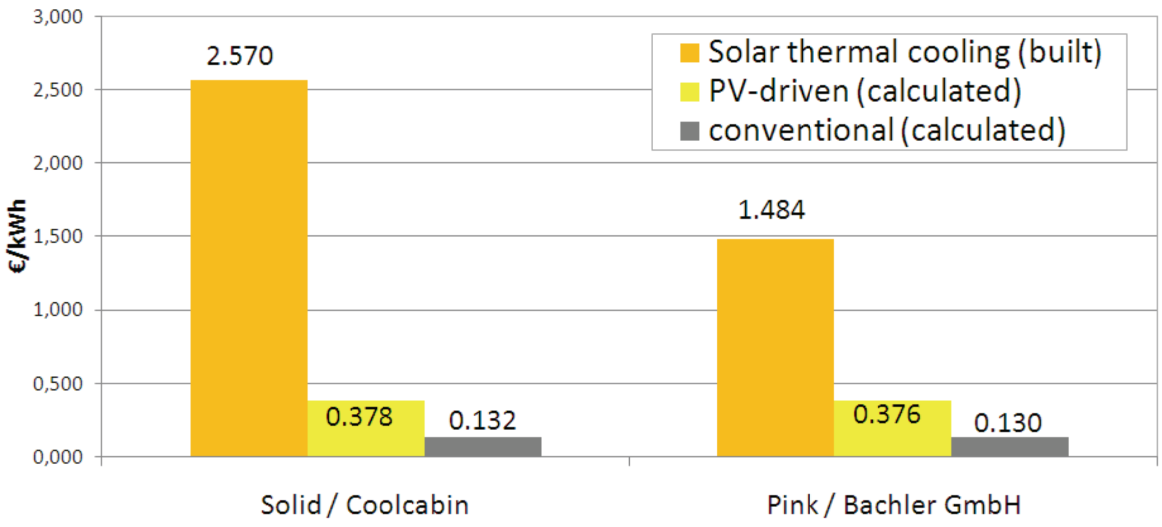


Figure 45: Cold production costs for the two monitored systems compared with PV driven and conventional cooling systems

Compared to the conventional system in Gröbming approximately 11 times higher costs per kWh evolve. The conventional cooling costs in this figure were calculated with the same boundary conditions as the PV driven compression chiller excluding the photovoltaic costs and revenues. Figure 45 shows that the first generation small scale solar thermal cooling plants in Austria are not economically competitive to comparable renewable cooling technologies such as the PV driven compression chiller systems. The next two figures (Figure 46, Figure 47) show a sensitivity analysis of the solar thermal and the PV driven cooling plant cost calculations for the location in Gröbming at the Bachler GmbH.

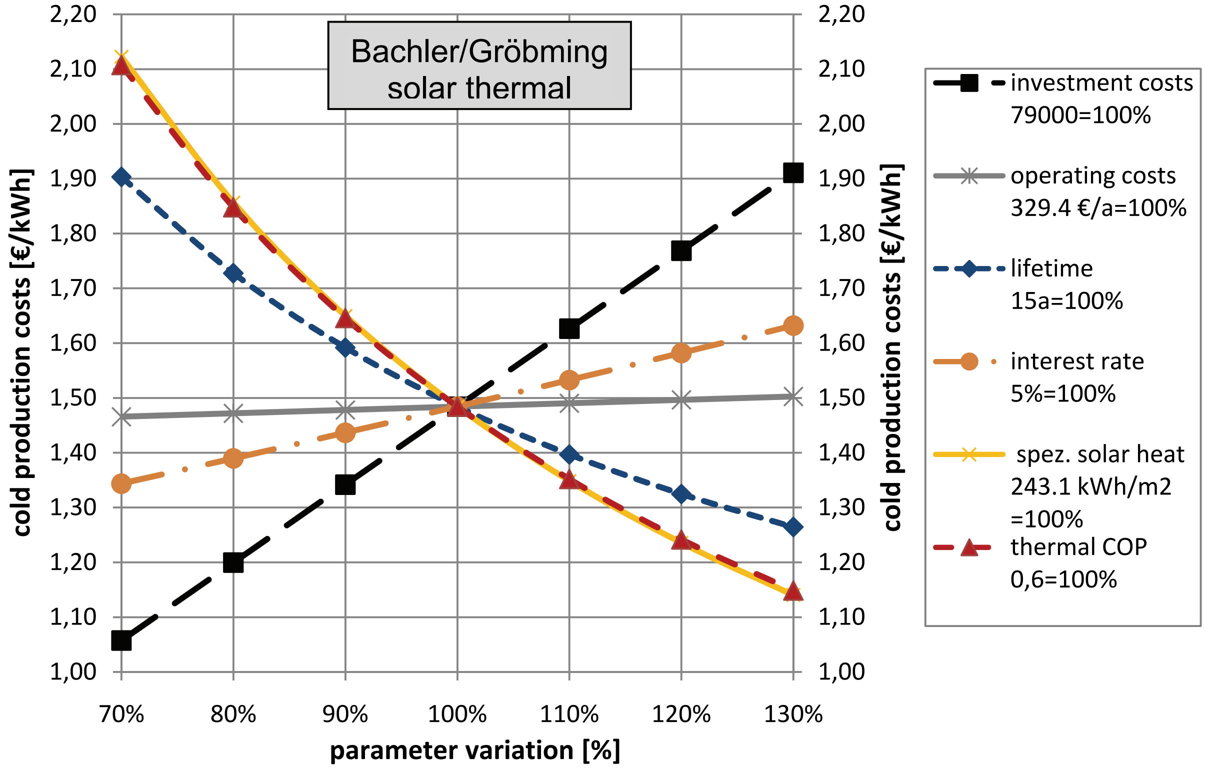


Figure 46: Parameter variation analysis of the solar thermal cost calculation at the Bachler GmbH

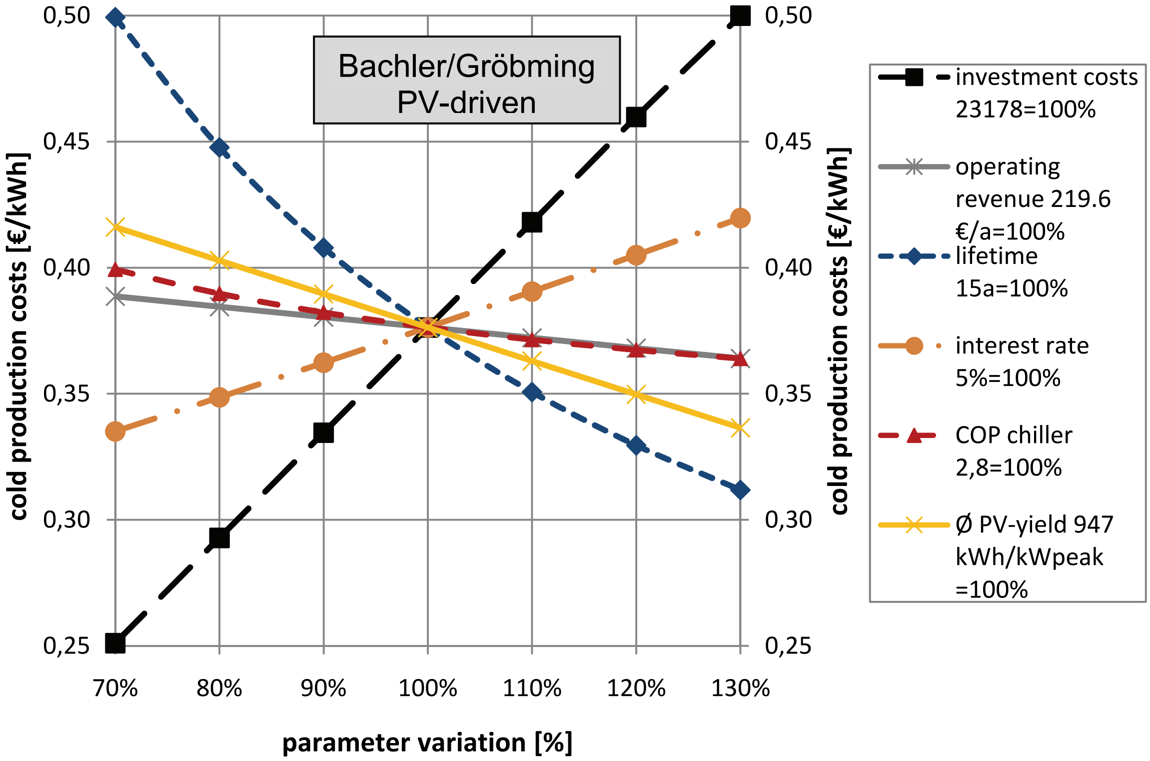


Figure 47: Parameter variation analysis of the PV-driven cost calculation in Gröbming

In those two diagrams the most important parameters for the cost calculation are identified. For the solar thermal part the specific usable solar heat, the thermal COP of the absorption chiller and the investment costs are the most important values. The operating costs and the interest rate have hardly any influence to the cold production costs. All sensitive parameters are measured or known values. Therefore the calculation can be rated as quite feasible. In case of the PV driven system the most influencing parameters are the investment costs and the lifetime. The same lifetime has been chosen for all variants so it has no influence to the comparison. The investment costs are assumed based on profound source values described under 4.2.2. Hence the cost comparison presented in this work can show a general cost position of the different systems for small scale cooling application in Austria. The sensitivity analysis also shows that low system losses and therefore higher specific usable solar heat as well as higher thermal COPs are crucial for solar thermal cooling plants. In this work only results of the parameter variation of the Bachler plant are shown. The variations of the Coolcabin parameters show similar result.

4.4 Discussion of the results

In the beginning of this chapter the question was asked, how much a PV driven compression chillers cooling system would cost compared to the built solar thermal system. The answer to this question is shown in Figure 45, comparing the cold production cost of the different variants. The results show that these two small scale solar thermal cooling plants are not economically competitive compared to other renewable cooling technologies such as the PV driven compression chiller system. Also a comparison to conventional cooling systems was done. It can be stated that in this cost calculation the renewable cooling systems have between 3 and 20 times higher cold production costs compared to the conventional system. The striking cost difference between the two renewable systems can be explained by different reasons. Both systems combine components that are already in a certain maturity phase. The PV driven compression chiller system benefits from the low specific chiller costs and the recently dropped photovoltaic costs. In contrast the solar thermal system suffers of high specific investment costs for numerous components, such as the absorption chiller and the solar thermal collectors. Also the more complex overall system can be considered as a burden for this technology. In the year 1983 an investment cost comparison of solar driven cooling systems showed nearly double investment costs for photovoltaic based systems compared to solar thermal driven systems *(Podesser, 1983)*. Figure 43 shows the tremendous price development during the last years and even much more compared to 1983.

Some criteria were not taken into account in this comparison, such as the space needed to install the various systems. The photovoltaic panels though need nearly the same area as the solar thermal collectors. For the PV driven plant at the Solid office an area of 58.5 m² was calculated for 6.5 kW_{peak}. Compared to the solar thermal collector area of 57.6 m² just a slightly bigger area is needed for the PV. The space which is needed for the chiller and all auxiliary components can be estimated much higher for the solar thermal system because of the higher complexity of the system as well as the bigger number of components.

Furthermore the cost results presented here are only valid for small systems. In the case of bigger systems cost reduction and rising competiveness can be expected for the solar thermal technology. Concluding the cost comparison it can be stated that solar cooling for small and medium cooling applications in Austria is better done with photovoltaic driven compression chiller systems rather than with solar thermal based systems.

5 SUMMARY AND CONCLUSION

Cooling and air conditioning is one of the major energy consuming services in buildings. Due to the ongoing worldwide increase of electricity consumed by conventional air conditioning units the renewable technology solar cooling gained much attention in the previous years. Several companies and research institutions are investigating in this area. The overall platform for bundling the research and development work is the IEA SHC Task 38. One part of this task is to monitor existing solar cooling plants in order to gain practical experience and operation behaviors. Two small scale solar cooling plants were monitored during the summer 2009 and the monitoring results for those two plants were presented. The measurements were monitored on the comparable level 3, following the IEA SHC Task 38. Furthermore all practical experiences gained during summer 2009 are shared and described in this work.

The plant of Solid GmbH in Graz was monitored successfully during the summer 2009. The measurement results showed some weaknesses of the plant design as well as a too small dimensioned radiant ceiling in the building. Especially the electrical consumption of the auxiliary equipment led to poor electrical COPs and therefore to a negative primary energy ratio. Following the results of Figure 30 and Figure 32 neither primary energy savings were realized during the entire summer nor could monthly electrical COP reach a value of 2. Nevertheless a remarkable optimization potential is expected within the control system and the heat rejection unit. Due to the fact that the cooling tower including the pumps of the recooling loop is responsible for nearly 80 % of the overall electricity consumption there is a call for action especially in this field. One suggested way to tackle this matter is to replace the existing cooling tower by a smaller and more efficient one or to control the fan speed of the existing cooling tower in combination with an adapted control strategy. Furthermore the results also indicate that the free cooling mode is not working in a sufficient way. The plant is consuming a lot of electrical energy when running on the mode. Compared to these operating expenses the produced cooling load is rather low. Thus it is not justifying running the free cooling mode on weekends and during night times when the office is not used. The suggested approach to this matter is to stop the free cooling mode completely or to adapt the timeframe of the mode with the working office hours. Operational aspects of the monitoring system were throughout positive. The online monitoring system worked reliable and was easy to handle. Further measurement points will be installed for the next summer in order to measure the inside and outside humidity. The suggestions of improvement regarding the Solid plant also include a reduction of the radiant ceiling inlet temperature from now 16°C to 15°C if the condensation sensors still allow a smooth and secure operation. If these optimizations and improvements for the plant are done, expected electrical COPs could range between 3 and 7 for the coming summer.

The second plant which was monitored during this summer was the Pink plant in Gröbming at the Bachler GmbH. Due to several monitoring and control strategy problems during this summer no comparable and representative data were collected. Mistakes in the control strategy and a broken controller were the main reasons for nearly no cooling before the end of August. The biggest issue of the solar cooling plant in Gröbming is the complicated interaction between the cooling system and all the other parts. High system losses shown in Figure 38 and space heating during this summer due to the low outside temperature in night times are just some problems of the plant. Therefore a clear division between summer and winter, cooling and heating conditions in the control strategy is necessary. Furthermore including the cooling cycle of the DHW priority rule will avoid too low cooling cycle temperatures caused by this rule. Figure 39 shows that the absorption chiller starts several times before it is running for a longer period. Mismatching volume flows of the solar cycle with the volume flow of the machine as well as too low driving temperatures lead to this problem. One suggested way to approach this problem is to rise the starting driving temperature in the control strategy. Additionally to the suggestions of improvement for the plant (listed in 3.5.6) the monitoring system did not work as expected. Thus the system will be reengineered in order to ensure a reliable monitoring period during the next summer. Some of the changes are including the room temperature and other important data points. changing the read out function to a UPS secured computer direct at the plant and repositioning the outside temperature measurement senor. If the suggested changes are implemented a reliable operation and satisfactory COPs for the next summer are projected.

Moreover a cost comparison of the two solar thermal cooling plants to photovoltaic (PV) driven compression chillers was carried out. Main assumptions for this calculation were not including any governmental subsidies. a lifetime of 15 years for both plants, rate of interest of 5 %. COP of the compression chiller of 2.8, an electricity tariff of 0.15 €/kWh, district heat cost of 0.09 €/kWh and assumed maintenance costs between 17 and 30 €/month. On the basis of the average consumer price point showed in Figure 43 the investment cost for the PV modules were calculated. The results show 4 to 7 times higher cold production costs (€/kWh) of the solar thermal plants compared to the PV driven systems and even 11 to 19 times higher production costs compared to the conventional compression chiller variant. In Figure 46 and Figure 47 results of the parameter variation are shown. The most important parameters are the investment costs, the solar yield as well as the thermal COP of the absorption chiller.

This shows that the economical performances of the two solar thermal systems in Austria are not competitive to other cooling systems. Furthermore the cost results presented here are only valid for small systems. In the case of bigger systems cost reduction and rising competiveness can be expected for the solar thermal technology. Concluding the cost comparison it can be stated that solar cooling for small and medium cooling applications in Austria is better done with photovoltaic driven compression chiller systems rather than with solar thermal based systems.

Due to the restricted time of the diploma thesis some working targets still have to be done. Therefore an outlook of further tasks related to those two plants shall be provided here. The two plants will be monitored during the next summer (2010). Therefore the implementation of the suggested changes as well as the further monitoring of both plants is important. Within the Solar Cool project which started in November 2009 the two plants will be simulated in order to compute possible physical changes of the plants. A comparison and systematic analysis of the monitoring results measured this summer to the ones of next summer will bring detailed results in order to estimate further improvement and optimization potentials.

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LIST OF ABBREVIATIONS

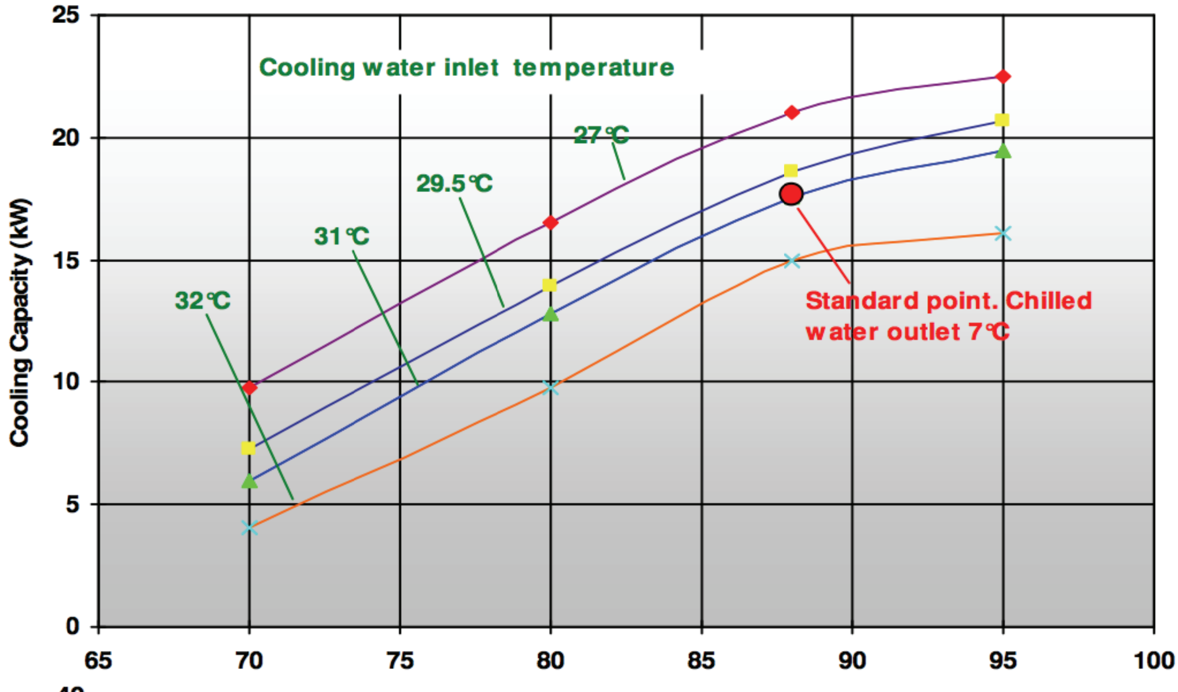
ACM	Absorption Cooling Machine
CAC	Central Air Conditioning unit
COP	Coefficient of Performance
CPC	Cold Production Costs
CT	Cooling Tower
CPK	Cost Per Kilowatt
DHW	Domestic hot water
WMZ	Heat Flow Meter
HVAC	Heating Ventilation & Air Condition
IEA	International Energy Agency
OT	Outside Temperature
PV	Photovoltaic
PAC	Portable Air Conditioning unit
PER	Primary Energy Ration
RAC	Room Air Conditioning unit
RT	Room Temperature
RPM	Rounds Per Minute
SH	Space Heating
TA	Technische Altenative
UPS	Uninterruptible Power Supply
VAT	Value Added Tax

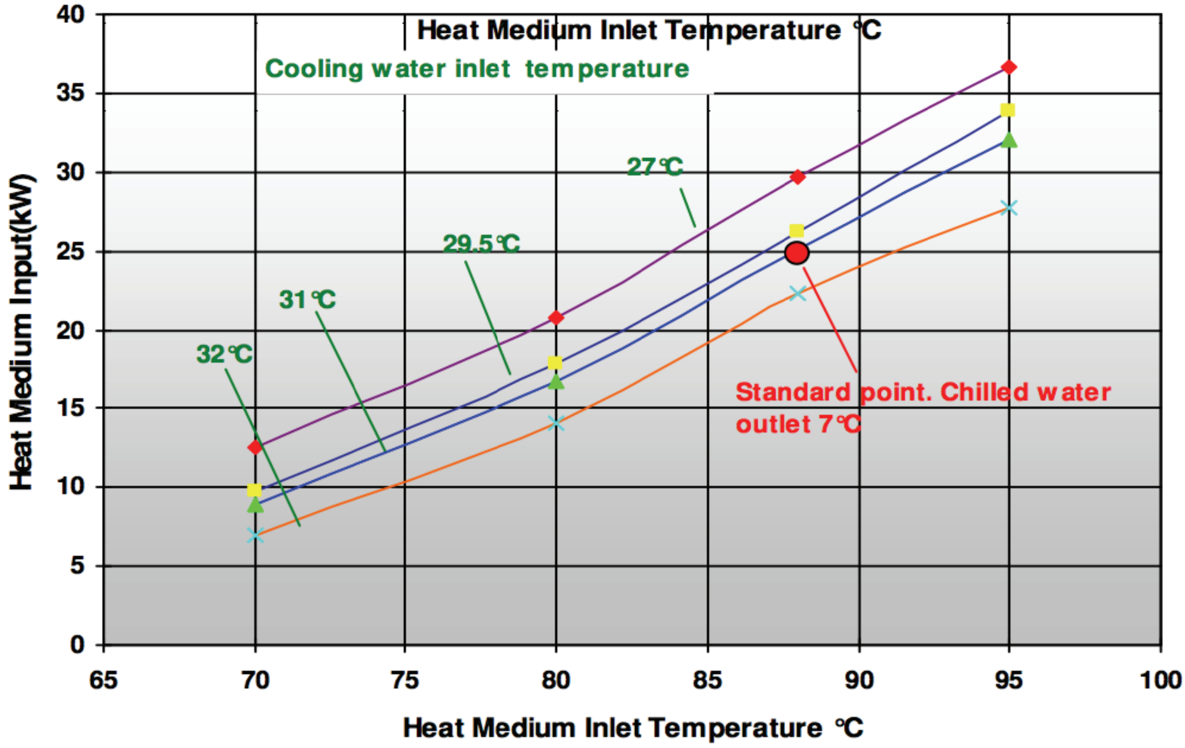
PROJECT SCHEDULE

24.07.2009	•diploma-thesis kick-off meeting
July-August 2009	•literature research •first visit of the Solid plant in Graz
12.08.2009	•first visit of the Bachler plant in Gröbming
26.08.2009	•successful failiure search and optimization at the Bachler plant
August-September 2009	•analyzing data of the Solid plant •trouble shooting at the Bachler plant
28.09.09-2.10.09	•IEA Task 38 Meeting in Palermo, Italy •presenting the Solid plant in Palermo at the Task meeting •OTTI 3rd International Solar cooling conference in Palermo, Italy
October 2009	•analyzing the Gröbming measurement & monitoring data •work out suggestions of improvements for both plants
21.10.2009	•meeting for implementation of improvents & concluding discussion
November 2009	•writing the diploma thesis •handing over the project
10.11.2009	•concluding discussion of the Coolcabin at Solid in Graz •presenting the suggestions of improvement
04.12.2009	•final presentation at the Seminar Wärmetechnik in front of the IWT
December 2009	•end of the project •debriefing •printing the diploma thesis

APPENDIX

Characteristic curves YASAKI WFC-SC5 (Yasaki, 2009)

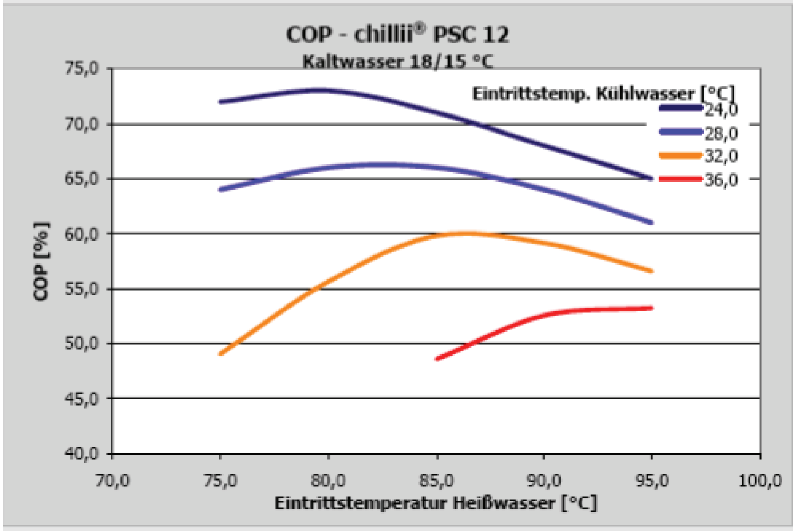




Characteristic curves Pink Chillii PSC 12 (Pink GmbH, 2009)

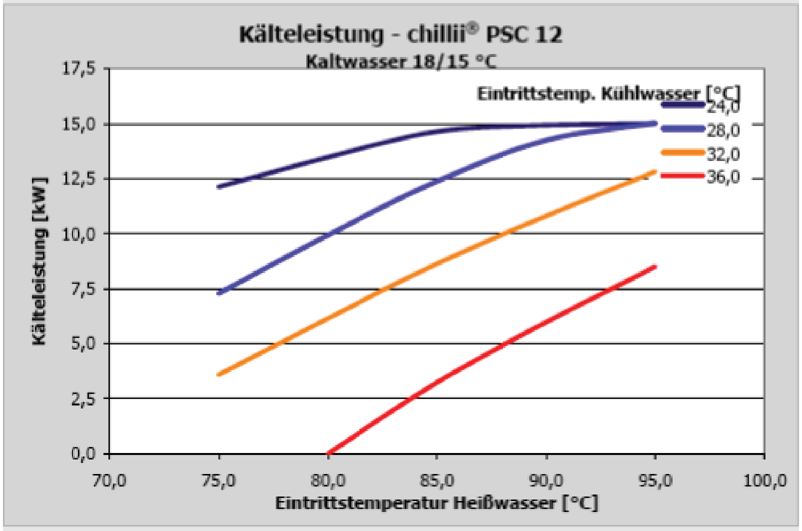
COP - chillii® PSC 12
Kaltwasser 18/15 °C

Kühlkreislauf		Heizungskreislauf				
	Ein	75,0	80,0	85,0	90,0	95,0
Ein	Aus	68,0	73,0	78,0	83,0	88,0
24,0	29,0	72,0	73,0	71,0	68,0	65,0
28,0	33,0	64,0	66,0	66,0	64,0	61,0
32,0	37,0	49,1	55,6	59,8	59,1	56,6
36,0	41,0	-	-	48,6	52,5	53,2



Kälteleistung - chillii® PSC 12
Kaltwasser 18/15 °C

Kühlkreislauf		Heizungskreislauf				
	Ein	75,0	80,0	85,0	90,0	95,0
Ein	Aus	68,0	73,0	78,0	83,0	88,0
24,0	29,0	12,1	13,5	14,6	14,9	15,0
28,0	33,0	7,3	9,9	12,4	14,2	15,0
32,0	37,0	3,6	6,2	8,6	10,8	12,8
36,0	41,0	-	-	3,2	6,0	8,5



Ammonia – water absorption systems below 20 kW chilling capacity
(Jaehning, et al., 2009)

Manufacturer		**AOSOL**	**Pink GmbH**	**Robur**
country		Portugal	Austria	Italy
Model name / number			Chillii PSC12	ACF60-00 LB
Contact data of supplier		Ao Sol. Energias Renováveis. SA. Parque Industrial do Porto Alto. Portugal. Email: aosol@aosol.pt Telf: +351 263 651 305	Pink GmbH Bahnhofstraße 22 8665 Langenwang Austria Tel: +43 (0)3854/3666	Robur S.p.A. Via Parigi 4/6 24040 Verdellino/Zingonia (Bg) Italy
internet		www.aosol.pt	www.pink.co.at	www.robur.com
other supplier / model name			SolarNext / chillii PSC12	
Basic information	**unit**			
technology		ammonia-water	ammonia-water	ammonia-water
nominal chilling power	[kW]	8	12	12
nominal COP		0.6	0.65	
heat rejection type		internal	external	internal
intended area of application		air-conditioning	air-conditioning	process cooling
development stage		pre-commercial	pre-commercial	pre-commercial
specification sheets available (yes/no)		not yet	Yes	restricted
(expected) investment costs	[€]	5000	n.a.	

Nominal Operating conditions					
Driving circuit	driving power	[kW]	13.3	18.5	n.a.
	operating temperature (in / out)	[°C]	96 / 86	75 / 68	240
	temperature range (from-to)	[°C]	80-110	75..85	180..240
	heat transfer fluid[5)]		water	water	diathermic oil
	flow rate	[l/h]	850	2300	3500
	operating pressure	[bar]	2	n.a.	n.a.
	pressure drop in circuit	[Pa]	0.5	n.a.	n.a.
Heat rejection circuit	re-cooling power	[kW]	21.3	30.5	n.a.
	re-cooling temperature (in / out)	[°C]	35 (air temperature)	24 / 29	35 (air temperature)
	temperature range (from-to)	[°C]	30-42	24	-12 ... 45
	heat transfer fluid[5)]		air	water	air
	flow rate	[l/h]	6300	5200	
	operating pressure	[bar]	-		
	pressure drop in circuit	[Pa]	-		
Chilling circuit	chilling power	[kW]	8	12	12
	chilling temperature (in/out)	[°C]	14 / 9	18 / 15	0 / -5
	temperature range (from-to)	[°C]	7-16	6..15	-10 ... 45
	heat transfer fluid		water	water	brine 40% glycol
	flow rate	[l/h]	1030	3400	2600
	operating pressure	[bar]	2	n.a.	3
	pressure drop in circuit	[Pa]	500	n.a.	400
Parasitic demand	electrical power	[kW]	0.5	0.3	0.84
	water consumption	[/s]	-		none
Dimensions & weight	Length / width / height	mm	n.a.	800 / 600 / 2200	890 / 1230 / 1290
	Weight in operation	kg		350	370

Water - lithium bromide absorption systems below 20 kW chilling capacity
(Jaehning, et al., 2009)

Manufacturer		**Sonnenklima GmbH**	**EAW**	**Yazaki**	**Rotartica**	**Rotartica**
country		Germany	Germany	Japan	Spain	Spain
Model name / number		suninverse	Wegracal SE 15	WFC-SC 5	045V	045
Contact data of supplier		SK SonnenKlima GmbH Am Treptower Park 28-30 D 12435 Berlin Tel: +49 30 53 0007 700 Fax: +49 30 53 00 07 17	EAW Energieanlagenbau Westenfeld GmbH Oberes Tor 106 98631 Westenfeld Telefon: 036948 84-132 Telefax: 036948 84-152 info@eaw-energieanlagenbau.de	Yazaki Europe Ltd. Environmental and Energy Equipment Operations Robert-Bosch-Strasse 43. 50769 Köln (Cologne). Germany Phone: (49) 221-59799-0 Fax: (49) 221-59799-197 Email: info@yazaki-airconditioning.com	Avda. Cervantes 45. 48970 Basauri (Bizkaia) Spain Tel: (+34) 94 402 51 20 Fax: (+34) 94 402 51 21 E-mail: rotartica@rotartica.com	
internet		www.sonnenklima.de	www.eaw-energieanlagenbau.de	www.yazaki-airconditioning.com	www.rotartica.com	
Basic information	**unit**					
technology		absorption water-LiBr	absorption water-LiBr	absorption water-LiBr	absorption water-LiBr	absorption water-LiBr
nominal chilling power	[kW]	10	15	17.6	4.5	4.5
nominal COP		0.78	0.71	0.7	0.62	0.62
heat rejection type		external	external	external	integrated	external
intended area of application		domestic. commercial	air-conditioning	air-conditioning	domestic	domestic
development stage		ID / N	commercial	commercial	Not available	Not available
specification sheets available (yes/no)		yes	yes	yes	yes	yes
(expected) investment costs	[€]	n.a.	15.000		n.a.	n.a.

Nominal Operating conditions							
Driving circuit	driving power	[kW]	13.6	21	25.1	6.7	6.7
	operating temperature (in / out)	[°C]	75 / 65	90 / 80	88 / 83	90 / 83	90 / 83
	temperature range (from-to)	[°C]	75 - 95		70 - 95	80-105	80-105
	heat transfer fluid		water	water	water	water	water
	flow rate	[l/h]	1200	1800	4320	1200	1200
	operating pressure	[bar]	<= 2.5	< 6	< 5.88	1.5	1.5
	pressure drop in circuit	[mbar]	200	400	770	200	200
Heat rejection circuit	re-cooling power	[kW]	24	35	42.7	11.7	11.7
	re-cooling temperature (in/out)	[°C]	27/35	30 / 36	31 / 35	40	40
	temperature range (from-to)	[°C]	20 - 35 (approx.)			25-45	25-45
	heat transfer fluid		water. open cycle	water	Water	Air	water
	flow rate	[l/h]	2600	5000	9180		1980
	operating pressure	[bar]		6	5.88		1.5
	pressure drop in circuit	[mbar]	320	900	383		1116
Chilling circuit	chilling power	[kW]	10	15	17.6	4.5	4.5
	chilling temperature (in/out)	[°C]	18 / 15	17 / 11	7 / 12.5	16	16
	temperature range (from-to)	[°C]	6 - 15			8-22	8-22
	heat transfer		water	water	Water	water	water

	fluid[5)]						
	flow rate	[l/h]	1300 - 2900	1900	2770	1200	1200
	operating pressure	[bar]	< 2.5	< 6	< 5.88	1.5	1.5
	pressure drop in circuit	[mbar]	350	400	526	300	300
parasitic demand	electrical power	[kW]	0.12	0.3	0.048	1.2 incl. fan	0.4
	water consumption	[/s]	none	none	None	none	none
Dimensions & weight	Length / width / height	mm	795 / 1130 / 1960	1750 / 760 / 1750	594 / 744 / 1736	1092 / 760 / 1150	1092 / 760 / 1150
	Weight in operation	Kg	550	660	420	290	290

Other technologies below 20 kW chilling capacity (Jaehning, et al., 2009)

Type of technology		**absorption water-LiCl**	
Manufacturer		**Climatewell**	
country		Sweden	
Model name / number		Climatewell 10	Climatewell 20
Contact data of supplier		ClimateWell AB Instrumentvägen 20 126 53 Hägersten Stockholm Sweden info@climatewell.com	
internet		www.climatewell.com	
other supplier / model name			
Basic information	**unit**		
nominal chilling power	[kW]	4	n.a.
nominal COP		0.68	0.68
heat rejection type[3)]		external	external
intended area of application		residential	residential
development stage [4)]		commercial	commercial
specification sheets available (yes/no)		yes	yes
(expected) investment costs	[€]	7500	n.a.

Nominal Operating conditions				
driving circuit	driving power	[kW]	n.a.	n.a.
	operating temperature (in/out)	[°C]	80 / 70	80 / 70
	temperature range (from-to)	[°C]	60-120	60-120
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	900	1500
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	200	450
Heat rejection circuit	re-cooling power	[kW]	n.a.	n.a.
	re-cooling temperature (in/out)	[°C]	30 / 40	30 / 40
	temperature range (from-to)	[°C]	20-40	20-40
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	1800	3000
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	250	580
chilling circuit	chilling power	[kW]	4	n.a.
	chilling temperature (in / out)	[°C]	18 / 13	18 / 13
	temperature range (from-to)	[°C]	8-18	8-18
	heat transfer fluid[5)]		Water	Water
	flow rate	[l/h]	900	1500
	operating pressure	[bar]	10 (max)	10 (max)
	pressure drop in circuit	[Pa]	200	450
parasitic demand	electrical power	[kW]	0.03	0.03
	water consumption	[/s]	-	-
Dimensions & weight	Length / width / height	mm	1685 / 1211 / 807	1940 / 1211 / 807
	Weight in operation	Kg	835	1078

Data of free cooling on and off switching points

number of starts	RT1	OT1	T16	T6	date	time
1(on)	25.21°C	21.29°C	27.17°C	66.40°C	01.08.2009	02:40
(off)	24.61°C	21.79°C	17.03°C	65.81°C	01.08.2009	08:22
2	26.20°C	21.09°C	20.71°C	66.50°C	03.08.2009	19:38
	23.71°C	21.89°C	16.73°C	62.53°C	05.08.2009	08:28
3	26.50°C	21.59°C	21.31°C	70.36°C	05.08.2009	22:40
	24.41°C	22.09°C	17.53°C	68.68°C	06.08.2009	08:57
4	25.41°C	21.29°C	21.01°C	66.40°C	06.08.2009	23:36
	24.11°C	22.19°C	16.63°C	65.41°C	07.08.2009	08:36
5	25.41°C	21.19°C	21.21°C	65.91°C	08.08.2009	02:14
	24.71°C	22.19°C	16.83°C	65.21°C	08.08.2009	08:42
6	25.21°C	21.49°C	26.68°C	66.40°C	09.08.2009	04:43
	25.01°C	22.09°C	18.13°C	66.30°C	09.08.2009	07:19
7	25.51°C	21.29°C	22.10°C	65.01°C	09.08.2009	22:38
	24.61°C	22.09°C	16.24°C	64.02°C	10.08.2009	08:34
8	25.71°C	21.19°C	21.01°C	67.79°C	11.08.2009	23:06
	24.71°C	22.19°C	16.24°C	66.60°C	12.08.2009	09:55
9	25.71°C	21.69°C	21.11°C	66.10°C	13.08.2009	01:55
	24.61°C	22.19°C	17.23°C	65.21°C	13.08.2009	10:01
10	26.11°C	21.49°C	20.91°C	66.30°C	13.08.2009	22:20
	24.61°C	22.29°C	17.33°C	65.01°C	14.08.2009	10:49
11	25.91°C	21.29°C	20.91°C	68.68°C	14.08.2009	21:50
	24.41°C	22.29°C	17.03°C	73.03°C	15.08.2009	09:51
12	25.21°C	20.49°C	27.27°C	66.40°C	16.08.2009	04:00
	24.61°C	22.09°C	16.73°C	66.10°C	16.08.2009	07:58
13	25.21°C	20.99°C	29.06°C	66.60°C	17.08.2009	06:01
	25.21°C	22.09°C	18.42°C	66.50°C	17.08.2009	07:40
14	25.01°C	21.49°C	21.11°C	65.91°C	18.08.2009	02:01
	23.81°C	22.09°C	17.03°C	65.41°C	18.08.2009	08:34
15	25.61°C	21.29°C	21.51°C	65.91°C	19.08.2009	04:18
	24.81°C	22.29°C	17.53°C	65.71°C	19.08.2009	07:30
16	26.11°C	20.99°C	22.70°C	72.44°C	21.08.2009	05:33
	25.61°C	22.09°C	18.72°C	72.24°C	21.08.2009	07:29
17	25.31°C	21.19°C	20.51°C	66.60°C	21.08.2009	21:13
	25.21°C	22.09°C	18.42°C	66.40°C	22.08.2009	00:14
18	25.21°C	21.39°C	20.81°C	65.91°C	22.08.2009	04:16
	24.71°C	21.79°C	18.52°C	65.41°C	22.08.2009	09:15
19	25.41°C	21.49°C	20.51°C	66.40°C	22.08.2009	18:09
	23.91°C	21.89°C	17.03°C	65.91°C	23.08.2009	09:48
20	24.41°C	21.19°C	20.41°C	70.36°C	23.08.2009	20:51
	22.81°C	21.09°C	16.34°C	75.31°C	24.08.2009	10:17
21	24.21°C	21.39°C	20.81°C	66.10°C	25.08.2009	01:08
	23.01°C	21.99°C	17.43°C	65.31°C	25.08.2009	08:43
22	24.91°C	21.29°C	20.91°C	72.44°C	25.08.2009	23:26
	23.31°C	21.99°C	16.24°C	71.15°C	26.08.2009	08:59
23	24.91°C	21.39°C	21.31°C	65.91°C	27.08.2009	02:31
	23.71°C	21.99°C	16.73°C	65.41°C	27.08.2009	08:38
24	25.51°C	21.09°C	21.71°C	66.60°C	28.08.2009	02:49
	24.61°C	21.99°C	17.23°C	66.10°C	28.08.2009	07:46
25	25.11°C	21.09°C	21.31°C	66.20°C	29.08.2009	01:31
	25.11°C	21.99°C	17.83°C	65.91°C	29.08.2009	03:15
26	25.21°C	21.29°C	20.51°C	65.71°C	29.08.2009	05:08
	24.71°C	22.09°C	17.83°C	65.41°C	29.08.2009	08:10
27	24.91°C	21.09°C	20.61°C	65.11°C	29.08.2009	11:22
	23.81°C	22.19°C	16.14°C	77.38°C	30.08.2009	11:33
28	25.71°C	21.29°C	20.41°C	68.98°C	30.08.2009	20:00
	23.21°C	20.89°C	15.84°C	75.31°C	31.08.2009	10:24
29	23.91°C	20.99°C	18.72°C	66.60°C	31.08.2009	20:53
	23.31°C	19.10°C	15.74°C	66.40°C	31.08.2009	23:59