Task 38
Solar Air-Conditioning
and Refrigeration

D-A1:

Market Available Components for Systems for Solar Heating and Cooling with a Cooling Capacity < 20 kW

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

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1 Introduction

Systems for solar heating and cooling with a cooling capacity below 20 kW should be as much pre-engineered as possible. That means that the entire system layout and the size of all components is pre-defined by the manufacturer or seller of the system. To install such a system there is no detailed planning process necessary. It can be bought “off the shelf” designed for a given heating and cooling load and can be installed by an HVAC installer. Unfortunately, only few companies offer this kind of pre-engineered system up to now.

To help companies in putting together such a system, this report aims at showing components for (small) solar heating and cooling systems that are on the market and can be used to design pre-engineered systems.

The report is divided into 4 subchapters:

1) Solar combisystems

Solar combisystems without cooling function are the basis for each solar heating and cooling system. The report shows the variety of systems on the market and discusses which characteristic makes them more or less suitable for coupling with a thermally driven chiller.

2) Thermally driven chillers

In this chapter, an overview of chillers on the market up to a cooling capacity of 20 kW is given. Only chillers that are commercially available or at least in a pre-commercial stage, are considered. Tables dealing with different chiller technologies show the most important technical data of each chiller.

3) Heat rejection units

As there is a large number of companies offering heat rejection units, there is not a complete list of manufacturers and models included in this report. Instead, there is a description of the most important technologies for heat rejection for small scale systems. In addition, there is a list of heat rejection units that is used in systems included the Task 38 (Subtask A) monitoring program. This list shows some key data of these systems.

4) Cold stores

Currently the technology for cold storage that is used for small systems are simply cold water stores. Because there are a lot of manufacturers on the market that offer this kind of stores, a list of manufacturers and models is not included.

However, ice storage is becoming increasingly interesting also for small-scale systems (for research and demonstration systems). Therefore, a short description of the available technologies is included in this report as well as a list of companies that is engaged in ice storage. Although small units are not available on the market they may be available on request.

2 Solar Combisystems

By Dagmar Jaehnig, AEE INTEC, Austria and Wolfgang Streicher, Graz University of Technology, IWT, Austria

2.1 Scope

This section aims at identifying solar combisystems for domestic hot water preparation and space heating that may be suitable for integration into a solar cooling system. The analysis is mainly based on the generic systems that were elaborated within IEA SHC Task 26 on Solar Combisystems. An overview of these generic systems was published in 2000 (Suter et al., 2000). In total, 21 generic systems were identified within Task 26.

2.2 Generic Solar Combisystems Without Cooling Function from IEA SHC Task 26

The 21 generic solar combisystems identified by IEA SHC Task 26 differ from one another in many ways.

Two systems (#20 and #21) are designed for seasonal heat storage. The excess solar heat in the summer months is stored to be used in the winter months. Therefore, a combination with a solar cooling system does not make much sense and these systems are excluded from this analysis of suitability of solar combisystems for integration of a cooling system.

Fig. 1 shows a classification of the remaining 19 systems according to three categories which will be explained in detail in the following three subchapters. The letters and numbers in the figure refer to the IEA SHC Task 26 lettering scheme of system categories and the IEA SHC Task 26 system numbering respectively.

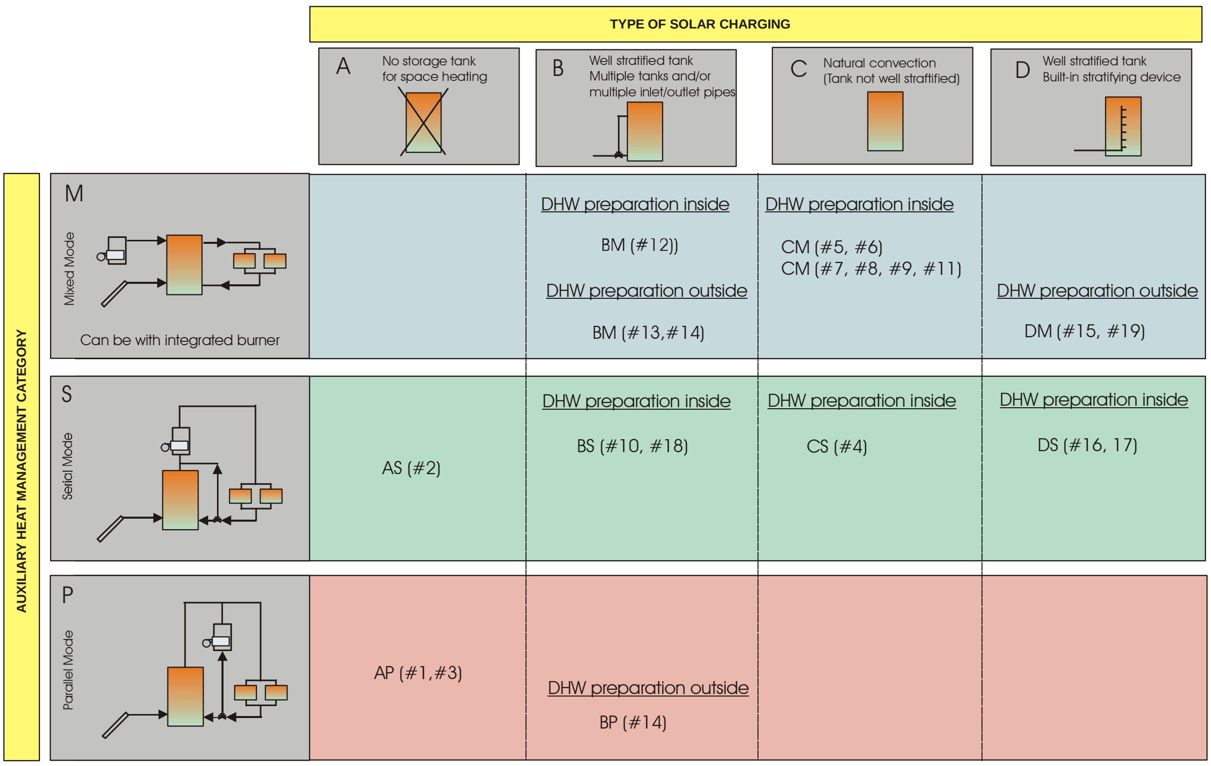


Fig. 1: Generic solar combisystems according to IEA SHC Task 26.

2.2.1 Heat Storage

There are 4 basic categories regarding heat storage.

- Category A: No controlled storage device for space heating. There is no tank for storing energy for space heating. These systems use either the floor slab of the building with an underfloor heating system for storage purposes or there is no possibility at all to store heat for space heating.
- Category B: Heat management and stratification enhancement by means of multiple tanks (“distributed storage”) and or multiple inlet/outlet pipes and/or 3- or 4-way valves to control the flows through the inlet/outlet pipes.
- Category C: Tanks using natural convection only to maintain stratification to a certain extent but without built-in stratification devices (“stratifiers”) for further stratification enhancement.
- Category D: Heat management using natural convection in storage tanks and built-in stratification devices

2.2.2 Auxiliary Heat Management

- Mixed mode (M): The heat storage tank is charged by both solar collectors and the auxiliary heater. Some of these systems have an auxiliary burner integrated into the storage tank; some use an external auxiliary heater.
- Serial mode (S): The space heating loop may be fed by the auxiliary heater, or by both the solar collectors (or a storage unit for solar heat) and the auxiliary heater connected in series in the return line of the space heating loop.
- Parallel mode (P): The space-heating loop may be fed alternatively by the auxiliary heater and by both the solar collectors (or a storage unit for solar heat); or there is no hydraulic connection between the solar-heat distribution and the auxiliary-heat emission.

The mixed mode is by far the most common mode. However, the serial mode is more common in Spain where the mixed mode is prohibited by law.

2.2.3 Location of DHW Preparation Unit

The domestic hot water preparation unit can either be located inside or outside of the heat storage tank.

- Outside the storage tank: This category includes external flat plate heat exchangers that are heated from the heat storage tank. In many cases, the heat exchangers are protected from high temperatures by a tempering valve. This protects the heat exchanger from scaling. Another option is to use a separate tank to store the domestic water. The temperature of this store can be limited to lower temperatures to prevent scaling.
- Domestic hot water preparation units inside the tanks are often used in solar combisystems. These include: Tank-in-tank systems where a small domestic hot water tank is integrated in the top part of the heat storage tank, different types of immersed heat exchangers such as corrugated-tubes that contain a relatively large amount of potable water that is kept at a temperature high enough to be able to deliver hot water to the tap at all times. For these systems scaling may also be a problem if the tank is used for storing heat at very high temperatures for chiller operation in summer.

2.2.4 Advantages and Disadvantages of DHW Preparation Methods

Separate DHW Tank

In Fig. 2 a picture shows a possible situation of a large space heating tank and a small hot water tank. The hot water tank (200-300 ltr) is typically heated by one or two immersed heat exchangers with heat either from the solar thermal collectors, from the space heating tank or from the boiler. Alternatively, when a lot of heating power is necessary, a solution with a flat plate heat exchanger is possible to heat the hot water tank. In principle many hydraulic concepts are possible. The hot water tank can be heated by the conventional boiler, by the solar collectors or both or only indirectly by transferring heat from the main energy store. Based on this flexibility, a proper design of the specific system and the used components has to be done and strongly influences the comfort, performance and efficiency of the system.

Fig. 2: Separate DHW-tank

Advantages

- With low temperature space heating systems (floor heating, wall heating) best “thermal stratification” can be achieved between the two very different required temperature levels (space heating loop: 25-40℃ and hot water demand: 45-60℃ with cold water supply from the mains of 10-5℃).
- Low return temperature to the collector when the collector loop is also directly connected to the hot water tank.
- Scaling problems occur mainly if a flat plate heat exchanger between main store and hot water tank is used. This can be avoided by limiting the flow temperature to the heat exchanger at 60℃.
- High hot water peak power on demand side.
- Maintenance and replacing of the hot water tank is facilitated.
- If the boiler directly heats the auxiliary volume of the domestic hot water tank, the set point temperature for the boiler can be low.

Disadvantages

- Two stores require more space and piping.
- Hot water tank must be resistant against corrosion.

- Very high ratio of surface to volume leads to relatively high heat losses.
- A pump is necessary to transfer energy from the one tank to the other.
- If an immersed heat exchanger only in the top of the hot water tank is used, to transfer heat from the main heat store to the domestic hot water store, the main heat store cannot be cooled to very low temperatures (mains temperature) leading to lower collector efficiencies. A concept with a flat plate heat exchanger between space heating tank and hot water tank performs better, but is much more expensive.
- A circulation loop connected to the hot water tank leads to very bad thermal stratification in the space heating tank due to the high circulation return temperature.
- Parasitic energy (electricity for the pump) is required for hot water preparation.
- Relatively large volume on a temperature level that favors legionella growth.
- If the boiler is only heating the auxiliary volume of the space heating tank, the boiler set point temperature must be relatively high.

Tank in Tank

The tank-in-tank concept was developed in order to reduce space requirements. The DHW tank (100-200 ltr) is integrated in the heat store as shown in Fig. 3.

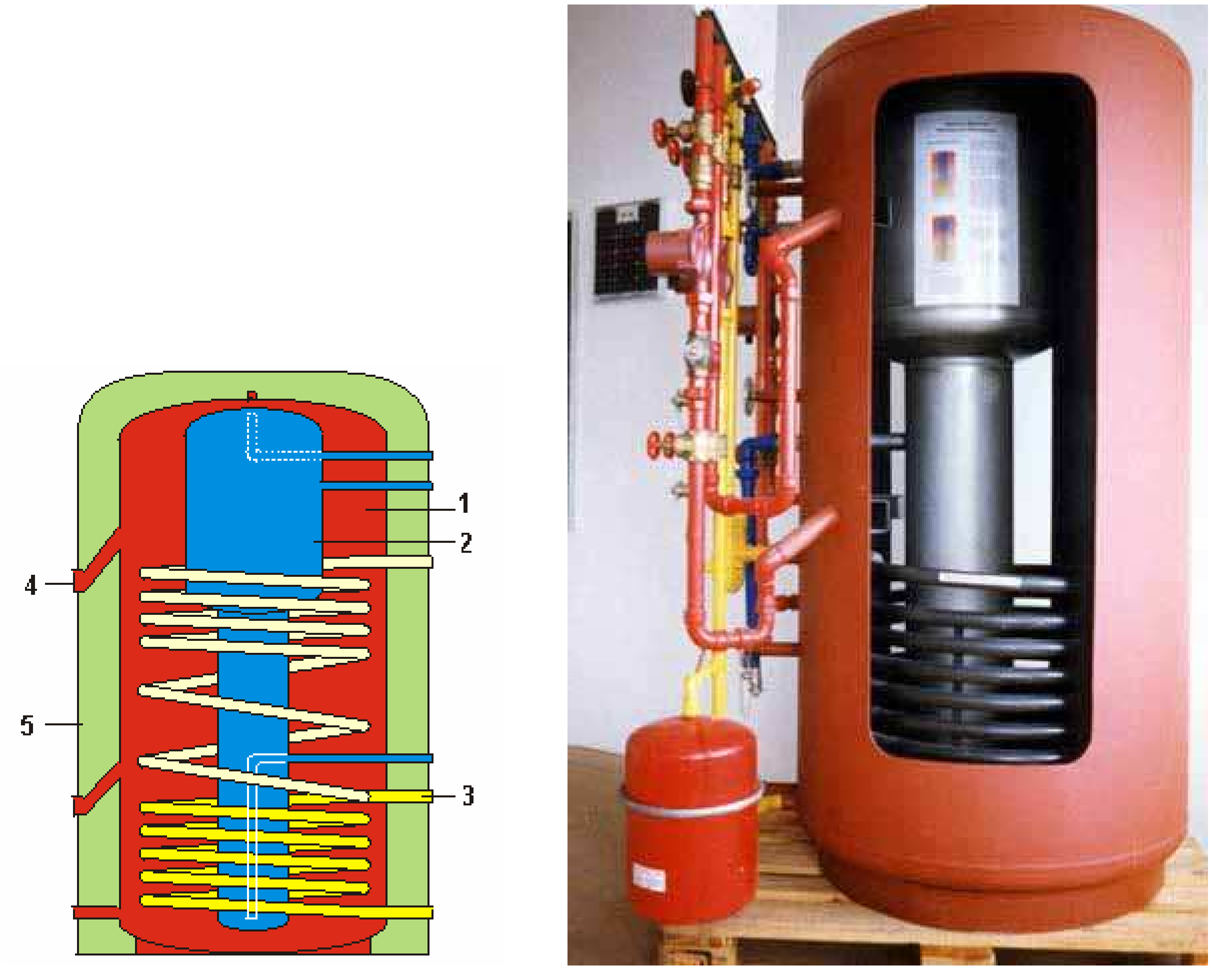


Fig. 3: Tank-in-tank system (Jenni, CH)

Advantages

- High peak power on demand side.
- Little effect of the circulation circuit if the circulation return line is connected at the height that corresponds to the temperature level of the circulation return line
- Scaling problems are rare. Because the diameters of the integrated tank are large compared to a heat exchanger, scaling usually has little effect. Lime can accumulate at the bottom end of the DHW tank, if the temperature of the heat store is above 60℃.
- No parasitic (additional electric) energy for hot water preparation.
- Little space requirement.
- Much smaller hot water volume compared to the hot water tank which reduces the legionella problem in the tank.
- Little heat losses due to the compact design.
- The set point temperature for the boiler can be relatively low.

Disadvantages

- Depending on details of how the hot water tank is designed, the temperatures in the tank cannot be reduced close to the mains temperature. This depends mainly on how far down the integrated tank reaches. If the hot water tank is only integrated in the top part of the store, the bottom of the tank cannot be cooled down properly.
- Some volume in the lower part is on a temperature level that favors legionella growth.
- Maintenance and replacement is almost impossible, but depends on the tank design.
- The top part of the space heating tank must be heated to hot water set temperature all the times.

Internal Heat Exchanger for DHW-preparation

The tank-in-tank concept was further developed to a DHW preparation system with immersed heat exchanger (DHW volume: 30-70 ltr) shown in Fig. 4 to reduce the legionella risk and the cost. Different systems are on the market, mainly differing in how the parts of the heat exchanger are situated at different heights of the heat store.

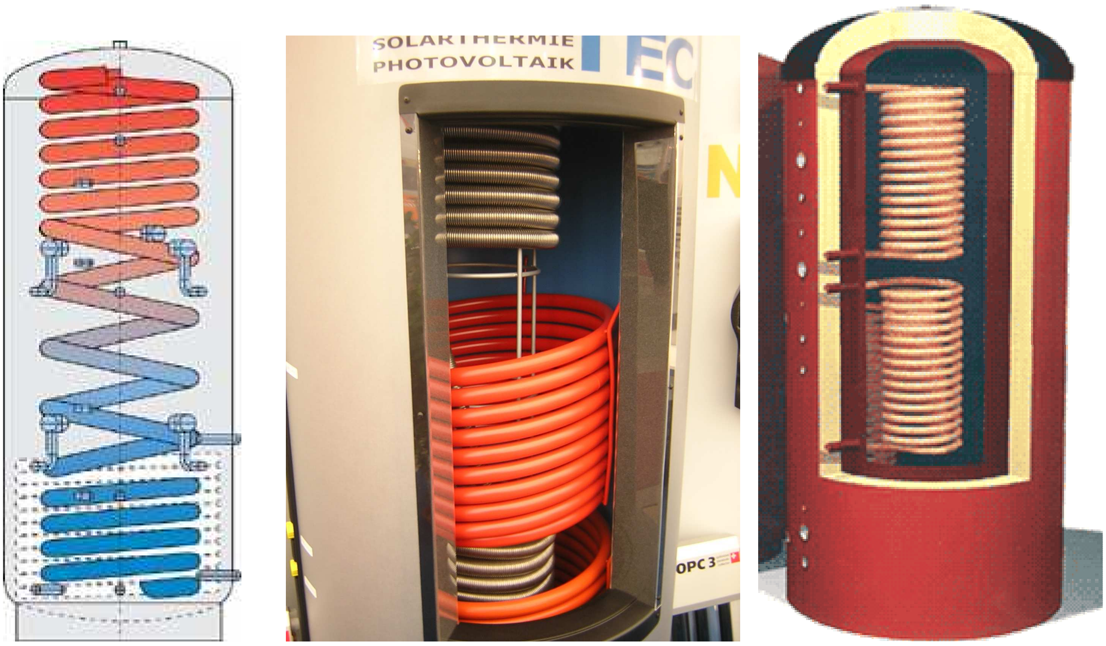


Fig. 4: Internal heat exchanger for DHW-preparation (left and middle: Feuron, CH / right: Solentek, S)

Advantages

- No parasitic (additional electric) energy for hot water preparation.
- Low legionella risk because only a very little volume is kept warm.
- Little space requirement.
- Little effect of the circulation circuit if the circulation return line is connected at the height that corresponds to the temperature level of the circulation return line
- Little heat losses due to the compact design.

Disadvantages

- Low peak power, therefore depending on comfort requirements, the auxiliary set temperature must be significantly higher than the domestic hot water set temperature.
- Depending on details of how the heat exchanger is designed and where in the store it is located, the minimum temperature in the store that can be reached due to the incoming mains temperature can be relatively high.
- Scaling problems possible if temperature in heat store exceeds 60℃.
- Maintenance and replacement is almost impossible, depending on the design.

Flat Plate Heat Exchanger Unit

The last step of the development to reduce the volume of DHW (1-2ltr) which is kept warm is to prepare DHW in the continuous-flow principle by an external flat plate heat exchanger.

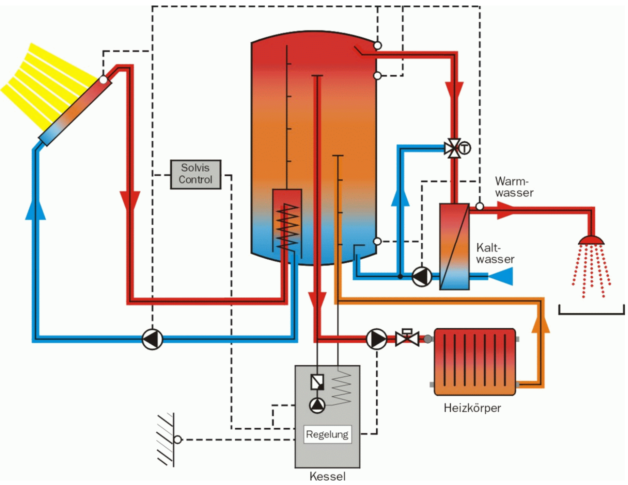


Fig. 5: External flat plate heat exchanger unit for DHW (left: SOLVIS, GER/right: Sonnenkraft, AUT)

Advantages

- Peak power is limited, but the power can easily be adapted using a correctly designed heat exchanger, depending on the requirements.
- Low return temperature to the heat store.
- Best usage of the stored energy in the tank.
- No scaling problems if flow temperature is limited.

- Lowest legionella risk because almost no volume is kept warm.
- Little space requirement.
- Maintenance and replacement is easily possible.

Disadvantages

- Circulation loop has negative influence on the thermal stratification in the heat store due to high return temperature if no stratification device for the return flow is used in the tank.
- Advanced control system is needed for good behavior of this concept.
- A pump is necessary to run the hot water preparation unit.
- Parasitic energy (electricity for the pump) is required.
- Depending on system design, increased heat losses due to the external installation of the hot water heat exchanger.
- The top part of the space heating tank must be heated to a temperature higher than hot water set temperature all the times to enable enough peak power during hot water preparation, if the boiler peak power is not high enough.

2.3 Suitability for Integration of a Solar Cooling System

Connecting a thermally driven cooling machine to such a combisystem means to add a heat sink in summer that needs a low temperature difference (5-10 K) at high absolute temperature level (60-90℃). This temperature leve l is well above the temperature needed for DHW preparation. Space heating normally does not occur at the same time as cooling is needed.

In a first step, some of the solar combisystems defined in IEA SHC Task 26 were excluded because their system concept is not suitable for integration of a solar cooling system for the following reasons:

- Three systems do not include a storage tank for storing heat for space heating. The heat is stored in the concrete slab of the building itself. Although it may be possible to operate a solar cooling system without a hot storage tank, by delivering the generated heat from the solar thermal collectors directly to the chiller, the heating and cooling systems would in this case not be truly integrated but rather two separate systems that use the same collector field. Therefore these systems are excluded from this report**. (Systems #1, #2, #3)**

- Three systems are very small (small storage tank and small collector area). Such systems are typically used e.g. in the Netherlands and Denmark. These systems are designed for relatively small solar fractions and the small tank sizes are not suitable for use as hot storage tank for a solar cooling system. **(Systems #4, #5, #6)**

After excluding these systems, there are 13 systems remaining: #7, #8, #9, #10, #11, #12, #13, #14, #15, #16, #17, #18, #19 (see Fig. 6).

To analyze which of these systems are more suitable to integrate solar cooling, the main aspect is how the domestic hot water preparation is done. The key problem here is to avoid excessive scaling if the tank is regularly heated above the 80℃ required to power a thermally driven chiller.

Four of these systems use an external domestic hot water preparation. Two use separate tanks and two others use an external flat plate heat exchanger. In both cases, the flow temperature to the store/HX can easily be limited to 60℃ which avoids scaling.

For the systems with integrated domestic hot water preparation the situation is more difficult. In the case of tank-in-tank systems, the diameters inside the integrated tank are much larger than in integrated heat exchangers. Therefore, scaling may not cause a great problem. Integrated heat exchangers on the other hand can be blocked completely by lime formation if the temperatures in the store are high and hard potable water is used. Some manufacturers recommend limiting the temperature in the store to e.g. 60℃ which avoids scaling. In that case, the system cannot be used for solar cooling application anymore because temperatures well above 60℃ are needed for the coo ling machine.

Therefore, a big question mark has to be put behind systems with integrated heat exchangers for domestic hot water preparation and a small question mark for tank-in-tank systems. For both system types experiences with installed systems will show whether they are suitable for integration of solar cooling or not.

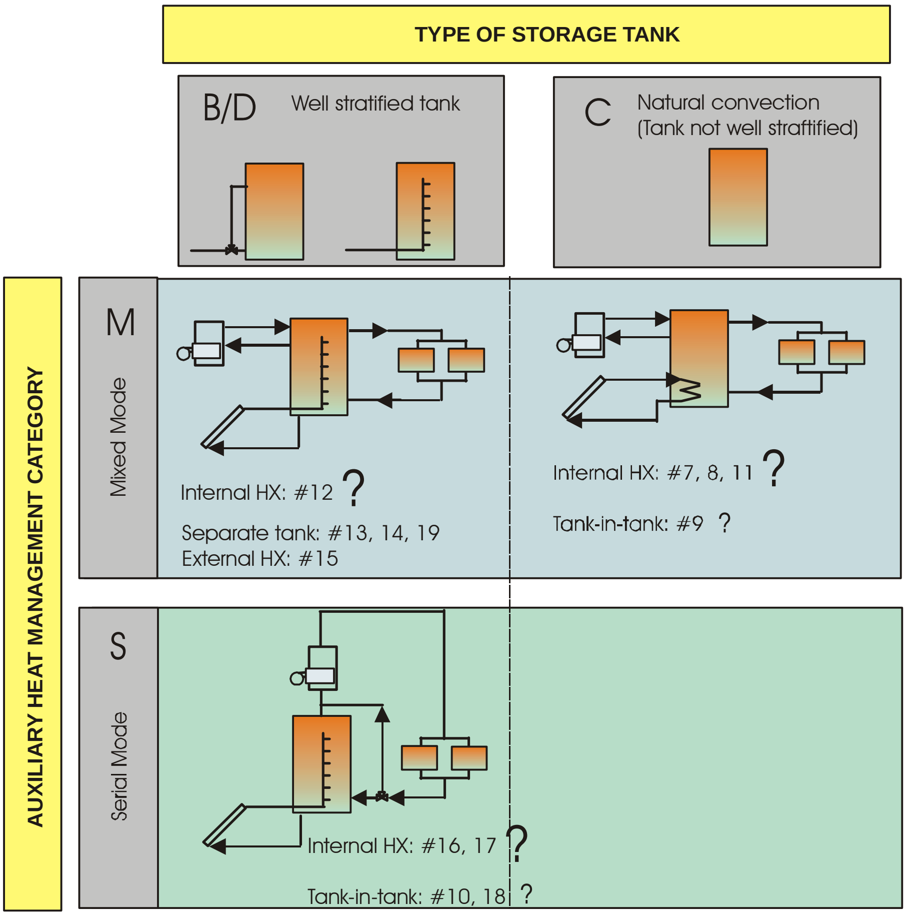


Fig. 6: Remaining 13 solar combisystems (from IEA SHC Task 26) after excluding all those that are not suitable for integration of solar cooling

2.4 Other Features Relevant for Solar Cooling Integration

After having described different solar combisystems without cooling function we now need to look at topics that are important for solar cooling into applications but are not relevant for the solar combisystem for heating only.

2.4.1 Possibility to Reduce Tank Size for Summer Operation

If the store size of the solar combisystem is relatively large, it may be necessary to reduce the tank size in summer to ensure that cooling operation can start relatively early in the day without having to heat the entire storage tank to high temperatures first. This can be done by including an extra outlet at mid height of the storage tank that is used in summer. Of course this outlet can also be used in winter to reach temperatures for space heating operation faster. This second outlet can be an additional integrated heat exchanger or an additional outlet to the external heat exchanger with a mixing valve. This feature is already included in system # 10, 12 and 18 but could also be included in other systems if necessary.

2.4.2 Stratification in the Tank

According to the high temperature level, low temperature lift and high mass flow needed to drive solar cooling machines, principally a new volume above the DHW preparation could be placed in the store. This would lead to a strong increase of the storage volume. Therefore in summer normally the whole storage volume is used for solar cooling and DHW preparation needs to be mixed down to lower temperatures. The auxiliary heat remains on the upper part of the store, as the power of the heater should be sufficient for driving the cooling machine and the DHW production. Additionally DHW production is for e.g. office buildings the smaller part compared to space heating, so the main focus of the hydraulics for summer operation can be laid on the cooling part. This is maybe different in hotels, where the DHW demand can be significantly higher. Here a solar preheating zone for DHW in the bottom of the tank may be used. In this case, stratifiers for the solar inlet may be used to allow preheating of the water at low temperatures and automatically switching back to the high temperature level for solar cooling. In spring and autumn, where DHW production, space heating and cooling may occur in the same time periods, the optimization of the volumes has to be done by simulation.

2.4.3 High-flow / Low-flow

In most cases a collector field connects several single collectors. These collectors can be either connected in series or in parallel; mainly combinations of the two forms of connection are used. To assure turbulent flow and therefore a high heat transfer rate in each collector tube, the volume flow through each collector should be kept above a certain level. On the other hand the volume flow should not be too high in order to avoid unnecessary pressure drop and therefore high electricity demand for the circulation pumps.

By connecting the collectors in series, the total mass-flow through the whole collector field is usually reduced (low-flow) and the temperature rise in the collector is increased. The advantage of this is that hot water can be supplied quickly. The disadvantage is a higher thermal loss of the absorber to the environment, which is due to the larger temperature difference. The pump electricity demand decreases due to the lower level of total mass flow in the collector loop but increases due to the higher pressure drop in the collectors. The higher pressure loss of collectors connected in series can be overcome partly by sizing the collector loop pipes bigger and therefore lower the pressure losses in these tubes. When connecting in series, there is a more regular flow through the collector area due to a higher driving pressure drop over the field. The hydraulic layout has to be adjusted to the total mass flow. Low-Flow systems normally use stratification units in the tank in order to prevent mixing the hot temperature from the collector fluid with colder temperature at the inlet of the store.

High-flow installations with collector areas below 15 m² are most often combined with internal heat exchangers in the storage tank. For bigger fields external heat exchangers with fixed inlets are used. There can be two or more internal heat exchangers or pairs of inlets located at different heights in the storage tank to allow stratification even with high-flow systems.

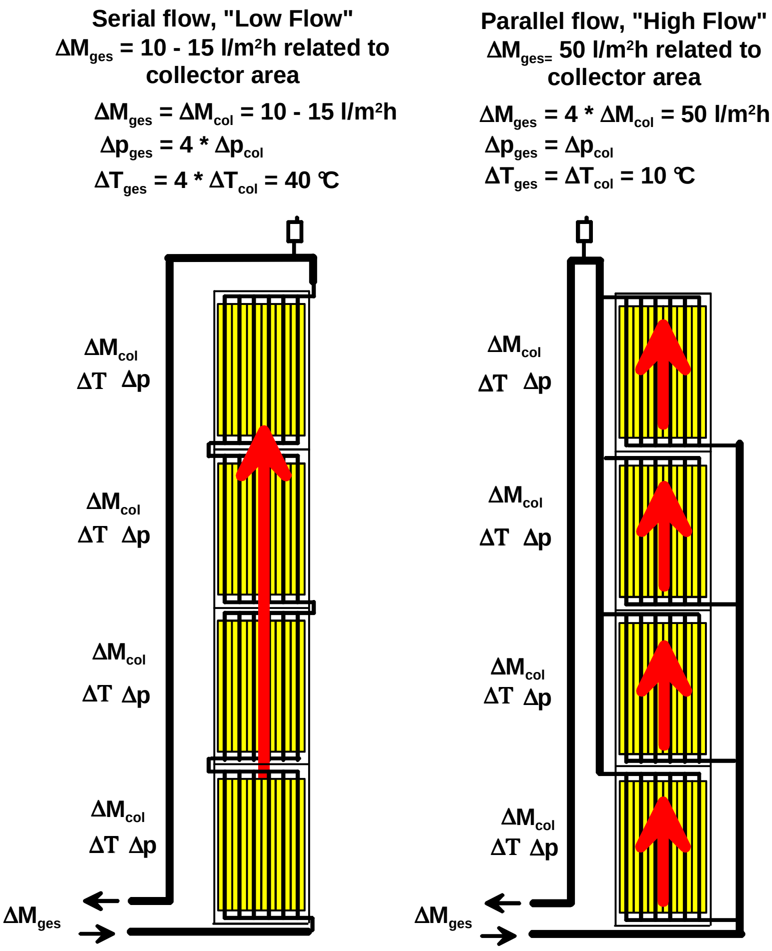


Fig. 7: Hydraulics for the collector field for low flow serial (left) and high flow parallel (right) systems. In all cases care has to be taken to maintain an equal resistance in all parallel lines. This can be partly achieved by a “Tichelmann” arrangement where the added length of the inlet and outlet pipes is the same for all parallel collectors (Streicher 2008).

Many existing solar combisystems (without cooling function) work with the low-flow principle in the collector loop. The advantage is that the necessary temperature lift for domestic hot water preparation and space heating can be reached in one step in the solar collector if the solar radiation is high enough. Therefore, the auxiliary heater has to be operated less frequently.

When connecting such a system to a thermally driven chiller, this can be a problem because the temperature difference between flow and return of the hot side of the chiller is typically small (~ 6 K). Therefore it would make more sense to operate the collector loop (at least for the cooling mode) at this small temperature difference rather than the roughly 30-40 K that are typical for low-flow solar combisystems due to the domestic hot water (DHW) demand. During the summer high flow operation would be favourable for solar cooling but low flow operation would be favourable for DHW production.

One option to overcome this dilemma is to switch the collector loop at least between low-flow operation in winter to high-flow operation in summer. However, care has to be taken to ensure proper turbulent flow and acceptable pressure drops in the collector circuit as well as on the balanced flow through all parallel collector tubes.

There are in principle four options:

- The collector hydraulics stay the same in summer and winter but the mass flow through the collector varies by switching the pump speed or the type of pump. In this case there is either a high pressure drop in the collector loop during high flow operation, that results in a high electricity demand of the collector pump or the collector loop is designed for high flow and during low flow operation the flow regime in the collector is laminar which reduces the collector efficiency by a few percentage points and may additionally result in non-uniform flow in the parallel collector tubes. As high flow has about four times the volume flow of low flow, the pressure drop is about 16 times (4^2) higher in high flow than for low flow (according to Bernoulli's law with dp ~ $(\rho v^2)/2$. The electricity demand is even 64 times higher ($P_{el} = (dV * dP)/ \eta \sim v^3$. Here also the question arises, that maybe two different pumps for high flow and low flow have to be chosen (high pump head and high volume flow and low pump head at low volume flow).

- The other possibility is to change the hydraulics of the solar collector. This can be done by installing switching valves that connect more collectors in series for low-flow operation. Again probably two pumps are needed (one for high volume flow and low delta T and one for low volume flow and high pump head).
- A third possibility would be to drive the collector field only in high flow and adjust the hydraulic system for space heating accordingly (e.g. no stratifiers in the tank etc.), which gives less useful solar gain in winter but makes the plant cheaper.
- A fourth possibility would be to drive the collector between high and low flow. This assures equal flow in the collectors but gives slightly less solar yield during the heating and cooling season.

Concluding, it becomes obvious, that there is room for plenty of optimization for solar thermal driven cooling systems coupled to solar combisystems.

3 Thermally Driven Chillers

By Tomas Núňez, Fraunhofer ISE, Germany

A survey of small capacity thermally driven chillers was carried out in the frame of the PolySMART project. The intention of the survey was to assess the present developments in the field taking into account laboratory developments, prototypes, small series products and commercially available machines. The survey was limited to the capacity range of about 70 kW cooling power. A total of 34 developments have been registered. The majority of the surveyed machines were prototypes that have been modified and changed in the last three years. Some developments were discontinued or are no longer available. In the tables below an extraction of the survey is presented. The criteria for this extraction are:

- The machine should be available as commercial product, as a small series product or a small series product is to be expected in short time
- The chilling power is up to 30 kW
- Performance data is available.

The machines are classified according to the technology: adsorption systems, including the pairs water / silica gel and water / zeolite, absorption systems with the working pairs ammonia / water and water / lithium-bromide and other technologies which include jet cycles and a three-phase Water / Lithium-chloride absorption chiller. This last chiller is also a thermo-chemical store for heat and cold and therefore a different technology.

3.1 Adsorption Systems

Table 1: Adsorption chillers below 20 kW chilling capacity

Manufacturer			**SorTech AG**	**SorTech AG**	**ECN**	**InvenSor GmbH**	**InvenSor GmbH**
country			Germany	Germany	Netherlands	Germany	Germany
Model name / number			ACS08	ACS15	SoCOOL	HTC 10	LTC 07
Contact data of supplier			SorTech AG Weinbergweg 23 D - 06120 Halle (Saale) Fon: +49 (0)345 279 809-0		ECN POBOX1 NL-1755 ZG Petten the Netherlands tel +31-224 564871	InvenSor GmbH Gustav-Meyer-Allee 25 D-13355 Berlin Email: info@invensor.de Telefon: +49 (0)30 - 46 307 - 396	
internet			www.sortech.de	www.sortech.de	www.ECN.nl	www.invensor.de	www.invensor.de
other supplier / model name			SolarNext / chillii® STC8	SolarNext / chillii® STC15		SolarNext / chillii® ISC7	SolarNext / chillii® ISC10
Basic information		**unit**					
	technology		water-silica gel	water-silica gel	water-silica gel	water zeolite	water zeolite
	nominal chilling power	[kW]	8	15	2.5	10	7
	nominal COP		0.6	0.6	0.5	0.5	0.54
	heat rejection type		external	external	external	external	external
	intended area of application		air-conditioning	air-conditioning	residential cooling	air-conditioning	air-conditioning
	development stage [4)]		pre-commercial	pre-commercial	ID	pre-commercial	pre-commercial
	specification sheets available (yes/no)		yes	yes	no	yes	yes
	(expected) investment costs	[€]	n.a.	n.a.	1500-2500 (expected)	n.a.	n.a.
Nominal operation conditions							
Driving circuit	power	[kW]	13.4	25	5.5	20	13
	operating temperature (in/out)	[℃]	72 / 65	72 / 65	85 / 79	85 / 77	65 / 59.5
	temperature range (from-to)	[℃]	55..95	55..95	60-95	65-95	55-85

	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	1600	3200	600	2200	2200
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	230	260	500 (indicative)	230	230
Heat rejection circuit	power	[kW]	21.4	40	8	30	20
	Temperature (in/out)	[℃]	27 / 32	27 / 32	30 / 40	27 / 33	27 / 31
	temperature range (from-to)	[℃]	22..37	22..37	15-40	27-41	22-37
	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	3700	7000	600	4500	4500
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	350	440	500 (indicative)	500	500
chilling circuit	nominal chilling power	[kW]	8	15	2.5	10	7
	chilling temperature (in/out)	[℃]	18 / 15	18 / 15	16 / 12	18 / 15	18 / 15
	temperature range (from-to)	[℃]	6..20	6..20	5--20	8-18	15-18
	heat transfer fluid[5)]		water	water	water	water	water
	flow rate	[l/h]	2000	4000	500	2900	2000
	operating pressure	[bar]	4	4	1--2	4	4
	pressure drop	[mbar]	300	500	300	240	130
parasitic demand	electrical power	[kW]	0.007	0.014	<0.3	0.02	0.02
	water consumption	[l/s]	none	none	none	none	none
Dimensions & weight	Length /width/height	mm	790 / 1060 / 940	790 / 1340 / 1390	n.a.	1300 /650 / 1650	1300 / 650 / 1650
	Weight (empty / operation)	Kg	265 / 295	530 / 590	n.a.	- / 370	- / 370

3.2 Absorption Systems

3.2.1 Ammonia – Water Systems

Table 2: Ammonia - water absorption systems below 20 kW chilling capacity

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	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)	[℃]	96 / 86	75 / 68	240
	temperature range (from-to)	[℃]	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)	[℃]	35 (air temperature)	24 / 29	35 (air temp erature)
	temperature range (from-to)	[℃]	30-42	24	-12 ... 4 5
	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)	[℃]	14 / 9	18 / 15	0 / -5
	temperature range (from-to)	[℃]	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

3.2.2 Lithium – Bromide / Water Systems.

Table 3: Lithium-bromide / water absorption systems below 20 kW chilling capacity

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	
other supplier / model name			SolarNext / chillii® ESC15	SolarNext / chillii® WFC18		
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		Not available	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	8 0 - 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 fluid[5)]		water	water	Water	water	water
	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

3.3 Other Technologies

Table 4: Other technology chillers below 20 kW chilling capacity

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 / 4 0
	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 / 1 3
	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

4 Heat Rejection

There is a number of different heat rejection units also for small thermal capacities available on the market. Therefore, not all units on the market will be presented in this report. However, this chapter should give an overview of the available technologies.

In the second part of this chapter, some key data of the heat rejection units that are used in systems being monitored within the framework of Task 38. The choice of heat rejection solution is often critical to the electrical power consumption of the thermally driven chiller. As it can be seen in chapter 3, most of the chillers are water cooled which means that the cooling water supplied to chiller has to be connected to some type of equipment rejecting the heat to the environment.

The third part of this chapter gives some general considerations on the electricity consumption of heat rejection units and shows ways how to reduce it.

4.1 Heat Rejection Technologies

By Harald Moser, Graz University of Technolgy, IWT, Erich Podesser, Podesser Consulting, Tomas Núñez, Fraunhofer ISE, Daniel Mugnier, TECSOL and Lars Reinholdt, DTI

Generally speaking, different heat sinks are possible to reject the heat, e.g. air, ground or water. While the use of ground and water depends strongly on the local conditions, air is available for almost all applications.

For rejection of heat to the ambient air, in principle two types of systems are available: open cooling towers (or wet cooling towers) and closed cooling towers or (dry coolers). As a combination of these, adiabatic pre-cooling of the air in the dry cooler and hybrid cooling towers should be mentioned.

The main difference between these technologies is that in the dry cooler the cooling water rejects the heat to the air via a heat exchanger and in wet cooling towers the cooling water is sprayed into the air and direct heat and mass transfer takes place. Thus in dry coolers only sensible heat and in wet cooling towers mainly latent heat is exchanged.

A further option is to use ground coupled systems like vertical boreholes or horizontal ground coupled heat exchangers for heat rejection. These systems are well known as low temperature heat sources for ground coupled heat pumps and as heat sink for non-mechanical cooling systems. The performance strongly depends on the ground characteristics and an accurate dimensioning.

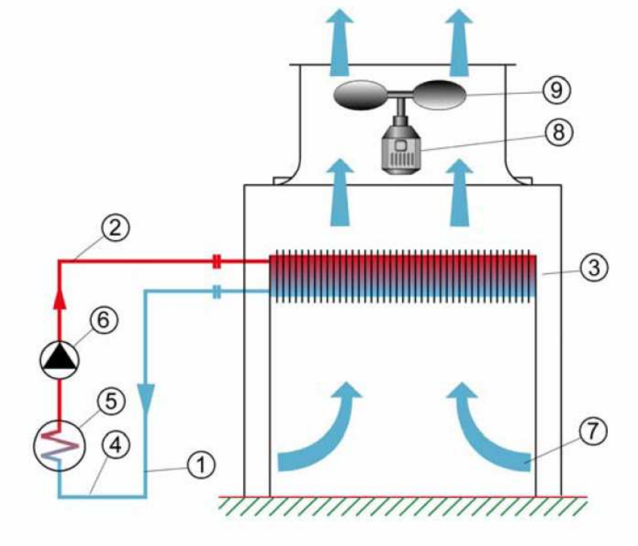
4.1.1 Dry Cooler

Dry coolers consist generally of finned heat exchangers (air to water), fans and a casing. The water circulates in a closed circuit and by passing ambient air over the finned surfaces the heat is rejected to the air (compare Fig. 8).

With air-cooled heat exchangers, it is not possible to cool the medium to below the ambient dry bulb temperature. In this case the approach temperature between the medium outlet temperature and the inlet temperature of the dry air depends mainly on the size and capacity of the dry cooler - typical values of approach temperatures are 5 to 9 K (SWKI, 2003).

Dry coolers are often used for cooling refrigerants, oils or water/glycol mixtures. Compared to wet cooling towers they have lower operational and maintenance cost and because the cooling water does not come in direct contact to the air they have no hygienic problems or legionnella risks. Further advantages are little noise, easy installation and a low profile.

The main disadvantages compared to wet cooling towers are higher heat rejection temperatures, much higher investment costs, parasitic energy consumption for the fan and space requirement.



1. cooling circuit
2. inlet flow
3. cooling element (heat exchanger)
4. return flow
5. heat source
6. circulating pump
7. cooling air
8. fan drive
9. fan

Fig. 8: Sketch of a dry cooler (SWKI, 2005)

4.1.2 Wet Cooling Towers

Cooling towers are characterized by the primary cooling being evaporation of water. The lowest achievable temperature is the wet bulb temperature for the ambient air. The wet bulb temperature is depending on both the dry bulb temperature and the moisture content of the air. At rising dry bulb temperature and constant moisture content the wet bulb temperature will rise. As a consequence the capacity of a cooling tower will go down as the ambient temperature raise doing the day.

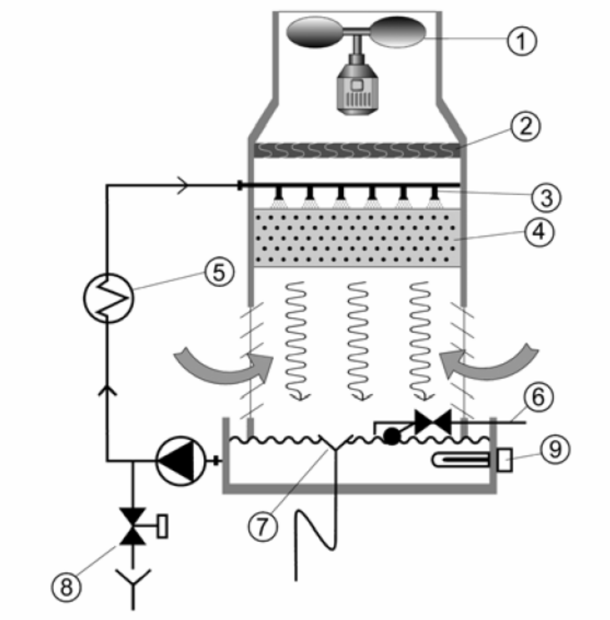
Cooling towers can be

- open type, having direct contact between cooling water and the air stream in the tower
- closed type, not having direct contact between cooling water and the air stream in the tower

The open wet cooling tower (open loop evaporative cooling tower) consists of a shell containing packing/fill material with a large surface area. Nozzles arranged above the packing, spray and distribute the cooling water onto the packing. The water trickles through the packing into a basin from which it is pumped back to the chiller. The water is cooled by air and drawn or blown through the packing by means of a fan. The air flow, which is either in counter or cross flow to the water flow, causes some of the water to evaporate, thus latent heat, is exchanged from the water to the air.

The evaporated water is continuously replenished by make-up water. However, evaporation also increases the concentration of the dissolved solids in the cooling water and blow down of the cooling water is therefore necessary. In wet cooling towers the wet-bulb temperature determines the degree of cooling and thus cooling below the ambient dry bulb temperature is possible. The characteristic approach temperature, which is the difference between the water outlet temperature and the ambient wet-bulb temperature, of open wet cooling towers lies between 4 to 8 K (SWKI 2005).

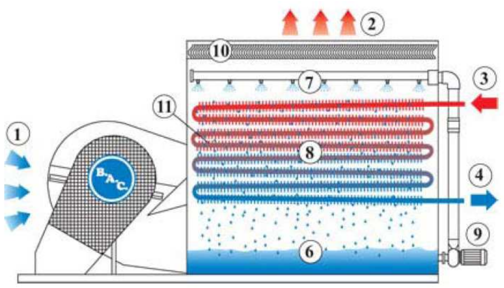
Compared to dry coolers wet cooling towers are able to cool the cooling water to a lower temperature level, require less space and have much lower investment costs. The main disadvantages of wet cooling towers are hygienic problems, water consumption and high maintenance effort.



1. fan with drive
2. drift eliminator
3. spray nozzles
4. trickle packing
5. heat source
6. float valve and fresh water inlet
7. overflow
8. bleed off
9. frost protection heating

Fig. 9: Sketch of an open wet cooling tower (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Air-cooled Cooling Systems.pdf, 26.03.2009)

The open loop wet cooling tower has the risk of fouling the heat transfer surfaces as dirt and dust from the air incl. biological material will be rinsed out of the cooling air stream. This is eliminated in closed cycle wet cooling towers as the cooling water is cooled in pipes over which water is distributed and will evaporate. Compared to open type wet cooling towers the approach temperature is higher but still lower than the dry coolers. Due to the more complex design the investment cost is higher, but the running cost is lower. An example of a closed cycle cooling tower can be seen in Fig. 10.

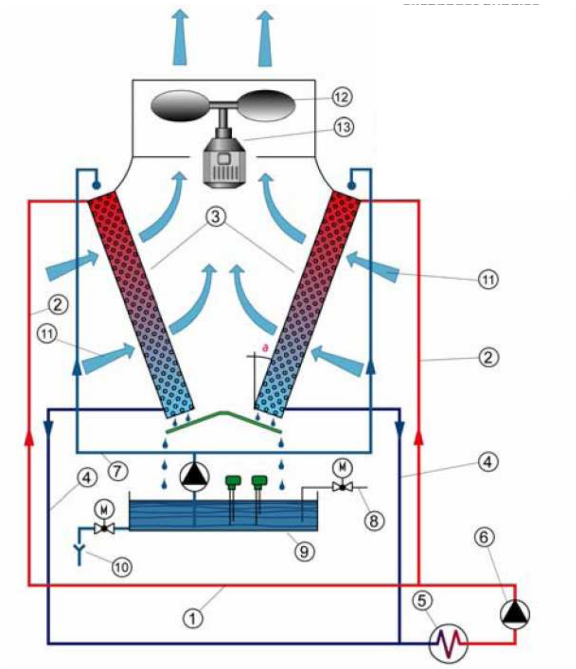


1. Air In;
2. Air Out;
3. Fluid In;
4. Fluid Out;
6. Cold Water Basin;
7. Water Distribution System;
8. Coil;
9. Spray Water Pump;
10. Eliminators;
11. Optional Extended Surface.

Fig. 10: Sketch of a closed cooling tower (BAC type VFL, www.baltimoreaircoil.be)

4.1.3 Hybrid

The hybrid dry cooler combines the two methods of dry cooling and evaporative cooling. The cooling water is circulated by a pump in a closed primary cooling circuit from the heat source to cross current air to water heat exchanger.



1. primary cooling circuit
2. inlet flow
3. finned tube heat exchanger
4. return flow
5. heat source
6. circulating pump
7. deluging water circuit
8. make up water inlet
9. water collector
10. bleed off
11. cooling air
12. fan
13. fan drive

Fig. 11: Sketch of a hybrid dry cooling system (Jaeggi, http://www.guentner.ch/pdfs/Evaluation of Aircooled Cooling Systems.pdf, 26.03.2009)

In cool weather conditions, this process cools down the cooling water sufficiently and the hybrid cooler operates like a dry cooler. At high air temperatures the hybrid cooler uses the principle of evaporative cooling in order to achieve lower cooling temperatures. Therefore, a pump circulates water from a basin to the cooling element where the water flows back via the finned surface of the air to water heat exchanger. The air flowing past the heat exchanger causes the water to evaporate on the fin surface, and takes the heat from the fins.

Comparing a hybrid dry cooler to common dry cooler we see that it has the advantage of using evaporative cooling at hot weather conditions and therefore cools down the cooling water below the dry bulb temperature, it has a higher capacity and lower energy consumption. On the other hand the hybrid dry cooler has higher investment costs, maintenance effort and water consumption. Furthermore, hygienic measures have to be taken as for the wet cooling tower.

4.1.4 Boreholes

Boreholes are vertical ground coupled heat exchangers. They are of special interest in cases of a little available surface area. The vertical boreholes use the relative low and constant temperature in the soil. Temperature fluctuations can be measured only down to a depth of 15 m. Below 15 m depth the temperature is constant with 10 °C over the season. This temperature increases every 30 m depth with 1 °C. F or four kinds of soils the guideline VDI 4650 (VDI 2009) provides the following specific heat transfer values: 30 W/m in dry soil, 55 W/m in schist and similar stone, 80 W/m in solid rock and 100 W/m in a soil with significant ground water flow. This classification of soil shows that the heat transfer depends strongly on the soil condition. The distance between two or more boreholes should be a minimum of 5 m.

Boreholes are carried out as single u-type, double u-type or coaxial tube heat exchangers made of a plastic material (high density polyethylene tubes HDPE). They are installed in vertical holes drilled into the ground. In order to improve the thermal contact to the ground and seal the borehole, a special sealing material (cement or bentonite) with high thermal conductivity is used. Boreholes are normally drilled down to a depth of about 100 m, but this depends on the geology of the ground and the intended use, and in most cases they are used as a low temperature heat source for heat pump applications. The following explanations refer to heat pumps (heat source): In this case, the heat extraction capability is the key factor for the dimensioning of the borehole system: if the system is too small for the intended heat extraction power the temperature of the borehole will decrease in time and the ground may even freeze (for chillers utilizing water as refrigerant the risk of freezing limits the lowest soil temperature to 4-5℃). The main par ameter for the evaluation of the borehole performance is the heat conductivity of the ground, which depends on the geology and ground water flows. This parameter can be experimentally determined through a 'Thermal Response Test'. In this experiment a constant power is injected into a finished borehole and the mean fluid temperature is measured. The measurement has to be carried out over a period of more than 50 hours in order to avoid transient and capacitive effects in the measurement.

An important figure is the specific heat capacity of the borehole. However, this is not the only important characteristic: the number of operation hours and thus the total amount of heat extracted annually is a decisive factor for the long term performance of a borehole system. While the specific heat capacity is a factor that is important during the actual operation of the borehole, the total extracted heat is the factor that determines the long term reliability and thus if the borehole can be considered as a renewable resource. Thus, the real specific power has to be calculated for a long period of time (15 to 30 years) taking into account the hours of full load operation per year. As a result the specific power that can be expected without depletion of the source decreases with the number of full load hours per year.

This is also valid for the use of a borehole system as heat sink: a specific power may be defined but the total amount of heat absorbed by the ground has to be considered. A borehole system can only work as a renewable heat sink if the heat absorbed can be dispersed sufficiently within the ground in order to not significantly affect the undisturbed temperature distribution of the ground. This factor determines the long-term suitability of the system as a heat sink. As a conclusion, monovalent systems (either as heat source or heat sink) can only be operated a limited number of hours a year.

Thus, bivalent systems which use the ground as heat source as well as heat sink are convenient. In this case the heat extracted from the ground and the heat rejected into the ground may be counterbalanced. The dimensioning of such a system with a reversible thermally driven chiller which can also be used as a heat pump is not straight forward. Since the ratio of heat to be rejected in the chilling mode to low temperature heat extracted in the heat pumping mode is typically around 2.25 to 2.5, the dimension of the borehole system can only be calculated with suitable simulation tools which take into account the heat extraction and heat rejection powers and temperatures as well as the expected operation hours in each mode over the whole expected lifetime of the system in order to avoid long-term disturbance of the ground.

It is recommended that a qualified advice and expertise should be obtained before the drilling of a borehole. Information about imposed conditions such as the expected kind of soil and the heat transfer should be obtained. The drilling has to be applied by a concessionary company. The specific cost of a borehole down to 150 m in Austria is in the range of 55 to 60 €/m. More information can be obtained, for example, in German in (Ochsner 2009).

4.1.5 Horizontal Ground Heat Exchangers

The horizontal ground heat exchanger is designed to the use of the cooling storage capacity of the ground. Heat exchanger made of polymer tubes (PER for example) is put at 0.5 to 2 m depth into the ground in order to reject heat to the ground. It is connected to the heat rejection circuit of the thermally driven chiller.

Fig. 12: Example of horizontal ground probe heat rejection system in INES office, Chambery (source: INES RDI)

This heat rejection technology is interesting for the following reasons: no need of any wet or dry cooling tower leading to far less electricity consumption, low heat rejection temperature (less than 30°C) if the geothermal heat exchanger is well designed, ease of implementation of the horizontal probe network for new buildings during the civil works phase, and the possibility to use these heat exchangers during winter in heat pump mode.

The heat transfer in horizontal ground heat exchangers for heat rejection systems depends above all on the kind of soil. In all cases of applications the soil should be natural and not a man made earth deposit. Regarding the guideline VDI 4640 Part 2 (VDI 2001) the following specific heat extraction values can be expected: 10 W/m^2 soil surface for dry solid soil, 20 to 30 W/m^2 in moist solid soil, and up to 40 W/m^2 for a water saturated soil. For the realization of horizontal ground heat exchanger it is recommended to use a 0.75“ or 1.0” PE-tube for a maximum pressure of 10 bar. The PE-tubes of the horizontal ground heat exchanger should be piped in a depth of 0.5 to 2 m with a horizontal distance of 50 cm in moist soil and with about 80 cm in dry soil. For having the desired surface it is normally necessary to pipe parallel loops with not more than a length of 100 m.

Thanks to a horizontal ground heat exchanger, it is possible to use very little electricity consumption to run the pump connecting the chiller to the heat rejection loop (only 25 W/kW$_{\text{heat rejection}}$ for a 4.5 kW chiller). Economically speaking, this solution is more expensive than a traditional wet cooling tower system (more than double cost) due to the significant length of polymer pipes but on a 20 years global cost calculation (avoidance of water treatment and water consumption), this investment is more interesting, especially for new buildings (civil works to burden the pipes more or less free) and for countries where legionella protection legislation consequences make wet cooling tower management expensive.

Table 5 shows example data of an installed horizontal ground heat exchanger

Table 5: Example data of a horizontal ground heat exchanger, source: Bengt Hedestam, ECONICsystems, Gars am Kamp, Austria

heat rejected	5 kW
horizontal ground area	540 m^2
PE-tube length	270 m
tube diameter	32 mm
tube wall thickness	3 mm
piping depth	2 m in a dry rocky ground
heat transfer area	27.14 m^2
specific heat transfer density	185 W/m^2 tube surface
specific heat transfer	18.5 W/m of tube
specific cost	12 to 15 €/ m PE-tube
water temperature of the heat rejection system	22/27 °C in the morning and 28/33 °C in the late afternoon
heat rejection temperature	starts every morning around 22/27°C

4.2 Examples of Heat Rejection Components

The following table shows key data of the wet, dry and hybrid heat rejection systems that are employed in the monitoring installations of subtask A.

Table 6: Key data of heat rejection systems installed in Task 38 monitoring installations

		Number of systems	Nominal thermal capacity	Nominal flowrate	Nominal Flow/Return Temperature	Nominal Outdoor Conditions (Dry bulb / Wet bulb)	Air volume flow	Nominal electricity consumption (max)	Heat Exchanger area	U-Value
			kW	kg/hr	°C/°C	°C/°C	m³/hr	W	m²	W/(m² K)
Wet Cooling Towers	AXIMA EWK 036 / 06	3	35	5000	32 / 26	/ 21		330		
Dry Air Coolers	CIAT AIRIAL 7023 HI 680	1	25	2905	35 / 27	25 /		600	133	
	Güntner GFH 080.2B/1-S(D)-F4/8P	1	28,5	5422	45 / 40	32 /	10500	340	197,6	25,63
	Güntner GFH 067B/2-S(W)-F6/12P	1	24	3351	41 / 34	37,8 / 32	12900	800	270,6	28,6
	Güntner GFH 052A/2-L(D)-F6/12P	1	27	2930	43 / 35	30 /	9620	570	168,7	
Hybrid Coolers	SORTECH RCS 08	3	21	3685	31.8 / 27	24.5 / ??	13000	650	221,4	32,65
	SORTECH RCS 15	0	42	7000				1200		
	Baltimore Air Coil 1 VXI 9-3X	1	67	7360	35 / 29.5	32 / 22	9000	2200	19	

In addition, one system uses a dry air cooler integrated in the chiller. Two installations use ground coupled heat rejection systems – one with a horizontal ground heat exchanger, one using boreholes.

The table shows that the characteristic numbers of the different heat rejection units vary significantly. One of the important figures is the electricity consumption of the unit. It influences significantly the operating costs of a system and also the primary energy efficiency. Fig. 13 shows the relative electricity consumption (i.e. the electricity consumption per kW rejected heat) as a function of the nominal thermal capacity of wet, dry and hybrid heat rejection units used systems in the Task 38 (Subtask A) monitoring program.

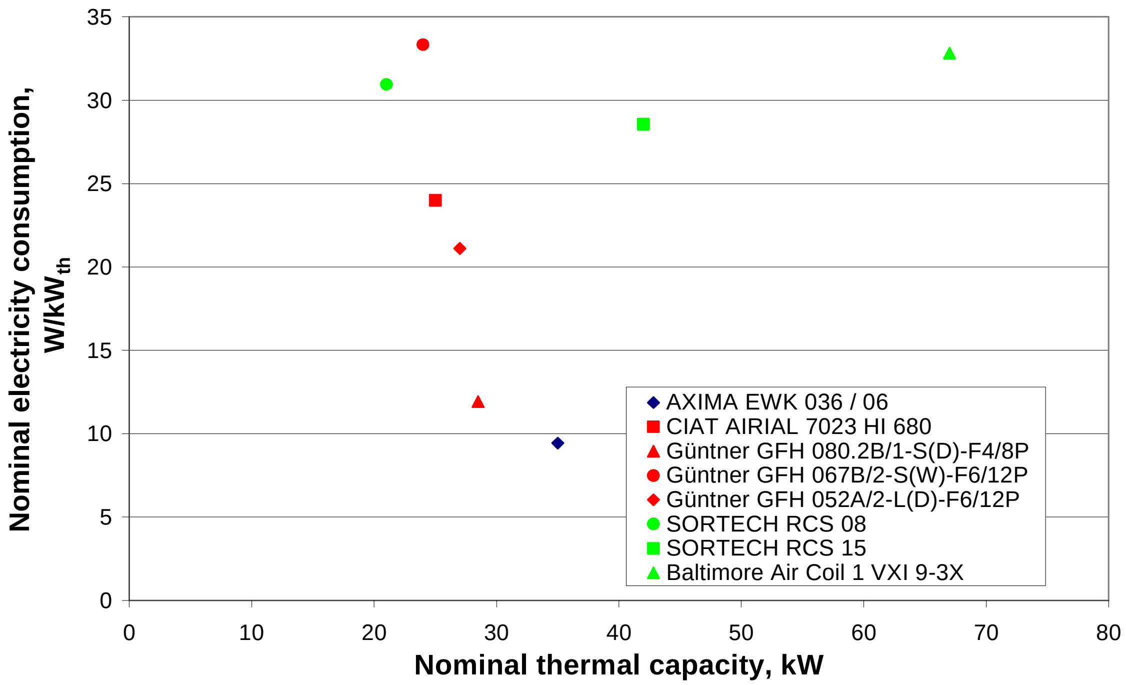


Fig. 13: Nominal electricity consumption of heat rejection systems installed in Task 38 monitoring installations in Watts per kW rejected heat

The figure shows that data points scatter significantly. For approximately the same nominal thermal capacity (25-35 kW) the electricity consumption varies from roughly 9 W/kW_{th} to 34 W/kW_{th}. The only wet cooling tower in the survey (blue dot) has the lowest consumption. The red dots show the dry heat rejection units and the green ones the hybrid systems.

This is not meant to be an exhaustive analysis of the topic. It only shows that it is important to pay attention to the choice of heat rejection unit for a specific system in order to reduce the primary energy consumption of the system.

4.3 Electricity Consumption

By Lars Reinholdt, Danish Technological Institute

If not carefully designed, the electrical consumption can grow to a level making the competition to traditional electrical driven vapor compression systems very hard. Of the total electricity consumption of the system the heat rejection system often can count for more than 50% making that a major focus point in the design phase. Three points have to be addressed

1. Flow rate and pressure drop in the water loop
2. Fans in the heat rejecting device
3. Control strategy

The electrical power for a pump can be estimated by the simple equation

$P = \eta_p * V * \Delta p_p$

η_p being the electrical efficiency of the pump, V being the volume flow of water and Δp_p being the pressure difference across the pump.

Both the volume flow and the pressure drop have a direct impact. As a rule of thumb the power consumption increase in a power of 3 of the flow rate, as the frictional pressure drop increases in a quadratic of the flow velocity.

The control strategy also has an impact on the electrical power consumption: Traditionally thermally driven chillers are controlled by controlling the driving temperature level keeping the cooling water temperature constant: If the cooling capacity is to be reduced the driving temperature is decreased.

At fixed driving and cold temperature the most thermally driven chillers have a strong dependence between the cooling capacity and the heat rejection temperature whereas the thermal COP is only affected a little. This opens for optimization of the electrical power consumption for the heat rejection in part load: By decreasing the fan speed (power savings) the heat rejection temperature will increase. As the temperature difference between air and cooling water goes up and the cooling capacity drops (constant COP) the heat rejection device will be oversized which normally results in higher efficiency and makes further power savings possible. Running in part load will normally also make it possible to decrease the cooling water flow rate resulting in reduction of the power for pumps. Further optimization can be done by letting the driving temperature increase until the limit for the respective chiller: Increasing driving temperature will increase the cooling capacity again making further savings on the pumps (lowering the flow rate) and heat rejection device. Savings up to 50% have been shown. Further this control strategy can make it possible to reach the cooling capacity at lower driving temperature by running the heat rejection fans at full (or over) speed resulting in a lower heat rejection temperature when needed (although resulting in higher power consumption).

5 Cold Stores

Some solar cooling systems using a thermally driven chiller use a cold store to be able to deliver cold at times when there is not enough solar radiation available to power the thermally driven chiller.

5.1 Cold Stores Used in Small-Scale Solar Cooling Systems

All small-scale solar cooling systems currently on the market use a water filled storage tank that stores cold at temperature between roughly 4 and 18℃. Such tanks are available from almost all manufacturers of heat storage tanks. As there are so many products available, it is not necessary to list manufacturers in this document.

5.2 Alternative Cold Storage Technologies

As an alternative to water filled storage tanks, latent heat storage can be used. The most popular option is an ice storage tank. Such tanks are used quite frequently to shift cooling loads for compression chillers to times when electricity prices are low. They are typically used for large capacity systems. However, some manufacturer would also be able to produce small tanks and some small size systems are currently under development.

In the following sections, first an overview of the different ice storage technologies that are available will be given. Then, a database of manufacturers of ice storage systems (although most of them typically manufacture large store sizes) is given. The idea behind this database is to give a starting point for companies or institutes interested in these technologies.

5.2.1 Ice Storage Technologies

By Torsten Koller and Alexander Eichhorn, ITW, University of Stuttgart, Germany

Ice storage systems are categorized under consideration of the kind of discharging. If the return flow of the cooling medium is directly mixed with the storage fluid (water/ ice) it is a so called "direct" ice storage system. If cooling fluid and storage mass are separated by a heat exchanger it is called "indirect" system. A "hybrid" system is a combination of the direct and the indirect system.

The following enumeration summarizes the different ice storage systems (Gasser, Kegel 2005) (Urbaneck et al. 2006):

- **Direct ice storage systems**
  - Ice-on-coil (external melt)
  - Sheet Ice Harvester / silo-ice storage systems
  - Ice Slurry

- **Indirect ice storage systems**
  - Ice-on-coil (internal melt) also called ice bank systems
  - Ice-on-coil (external melt use water circuit is directly coupled with the discharging circuit of the ice store)
  - Encapsulated Ice

- **Hybrid ice storage systems**
  - Ice-on-coil (combination of external and internal melt)

5.2.1.1 Ice-on-coil

The major component of an ice-on-coil system is a heat exchanger (e.g. tube coils or plate heat exchangers made out of copper, stainless steel or polypropylene) which is implemented into a non-pressurized storage tank filled with water. These heat exchangers are either directly connected to the refrigerant circuit of the chiller or have their own secondary circuit which is coupled to the refrigerant circuit. In the first case the heat exchanger in the ice store is working as an evaporator of a chiller. No additional heat exchanger is needed and temperature differences are minimized.

In case of a secondary circuit for heat transfer purpose for example a water glycol composition is used as heat transfer fluid. The main advantage is the reduction of the amount of refrigerant on the one hand and reduction of tendency for leaks on the other hand side.

In ice-on-coil systems water freezes on the outer surface of the heat exchanger tubes and a constant growing ice layer is established. The growing rate of this ice layer and the transferred cooling capacity decreases as the layer grows thicker due to the increased thermal resistance of the ice layer.

External melt ice storage systems

For external melt ice storage systems (EMISS) two different, completely separated circuits for charging and discharging are used. The ice store is charged by a heat exchanger which is implemented into the storage vessel. This heat exchanger works either directly as an evaporator or is coupled to the chiller by a secondary circuit.

During discharging mode the ice adhered to the heat exchanger melts from the outside of the ice mass to the inside. The water of the discharging circuit is returned to the storage vessel at the top, is removed at the bottom and pumped back for example to chilled ceilings at a lower temperature level. The water of the discharging circuit is the storage mass.

It is possible to achieve high discharging rates and constant water outlet temperatures of 1℃ to 2℃ with this type of system if a steady flo w through the storage vessel is guaranteed.

To ensure constant and steady flow through the ice store complete freezing of all water in the storage vessel has to be prevented. By utilization of agitators and bubblers the flow within the ice store can be significantly enhanced. The ice melts more evenly because of the enhanced mixing.

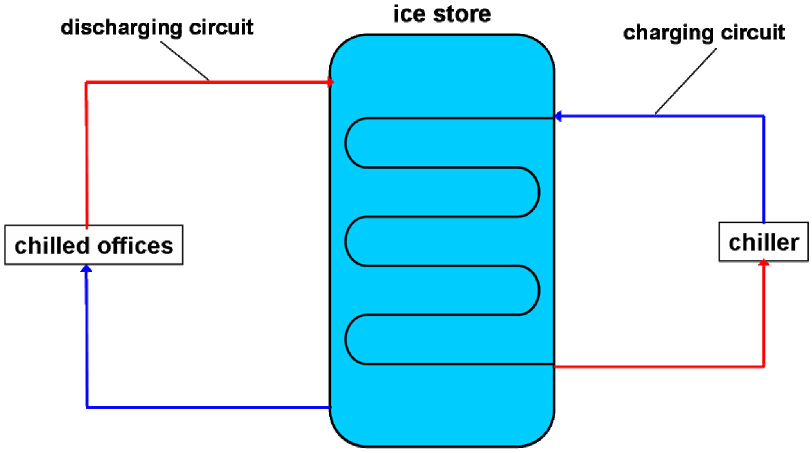


Fig. 14: Schematic diagram of an ice store with external melt

Since ice block generation has to be prevented the maximum capacity compared to internal melt systems is significantly reduced. Fig. 14 shows a schematic chart of an external melt ice storage system setup.

Internal melt ice storage systems

Internal melt ice storage systems (IMISS) use the same heat exchanger for charging and discharging. During discharging the heat transfer medium inside the coils of the heat exchanger has a higher temperature than the melting temperature of ice. The ice adhered to the heat exchanger starts to melt from the outside of the heat exchanger surface to the outer boundaries of the ice layer (Fig. 15). A growing layer of liquid water between the heat exchanger surface and the ice is generated. Hence the discharging rate decreases since the driving temperature difference decreases.

The most important advantage of internal melt systems is the possibility to realize very high storage capacities since an ice block generation doesn't have to be prevented.

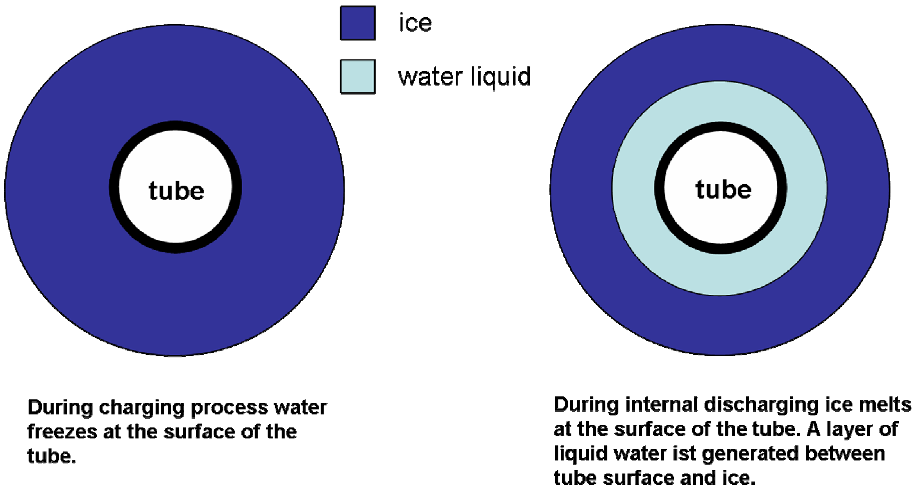


Fig. 15: Charging and discharging process for an internal melt system

A disadvantage of internal melting is that simultaneous charging and discharging is not possible. A simultaneous charging and discharging could be useful for example when an absorption chiller generates less cooling capacity than needed to meet the cooling demand. With external melt systems cooling can be maintained under these conditions. The absorption chiller then keeps charging the ice store while the cooling demand is met by discharging the ice store.

Fig. 16 shows a schematic chart of an internal melt ice storage system.

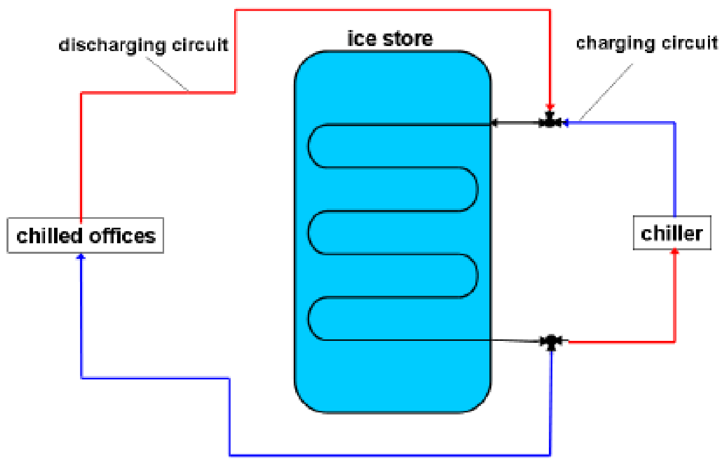


Fig. 16: Schematic diagram of an ice store with internal melt

Hybrid systems

If an ice store can be discharged via external and internal melt simultaneously it is called a hybrid system. The advantages of both systems are combined. The internal discharging circuit melts the ice mass from the outer surface of the heat exchanger. Channels of liquid water are generated between the heat exchanger surface and the ice mass. Simultaneously the external discharging circuit circulates the liquid water.

Therefore hybrid systems allow a high discharging rate at a constant temperature level. Fig. 17 shows a schematic chart of a hybrid system.

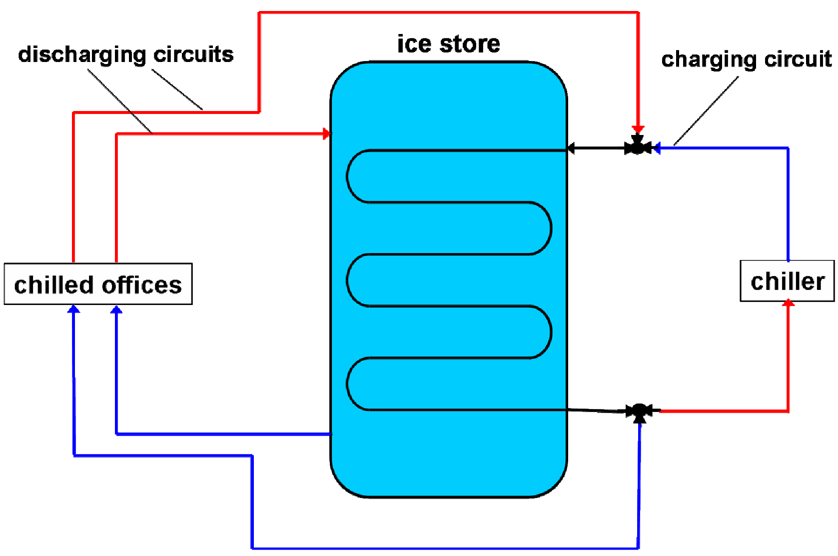


Fig. 17: Schematic diagram of a hybrid ice store

5.2.1.2 Encapsulated Ice

An encapsulated ice storage system (EISS) consists of a steel containment vessel filled with plastic containers [Siemens Schweiz AG] up to 50% to 70% of its volume capacity. The containers have various shapes and usually consist of polyethylene. The containers are filled with water or brine. They are not completely filled to account for the volume expansion during the freezing process. The containers are in direct contact with the heat transfer medium flowing through the tank. For non pressurized systems a barrier has to be installed to keep the plastic containers from buoying upwards once they are frozen. Fig. 18 is a schematic chart of an encapsulated ice storage system.

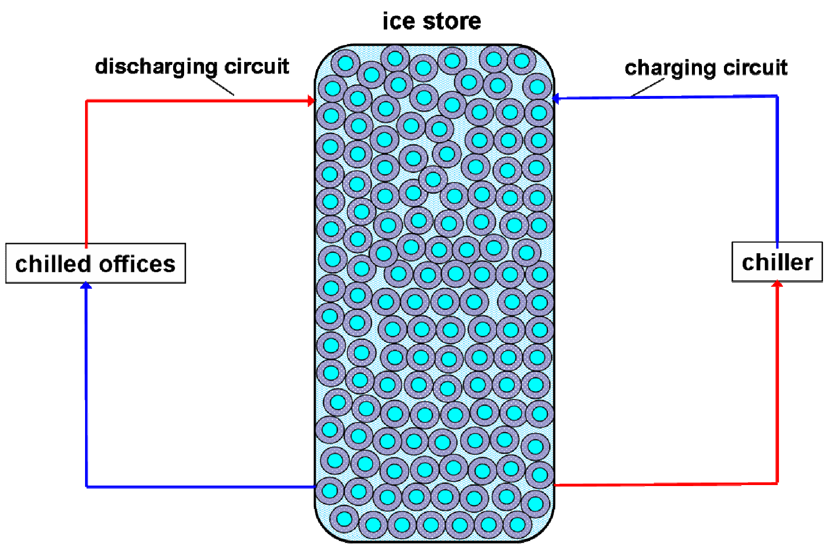


Fig. 18: Schematic diagram of an encapsulated ice storage system

During the charging process the plastic containers are floating in the heat transfer medium (e.g. water glycol mixture) which has a temperature between −3 °C and −6 °C. The ice grows from the inside of the plastic container walls to their centre.

During the discharging process the ice inside the plastic containers starts melting. It melts at the container wall first. Once the ice is completely detached from the container wall the heat transfer rate decreases since there is no direct contact between ice and plastic container wall anymore (Fig. 19). Because of this effect the discharging rate decreases and the discharging temperature increases.

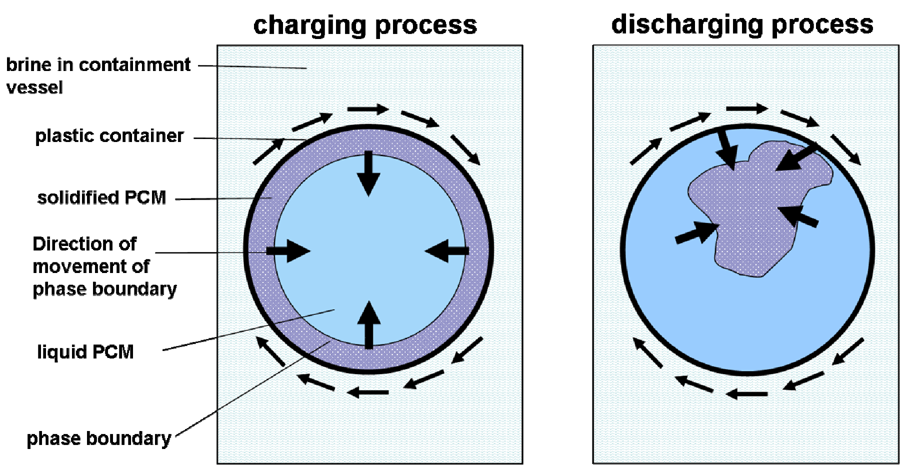


Fig. 19: Solidification and melting processes in a plastic container during charging and discharging of an encapsulated ice system

5.2.1.3 Sheet Ice Harvester

A sheet ice harvester system (SIHS), shown in Fig. 20, generates the ice outside of the storage vessel. Water extracted from the bottom of the storage vessel is sprayed on evaporator plates. An additional recirculation circuit is needed. A thin ice layer is formed at the evaporator plates. Once a thickness between 6 mm and 10 mm is reached the ice is removed by a mechanical process if the evaporator consists of tubes or by a periodical injection of hot gas if the evaporator consists of plates. The ice drops into the storage vessel below the ice generator. Water flows through this tank and the system is discharged in the same way as an external melt ice-on-coil system. Likewise low and constant discharging temperatures could be achieved. Water is sprayed on the crushed ice to guarantee a sufficient wetting of the ice. Due to this and the large surface of the small ice particles high discharging rates are achievable.

Compared to ice-on-coil systems the necessary heat transfer area is small, since the ice is directly generated on the evaporator of the chiller. The disadvantage of this system is the high complexity of controlling the SIHS due to the third recirculation circuit and the complicated process of evaluating the charging conditions. Since hot gas is injected into the ice generation process, the overall efficiency is lowered.

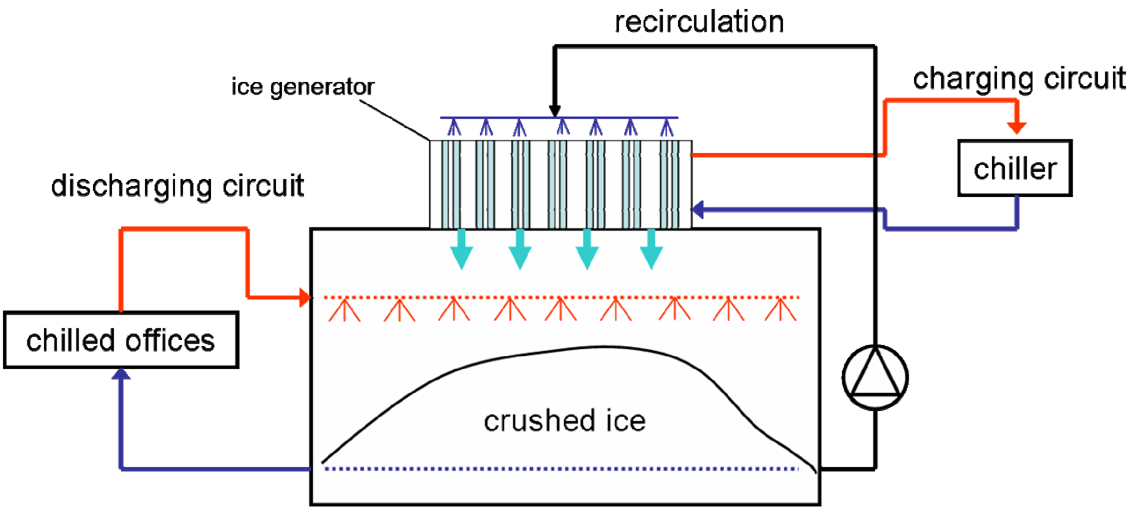


Fig. 20: Schematic diagram of a sheet ice harvester system

5.2.1.4 Ice Slurry Systems

Ice slurry is a non-Newtonian fluid (Bingham fluid) which consists of ice crystals in an aqueous solution. The main advantage of ice slurry systems is the high specific energy content due to the latent stored heat. Ice slurries are pumpable up to an ice content of about 40%. These systems can be used within a wide temperature range. The ice content can be influenced by the use of additives.

The working principle of an ice slurry system is shown in Fig. 21.

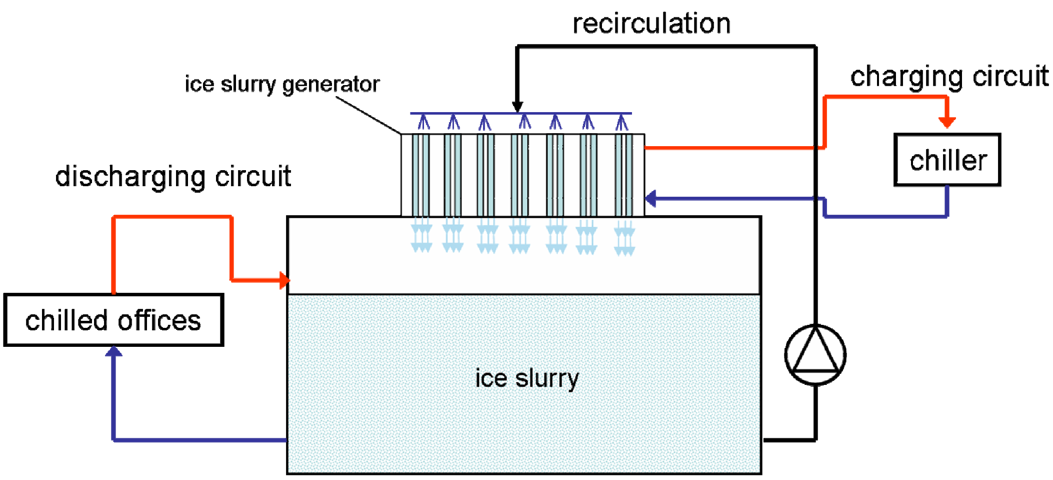


Fig. 21: schematic diagram of an ice slurry system

5.2.1.5 Summary of all discussed ice storage systems

The following Table 7 summarizes the discussed ice storage systems.

Table 7: Summary of different ice storage systems (Hilliweg, Hofmann, 2003)

physical storage principle	storage system	storage material	material carrying heat during charging	material carrying heat during discharging	specific storage capacity
sensible	chilled water	water	water	water	7 kWh/m^3 at $\Delta T = 6$ °C
latent	ice-on-coil (external melt)	ice	coolant or brine	water or brine	40-44 kWh/m^3
latent	ice-on-coil (internal melt)	ice, paraffin or eutectic mixture	brine	brine	53 kWh/m^3
latent	hybrid system	ice	brine	water or brine	53 kWh/m^3
latent	encapsulated ice	ice or eutectic mixture	brine	brine	53 kWh/m^3
latent	sheet ice harvester	crushed ice	coolant	water	–
latent	ice slurry	ice or water with additives	ice or water with additives	ice or water with additives	–

5.2.2 Addresses of Manufacturers

This is a non exhaustive list of ice storage manufacturers.

Baltimore Aircoil Company, USA P.O. Box 7322, Baltimore, MD 21227 7600 Dorsey Run Road, Jessup, MD 20794 Phone: +1 (410) 799-6200 Fax: +1 (410) 799-6416 http://www.baltimoreaircoil.com
BUCO WÄRMEAUSTAUSCHER INTERNATIONAL GMBH, Germany Sandstrasse 31 D-21502 Geesthacht Phone: +49 (0)41528082-0 +49 (0)4152-8082-43http://www.buco-international.com
CALMAC, USA Manufacturing Corporation Headquarters 3-00 Banta Place Fair Lawn, NJ 07410 Phone: +1 (201) 797-1511 Fax: +1 (201) 797-1522 http://www.calmac.com
CLIMATIC GfKK mbH, Germany Gradestraße 113-119 D-12347 Berlin (Britz) Phone +49 (0) 30 / 60 09 94 - 0 Fax: +49 (0) 30 / 60 09 94 – 99 http://www.gfkk.de
CRISTOPIA Energy Systems, France 78, chemin du Moulin de la Clue Quartier Cayrègues F- 06140 VENCE (France) Phone : +33 (0) 4 93 58 40 00 Fax: +33 (0) 4 93 24 29 38 info@cristopia.com http://www.cristopia.fr/
FAFCO S.A., Switzerland Johann Renfer-Strasse 4-6 CH-2504 Biel/Bienne Phone: +41 (0)32 / 342 33 52 Fax: +41 (0)32 / 342 39 47 Email: info@fafco.ch http://www.fafco.ch
GP Klimatechnik GmbH, Germany Hohlstr. 25 D-76275 Ettlingen Phone: +49-(0)7243-526868 Fax: +49-(0)7243-526869 Email: vertrieb@gpklimatechnik.de http://www.gpklimatechnik.de
Integral Energietechnik GmbH, Germany LISE-MEITNER-STR. 2 D-24941 FLENSBURG Phone: +49 (0)461-9993-33
Paul Mueller Company, USA 1600 W. Phelps

Springfield, Missouri 65801-0828 Phone: +1 (417) 575-9000 · 1-800-MUELLER Fax: +1 (417) 575-9669 http://www.muel.com/
Raffel - Searle GmbH Kältetechnik, Germany Auf dem Kummgraben 17 D-53343 Wachtberg Phone: +49 (0)2 28 95 29 8-0 Fax: +49 (0)2 28 95 29 8-10 Email: info@raffel-kaeltetechnik.de http://www.raffel-kaeltetechnik.de
Rössler Kühltechnik, Germany Uracher Str. 6 D-71229 Leonberg Phone: +49 (0)7152/73305 Fax: +49 (0)7152/73864
SOREMA, France Z.I. Sud Ouest - Bd du Cormier BP 455 F - 49304 Cholet Cedex Phone. +33 (0)2 41 62 30 29 - Fax : +33 (0)2 41 62 81 25 http://www.sorema.com/web/froid-industriel/
Sunwell Technologies Inc., Canada Phone :+1.905.856.0400 Fax :+1.905.856.1935 Email : inquiries@sunwell.com http://www.sunwell.com/
Tankki Oy, Finland Oikotie 2 FIN-63700 ÄHTÄRI Phone: +358-(0)6-510 1111 Fax: +358-(0)6-510 1200 Email: tankki@tankki.fi http://www.tankki.fi/
TH. WITT Kältemaschinenfabrik GmbH, Germany Lukasstr. 32 D-52070 Aachen Phone: +49 (0)2 41 1 82 08-0 Fax: +49 (0)2 41 1 82 08-190 Email: info@th-witt.com http://www.th-witt.com
VRITHERM ®, Germany Finkenweg 1 D-70771 Leinfelden-Unteraichen Phone: +49 (0)7 11 / 754 50 18 19 Fax: +49 (0)7 11 / 754 50 10 Email: info@vritherm-energie.de http://de.vritherm-energie

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