SOLAR HEATING & COOLING PROGRAMME
INTERNATIONAL ENERGY AGENCY

Task 38
Solar Air-Conditioning and Refrigeration

C5:
Heat rejection

Technical report of subtask C, work package 5

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Heat Rejection

The choice of heat rejection solution is often critical to the electrical power consumption of the thermally driven chiller. The possible lowering of the electrical consumption compared to traditional cooling solutions is in many cases the driving force towards utilizing the solar energy for cooling. Therefore the total electrical consumption is crucial. Investigation of realized systems shows that up to 50-60 % of the total electrical consumption is used in the heat rejection system, depending on the type and design of the system.

There are a number of different heat rejection units also for small thermal capacities available on the market.

In Appendix A of this document, some key data of the heat rejection units that are used in TDC (Thermal Driven Cooling) systems being monitored within the framework of IEA SHC Task 38. 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. On water cooled systems the risk of freezing at low ambient temperatures has to be taken into account. The two traditionally used solutions are

- anti freezing agent in the whole heat rejection loop
- using a heat exchanger between an in house loop based on water and a outdoor loop having anti freezing agent

In the cases where the cooling is not needed at winter time a third solution of draining off of the water is also possible.

Therefore, not all units on the market will be presented in this report, but a non-priority list of potential suppliers is available in Appendix A of this document. However, the document should give an overview of the available technologies.

Chapter 3 gives some general considerations on the electricity consumption of heat rejection units and shows ways how to reduce it.

1 Heat Rejection Technologies

Generally speaking, different heat sinks are possible to reject the heat, e.g. air, ground or water (river, lake, sea etc.). 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:

- Cooling tower (open (or wet cooling towers) and closed cooling towers
- Dry coolers

And a combination of these should be mentioned

- adiabatic pre-cooling of the air in the dry cooler
- hybrid cooling towers

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 combined 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.

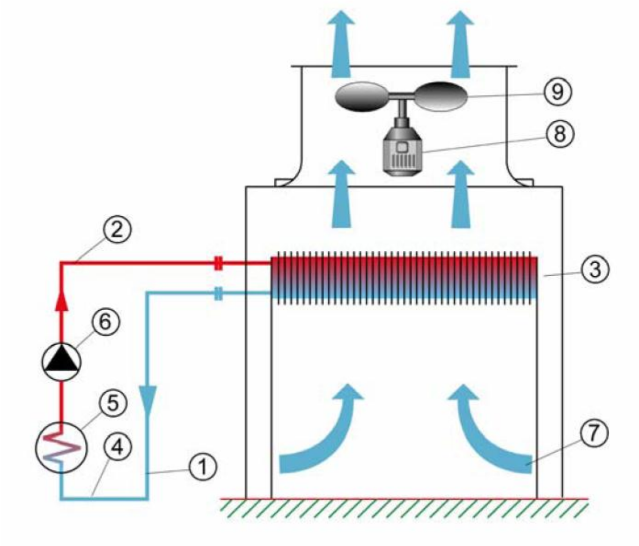
1.1 Dry Cooler

Dry coolers consist generally of finned heat exchangers (air to water), fans and a casing as schematic shown in Fig. 1. The water circulates in a closed circuit and by passing ambient air over the finned surfaces the heat is rejected to the air.

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 water 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, 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. 1: Sketch of a dry cooler (SWKI, 2005)

The performance of a dry cooler can be improved by pre cooling the air by evaporation of water. The simplest way is to spray water info the intake air but one has to careful as cark and other solutes (for instance salts) will be deposited on the surface of the cooling coil and fins even if all the water is evaporated before the air enters the cooling coil. Water treatment will reduce this, but further the risk of pollution from the ambient air (for instance acid) exists. Water treatment also adds to both installation and operation cost. Different more or less sophisticated systems exist but one has to be very careful concerning the drift of water into the cooling coil. The evaporative pre cooled dry cooler presented in 1.3 is eliminating this risk.

The cooling process in a dry cooler is shown in Fig. 2: Point A (32°C) is the ambient condition and the temperature increase of the air through the dry cooler is in the direction of B (38°C). If evaporative pre cooling of the air is done the inlet temperature will be reduced in the direction of the wet temperature C (22°C).

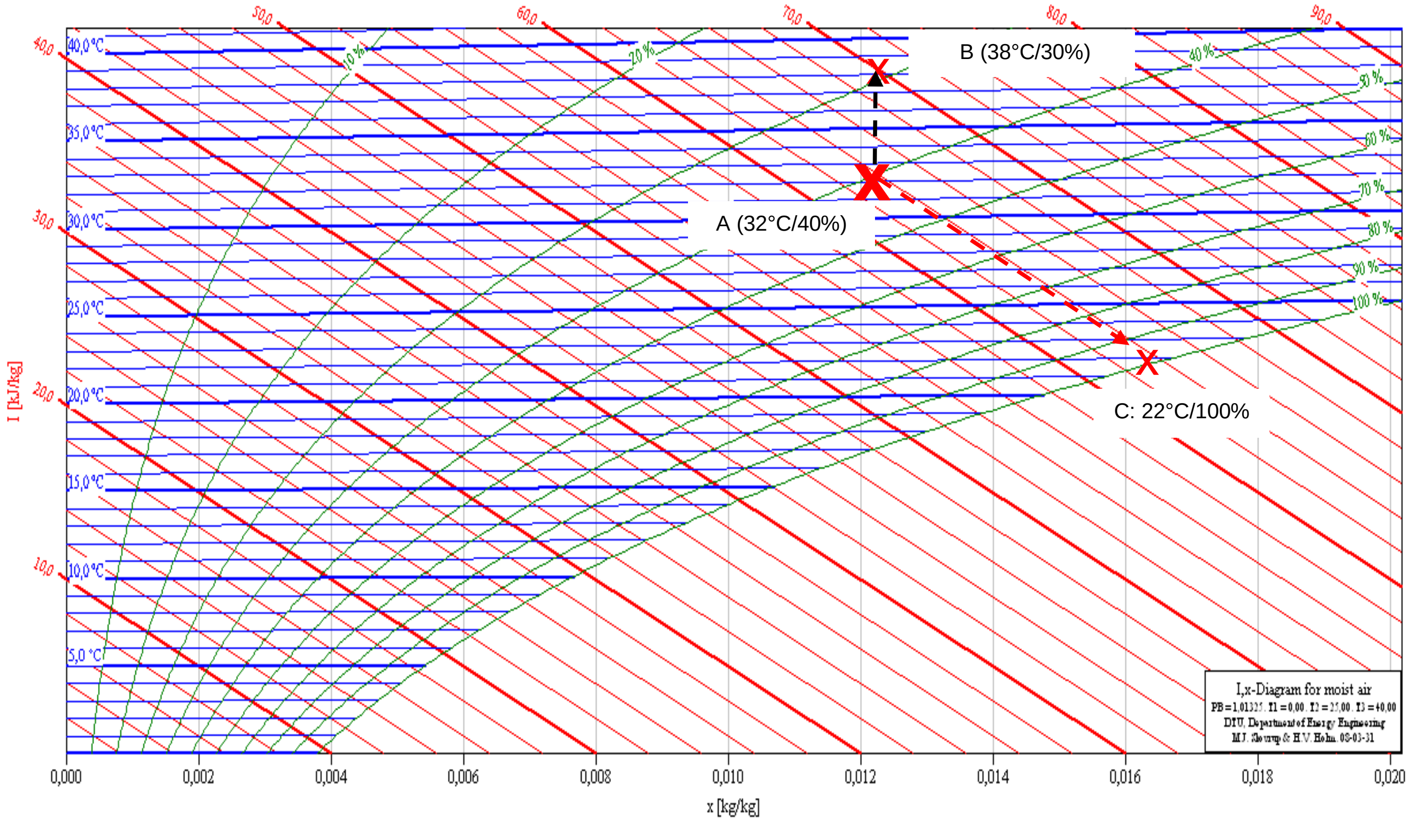


Fig. 2: The temperature change of the air in a dry cooler: A: Ambient condition, B outlet temperature from dry cooler and C wet bulb temperature of the ambient condition

1.2 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. This is described in more details later.

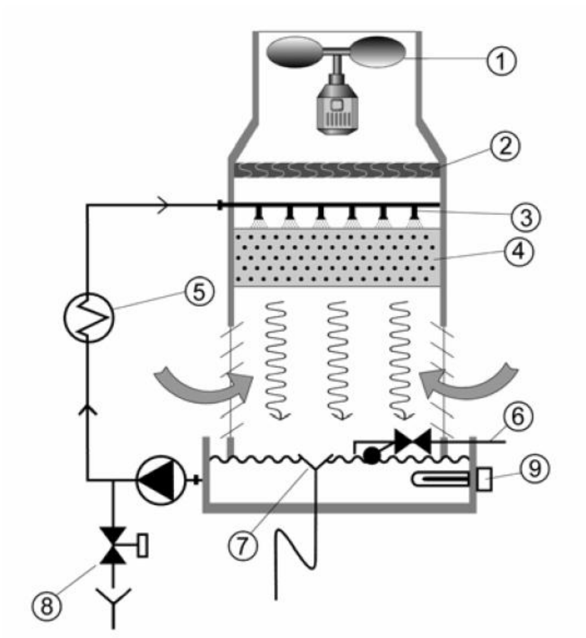
Cooling towers can be

- 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 which is 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 replaced 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 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. 3: Sketch of a open 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 closed cycle cooling tower can be seen in Fig. 4.

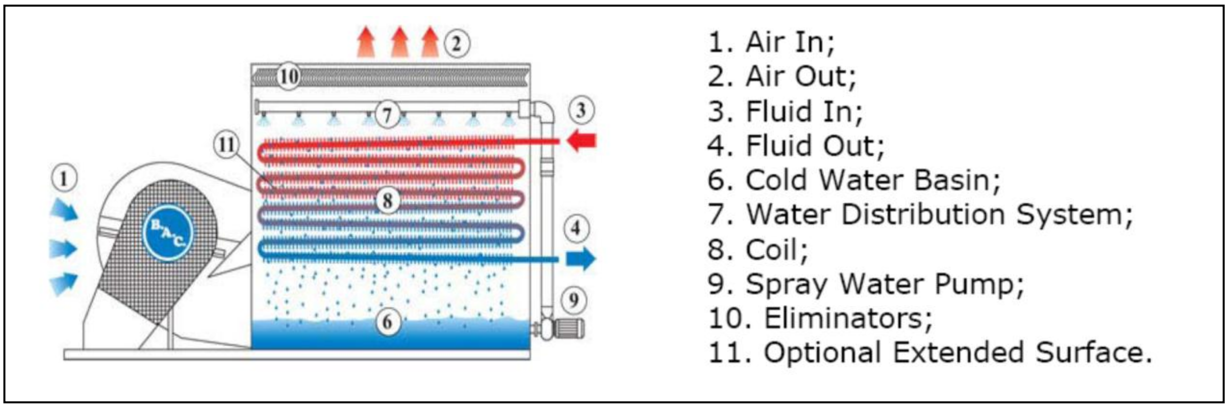


Fig. 4: Sketch of a closed cooling tower (BAC type VFL, www.baltimoreaircoil.be)

As stated the wet bulb temperature is the theoretical lowest water outlet temperature, but in reality this is not unattainable.

The internally process of the air flowing through the cooling tower can be illustrated in an IX-diagram as shown in Fig. 5.

Air with an inlet drybulb temperature T_{airi} and a wetbulb temperature T_{wb} is in contact with the water at the outlet temperature T_{wo}. The water temperature is lying on the saturation line (RH=100%). As the air is flowing through the cooling tower it absorbs more heat. The moisture content of the air is increasing as well as the temperature. The air in the cooling tower always “sees” the water at the saturation line, so the change of state of the air is always moving in direction of the temperature of the water at the saturation line. This is why the slope of the air temperature in cooling tower has the curved shape as shown.

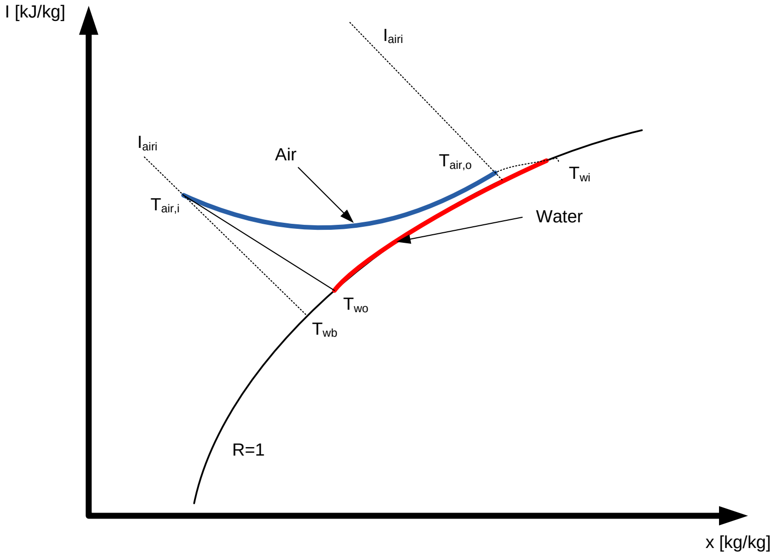


Fig. 5 IX diagram showing the air process in the cooling tower

A cooling tower is usually design according to the theory of Merkel. In his theory the outlet conditions of the air from the cooling tower always is assumed to be saturated. This is not always the case as describe in Fig. 5.

As stated the cooling capacity of the air flowing through the cooling tower is closely related to the wet bulb temperature of the entering (ambient) air. The water consumption is on the other hand highly depending on the dry bulb temperature. This is illustrated in Fig. 6. z

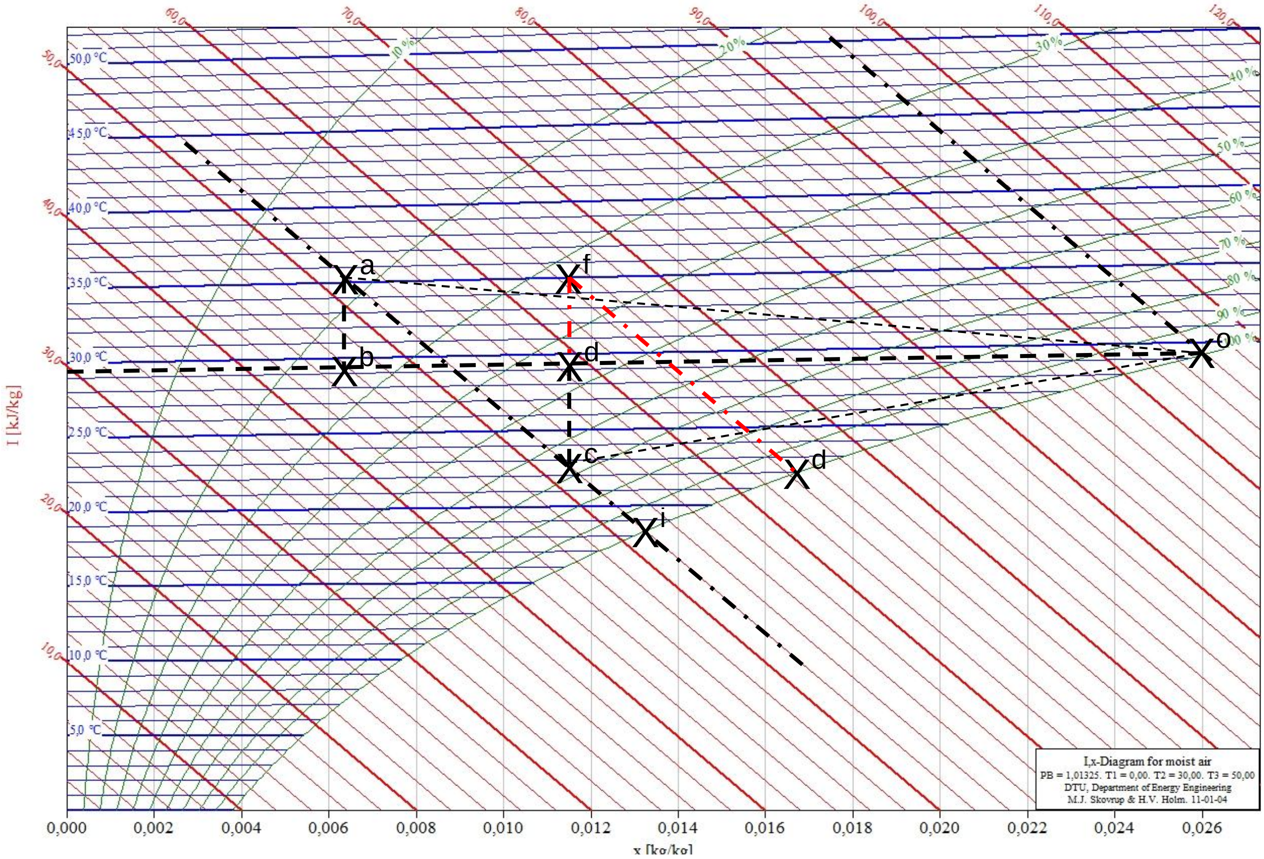


Fig. 6 Open cooling tower: Impact of ambient temperature and humidity

In the example the ambient wet bulb temperature is 18,5 °C (point "i") and the outlet temperature is 29°C ("o"). The specific cooling capacity is the enthalpy difference between hese two points: 45 kJ/kg air. Two different ambient dry bulb temperatures are indicated: 35°C ("a") and 28°C ("e") being higher and lower than the out let temperature (the line at 29,3 °C). The sensible part of the change of the air temperature is illustrated as the vertical lines "a-b" which is a cooling process! and "e-d" which takes up energy i.e. cooling the water. The latent part are the horizontal lines "b-o" and "d-o" respectively, which show the specific amount of water to be evaporated respectively in order to obtain the cooling capacity, the first being 36% higher than the other.

The figure also shows the impact of rising dry bulb temperature at constant moisture content: When the dry bulb temperature is increased from 22,5°C at "c" to 35°C at "f" the wet bulb temperature is raised from 18,5°C at "i" to 22°C at "d" lowering the cooling capacity by 27%. In other words the capacity of a cooling tower will be higher in the morning than at the peak temperature in the afternoon.

To evaporate 1 kg water from the hot water into the air 2500kJ/kg is needed and to give an idea of the water consumption the following example is set up:

In the case that the heat exchange between the hot water and the air consists of 95% latent heat and 5% sensible heat (typical values) the water consumption for evaporation in the cooling tower can be calculated to 1,37 kg cooling water/h (= 3600/2500*0,95) for every 1 kW rejected in the cooling tower.

1.3 Evaporative pre cooled dry coolers

An example of an evaporative dry cooler can be seen in Fig. 7. It consists of a water wetted pad (A) through which the ambient air is drawn before it enters the dry cooler (B). The cooler is not to been seen as a cooling tower: The air is evaporative cooled to near the wet bulb temperature when passing through the pad after which the heat is rejected as sensible heating of the air in the dry cooler. Compared to the cooling tower where the whole heat is rejected through evaporation of water this reduces the water consumption down to less that 10% (depending on the ambient condition), but the efficiency is also lower that the cooling tower.

If the air temperature is low enough the cooler can operate as a dry cooler although the pressure drop through the pads will cause a higher electrical consumption for the fan.

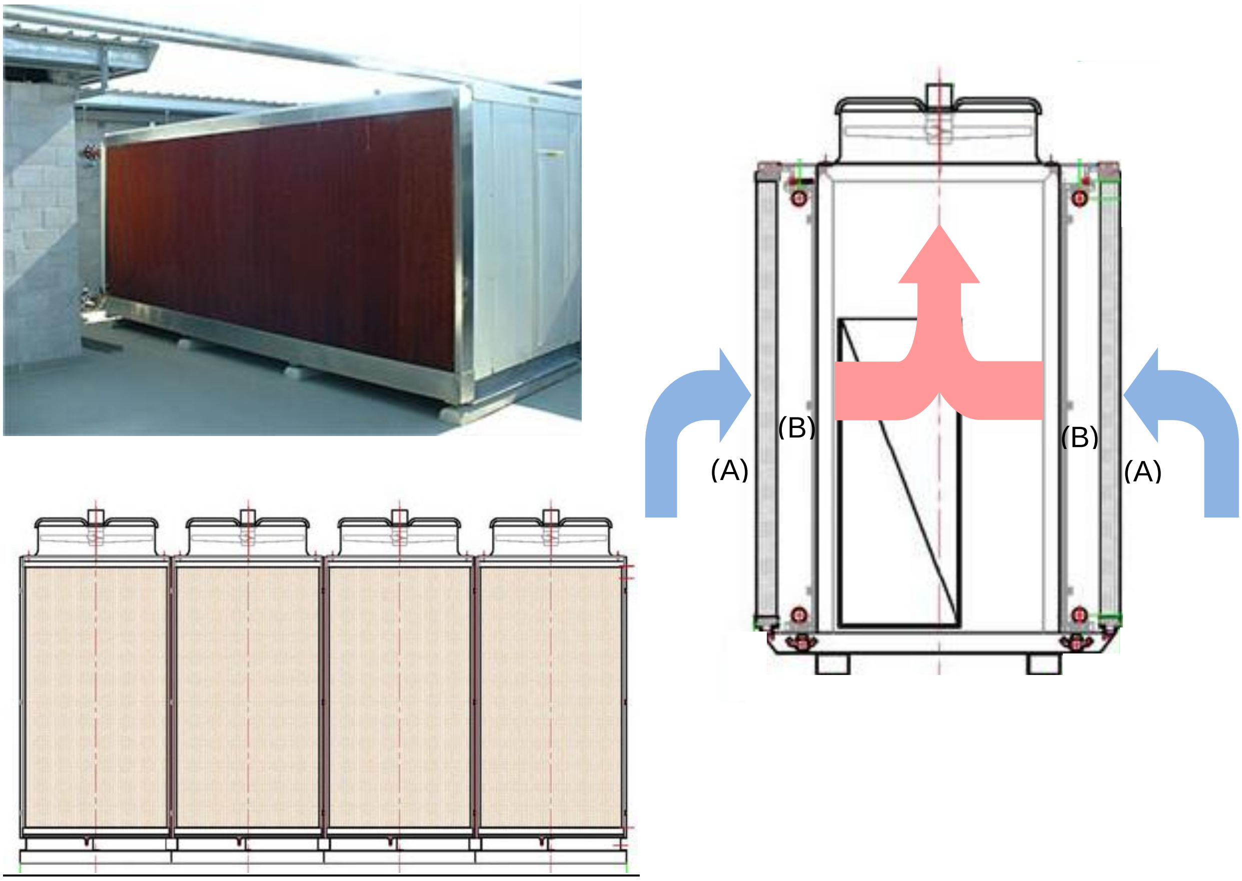


Fig. 7: Evaporative pre cooled dry cooler (Mueller Industries, Australia)

The cooling process in a dry cooler is shown in Fig. 8: Point A (32°C) is the ambient condition. The evaporative pre cooling is cooling the air to let's say point B (23°C) in the direction the wet bulb temperature. Through the dry cooling pad the temperature increases in the direction of C (28°C). The water consumption in this example is 2 g/kg air.

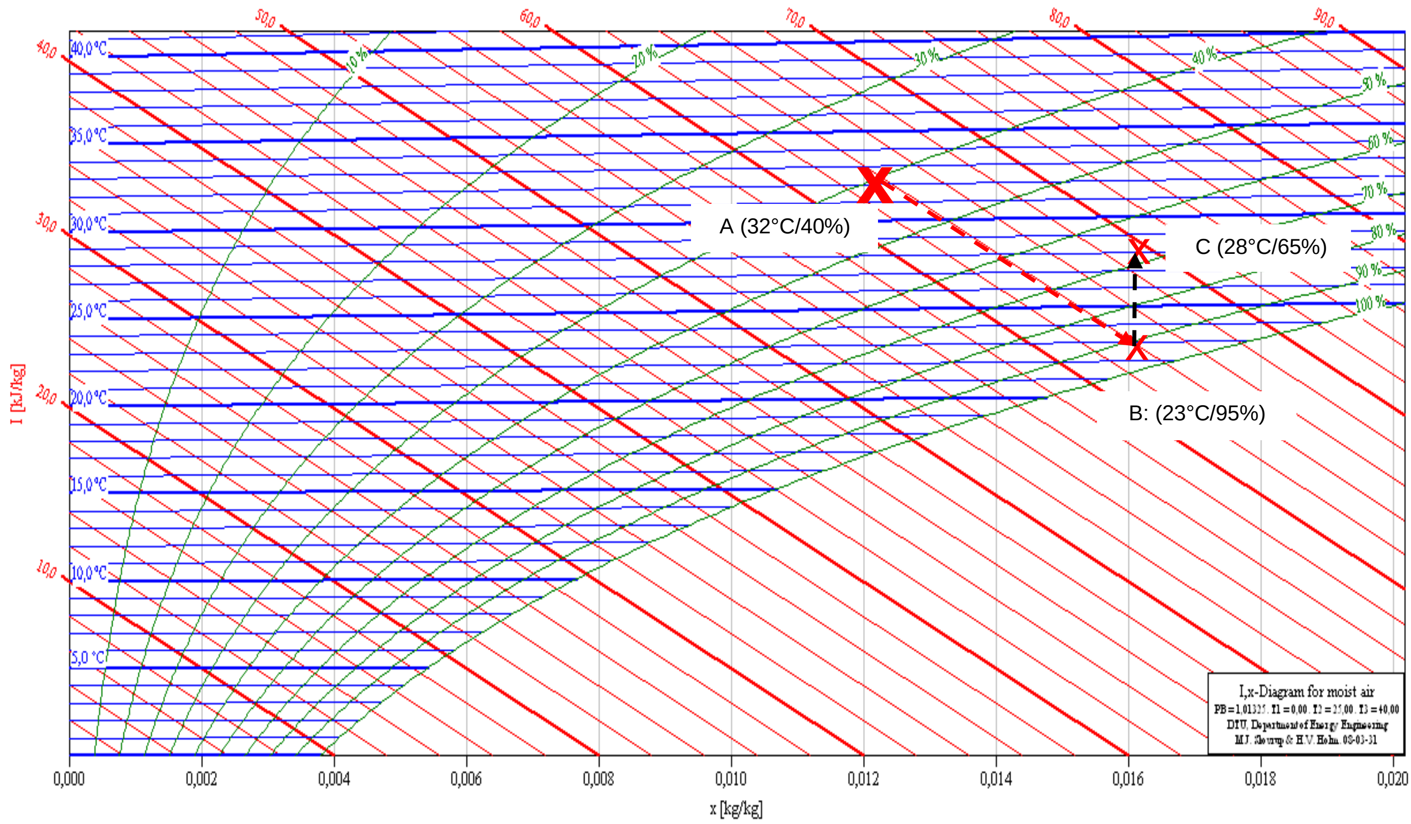
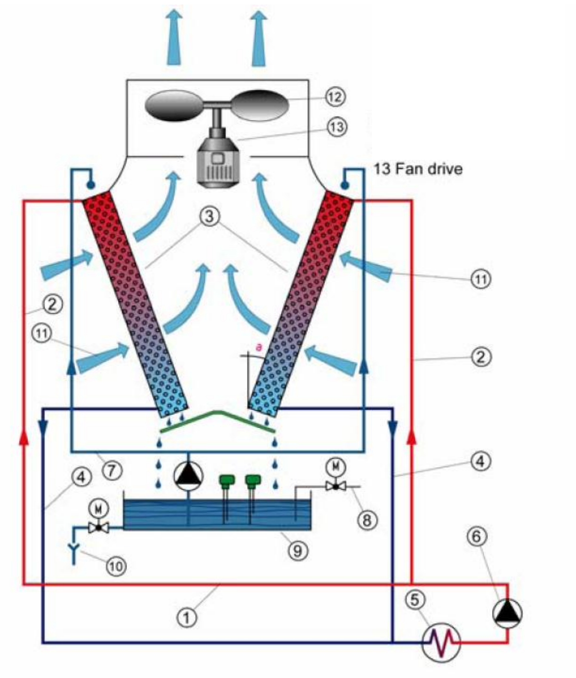


Fig. 8: The temperature change of the air in a evaporative pre cooled dry cooler: A: Ambient condition, B outlet temperature from dry cooler and C wet bulb temperature of the ambient condition

1.4 Hybrid dry cooler

The hybrid dry cooler which is shown in Fig. 9 combines the two methods of dry cooling and evaporative cooling into the same heat exchanger. 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. 9: 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.

Compared to common dry cooler a hybrid dry cooler 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.

1.5 Vertical Ground Heat Exchanger (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. For 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°C). The main parameter 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].

1.6 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. 10: 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 [VDI 2001] the following specific heat transfer numbers 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 $W/kW_{heat\ rejection}$ 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.

The Table 1 shows example data of an installed horizontal ground heat exchanger

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

Table 1: Example data of a horizontal ground heat exchanger, source: Bengt Hedestam, ECONICsystems, Gars am Kamp, Austria

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 IEA SHC Task 38 subtask A.

		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	

Table 2: Key data of heat rejection systems installed in Task 38 monitoring installations

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. 11 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.

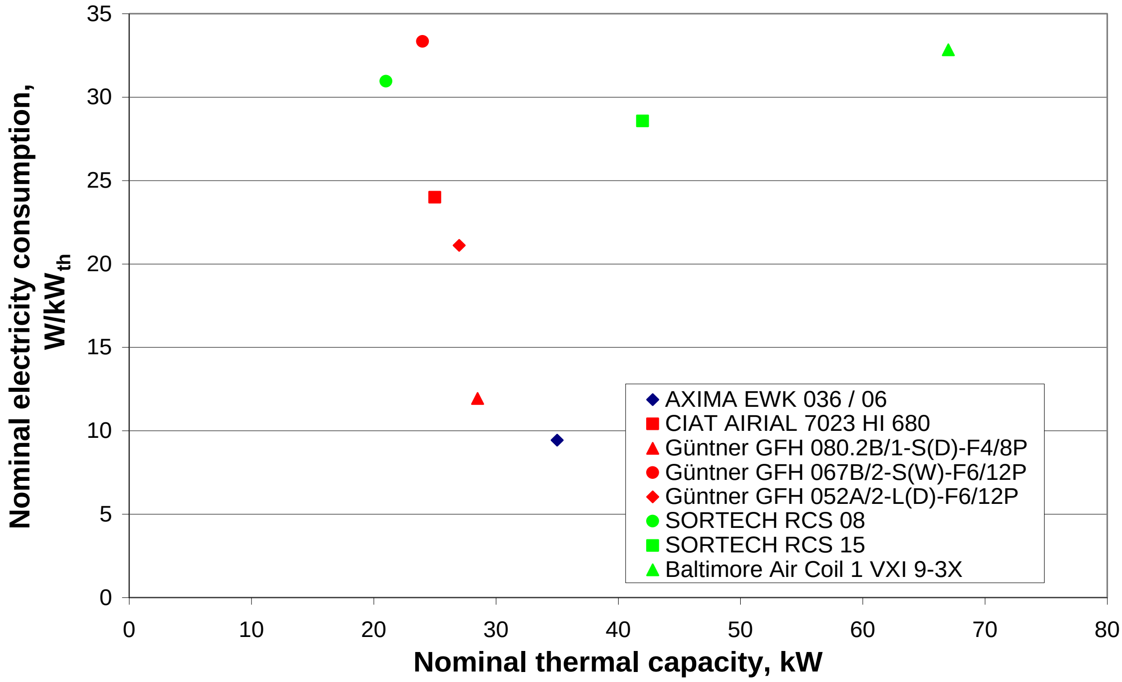


Fig. 11: Nominal electricity consumption of heat rejection systems installed in Task 38 monitoring installations in Watts per kW rejected heat

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.

3 Electricity consumption

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

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

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. As the cooling water temperature level is kept constant power and water consumption of the cooling tower is constantly high.

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 (NB: It has to be considered that this is depending on the characteristics of the specific cooling machine and does not hold for very high cooling temperatures anymore especially if driving temperature is low). 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 and cooling capacity drops. 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. As described in (Kühn et al., 2008) further optimization can be done by letting the driving temperature decrease until the limit for the respective chiller: Decreasing driving temperature can decrease the cooling capacity again making further savings on the power consumption in pumps (lowering the flow rate) and fans in the 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). This can be the case for example in the morning hours when solar irradiation is not yet high enough. By lowering the cooling water temperature full cooling capacity can be provided earlier than in the case of the conventional control strategy. This also offers the possibility to save a short-time storage tank.

4 Thermal driven cooling unit

4.1 Adsorption systems

Table 3: 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)	[°C]	72 / 65	72 / 65	85 / 79	85 / 77	65 / 59.5
	temperature range	[°C]	55..95	55..95	60-95	65-95	55-85

	(from-to)						
	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)	[°C]	27 / 32	27 / 32	30 / 40	27 / 33	27 / 31
	temperature range (from-to)	[°C]	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)	[°C]	18 / 15	18 / 15	16 / 12	18 / 15	18 / 15
	temperature range (from-to)	[°C]	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

4.2 Absorption systems

4.2.1 Ammonia – water systems

Table 4: 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)	[°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

4.2.2 Lithium – Bromide / Water systems.

Table 5: 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	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 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

4.3 Other technologies

Table 6: 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 / 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

5 Bibliography

Gasser, B.; Kegel, B. (2005): Gebäudetechnik: Faktor 10; Bau + Architektur 4/05; p. 34-35.

Grandegger, K. (2005): Einsatz von Eisspeichern in der Gebäudetechnik; FAFCO Deutschland; Leinfelden-Echterdingen; 2005

Hilliweg, A; Hofmann, P. (2003): Die Größe von Kälteanlagen minimieren; TGA Fachplaner 6-2003; p.29-35.

Juran, H. (2000).: Eis im Rohr; Chemie Technik Nr. 5; Jahrgang 29, 2000; S.92-93

Ochsner, K. (2009): Wärmepumpen in der Heizungstechnik, Praxishandbuch für Installateure und Planer, 5. überarbeitet Auflage, C.F. Müller.

Siemens Schweiz AG: Kältetechnik; Siemens Building Technologies; p.76-85.

Streicher, W. (2008): Solar Thermal Heating Systems, lecture book, Institute of Thermal Engineering, Graz University of Technology

Suter, J.-M., Letz, T., Weiss, W., Inäbnit, J. (ed.), 2000: IEA SHC – Task 26, Solar Combisystems in Austria, Denmark, Finland, France, Germany, the Netherlands and the USA, Overview 2000, Büro n+1, Bern, Switzerland, www.iea-shc.org

SWKI (2005): Richtlinie 2003-3 „Rückkühlung“, Schweizer Verein für Wärme und Klimaingenieure

Urbaneck, T.; Uhlig, U.; Platzer, B.; Schirmer, U.; Göschel ,T.; Zimmermann, D. (2006): Machbarkeitsuntersuchung zur Stärkung der Kraft-Wärme-Kopplung durch den Einsatz von Kältespeichern in großen Versorgungssystemen; Stadtwerke Chemnitz AG und Technische Universität Chemnitz Technische Thermodynamik; ISBN 3-00-015770-0.

VDI (2001): VDI 4640:2001: Thermal use of the underground, Verein Deutscher Ingenieure, 2001.

VDI (2009): VDI 4650 – 1: 2009: Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water

Kühn A.; Corrales Ciganda, J. L.; Ziegler, F. (2008): Comparison of control strategies of solar absorption chillers, Proceedings of the 1st International Conference on Solar Heating, Cooling and Buildings (Eurosun), 7-10 October 2008, Lisbon, Portugal