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PROMOTING
SOLAR AIR CONDITIONING

Technical overview of active techniques

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1 Introduction

Chiller systems based on thermally driven cold water production and desiccant cooling systems are key solutions for solar-assisted air-conditioning systems. For this reason, the most common types of thermally driven chillers are presented, namely absorption chillers and adsorption chillers, and in particular the ones which are feasible for coupling with a solar thermal energy source. Desiccant cooling technology is also presented, including a brief description of the processes.

Furthermore, vapour compression chillers are introduced since they may be used as a cold back-up source in a solar air-conditioning system and they serve as a reference for comparison between solar-assisted and conventional systems.

General remarks on the process principle

A refrigeration machine consumes energy to transfer heat from a source at a low temperature to a sink at a higher temperature. In case of air-conditioning, the heat extracted from the low temperature source is the useful cooling, i.e., the heat removed from the conditioned space, thereby producing the cooling effect. In the vast majority of air-conditioning applications, the intermediate temperature heat sink is the external environment and the heat is rejected to the external air. The driving energy is heat in case of a thermally driven process and is mechanical energy in case of a conventional refrigeration machine. In most cases the mechanical energy is delivered by an electrically driven motor, at least in case of building air-conditioning.

As a result of the first law of thermodynamics, the flux of heat rejected at the intermediate temperature level, $\dot{Q}_{intermediate}$, is equal to the sum of the heat flux extracted from the low-temperature heat source, $\dot{Q}_{low}$, and the driving power of the the process, P_{drive}, i.e., $\dot{Q}_{intermediate} = \dot{Q}_{low} + P_{drive}$. For an electrically driven chiller, the driving power ist the elctricity input to the motor, P_{el}. In the case of thermally driven chillers, the driving energy flux, P_{drive}, is a heat flux at a high temperature level, $\dot{Q}_{low}$. The principle for the example of a thermally driven chiller is shown in Figure 1.1.

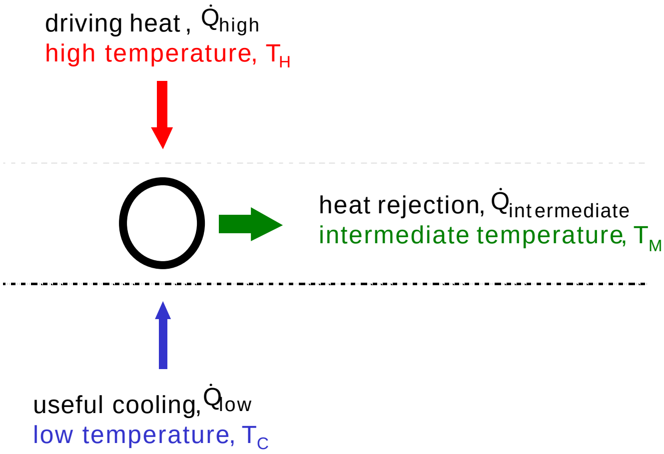

Figure 1.1. Schematic diagram of energy flows in a thermally driven machine operating a refrigeration cycle.

A key figure to characterise the energy performance of a refrigeration machine is the *Coefficient of Performance*, COP. For thermally driven air-conditioning systems, the $COP_{thermal}$, which indicates the required heat input for the cold production, can be defined as follows:

$$COP_{thermal} = \frac{\dot{Q}_{low}}{\dot{Q}_{high}} = \frac{Heat\ flux\ extracted\ at\ low\ temperature\ level}{Driving\ heat\ flux\ supplied\ to\ cooling\ equipment} \qquad \text{(Eq. 1.1).}$$

The $COP_{thermal}$ varies with the equipment operation conditions, i.e., the three temperature levels, the percentage of load etc.; therefore COP-values of different systems are only comparable if the same operation conditions are considered. For a conventional, electrically driven vapour compression chiller, the COP_{conv} is defined as the required electricity input for production of cooling energy:

$$COP_{conv} = \frac{\dot{Q}_{low}}{P_{el}} = \frac{Heat\ flux\ extracted\ at\ low\ temperature\ level}{Electrical\ power\ supplied\ to\ the\ chiller} \qquad \text{(Eq. 1.2).}$$

The COP-values of conventional chillers and of thermally driven refrigeration machines cannot be directly compared since the quality of the energy input (exergy content) is different. A method that is commonly used for an appropriate comparison is based on the primary energy consumption. This method is outlined in section 6 of this chapter.

2 Chillers

2.1 Absorption chillers

The working principle of an absorption system is similar to that of a mechanical compression system with respect to the key system components evaporator and condenser. A vapourising liquid extracts heat at a low temperature (cold production). The vapour is compressed to a higher pressure and condenses at a higher temperature (heat rejection). The compression of the vapour is accomplished by means of a thermally driven ‘compressor‘ consisting of the two main components absorber and generator. Subsequently, the pressure of the liquid is reduced by expansion through a throttle valve, and the cycle is repeated.

Absorption cycles are based on the fact that the boiling point of a mixture is higher than the corresponding boiling point of a pure liquid. The steps of the absorption cycle are:

1. The refrigerant evaporates in the evaporator, thereby extracting heat from a low-temperature heat source. This results in the useful cooling effect.
2. The refrigerant vapour flows from the evaporator to the absorber, where ist is absorbed in a concentrated solution. Latent heat of condensation and mixing heat must be extracted by a cooling medium, so the absorber is usually water-cooled using a cooling tower to keep the process going.
3. The diluted solution is pumped to the components connected to the driving heat source (i.e., generator or desorber), where it is heated above its boiling temperature, so that refrigerant vapour is released at high pressure. The concentrated solution flows back to the absorber.
4. The desorbed refrigerant condenses in the condenser, whereby heat is rejected at an intermediate temperature level. The condenser is usually water-cooled using a cooling tower to reject ‘waste-heat‘.
5. The refrigerant flows to the evaporator through an expansion valve, the pressure of the refrigerant condensate is reduced in this step.

A schematic drawing of a basic absorption cycle is shown in Figure 2.1.

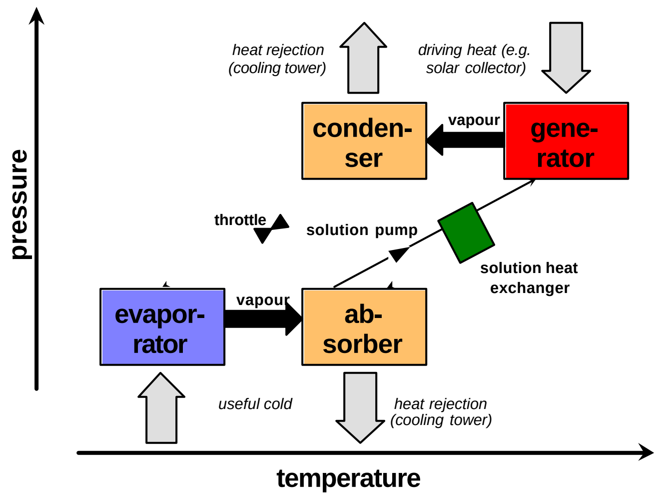

Figure 2.1. Schematic drawing of an absorption chiller for chilled water production. The main energy input is the heat supplied to the generator. Electrical energy is necessary to drive the solution pump, unless a system with a bubble pump is used.

The heat required for step 3 as desribed above, can be supplied, for instance, by direct combustion of fossil fuels, by waste heat or by solar collectors. Depending on the required cooling effect, one of the following working pairs for absorption chillers is commonly used:

- For a temperature of the low temperature heat source higher than 5°C, for example when used for air-conditioning, a water/lithium-bromide (LiBr) pair absorption machine is most frequently used, which must be water cooled.
- For a temperature of the low temperature heat source below 5°C, for example when used for refrigeration, an ammonia/water machine can be used, which may be cooled by either air or water.

In the water/lithium-bromide absorption chiller, water is the refrigerant, and cooling is based on the evaporation of water at very low pressures. Since water freezes below 0°C, the chilling temperature meets a physical limit at this level. LiBr is soluble in water if the LiBr mass fraction of the mixture is less than 70%. Crystallisation of the LiBr will occur at higher concentrations and may damage the machine. This sets a maximum temperature for the absorber. Poor control of temperature or a fast change of conditions may cause crystallisation. Appropriate operating controls will prevent this kind of problem. In order to sufficiently reduce the temperature of the absorber and dissipate the heat from the condenser, it is necessary to use a wet cooling tower.

For solar-assisted air-conditioning systems with common solar collectors, single-effect LiBr absorption chillers are the most commonly used systems, because they require a comparatively low temperature heat input. The term ‘single-effect‘ refers to the fact that the supplied heat is used once by a single generator. Thermodynamic restrictions in the system dictate that the cooling capacity for ideal and real systems is always less than the heat input. As a consequence, the COP for large single-effect machines lie in the range of 0.7 to 0.8 for standard operation conditions. Any deviation from the standard operation conditions, i.e., from the nominal volume flow rates and from the nominal temperatures in all of the three temperature levels, will cause a deviation in the chilling capacity and in the COP from the nominal values.

A double-effect absorption chiller can be viewed as two single-effect cycles stacked on top of each other. The top cycle requires heat at a higher temperature level compared to a single-effect machine. Generally, it is driven either directly by a natural gas or oil burner, or indirectly by supplying steam. In the top cycle (primary generator), refrigerant vapour is generated at a higher temperature and pressure relative to the bottom cycle. The vapour is then condensed at this higher temperature and pressure, and the heat of condensation is used to drive the generator of the lower cycle (secondary generator), which is at a lower temperature. Double-effect cycles have a higher $COP_{thermal}$ than single-effect cycles. Typical operation COP's of double-effect absorption chillers are close to 1.1 or slightly above and typical driving temperatures lie in the range of 140°C to 160°C. Current research is concentrating on three and four-effect systems, which present an attractive potential for improved cooling performance, with a $COP_{thermal}$ of 1.7 to 2.2; but these systems require a distinctly higher temperature of driving heat.

Figure 2.2. Photograph of a 52 kW single-effect absorption chiller installed in a plant for solar air-conditioning of a wine cellar in Banyuls/France.

The need for higher driving temperatures makes double-effect chillers less suitable for solar-assisted air-conditioning systems using common solar collectors. It is possible to use high efficient solar collectors to reach higher temperatures but this will increase the installation, operation and maintenance costs.

Absorption chillers are commercially available from many manufacturers. The choice on the market is quite extensive, but most machines have a large capacity. Examples of commercially available absorption chillers suitable for solar-assisted air-conditioning are presented in Table 2.1. Only the smallest available size identified from each manufacturer is shown. An example of a 52 kW single-effect chiller is shown in Figure 2.2.

Manufacturer	Chilling power, type*	Driving temperature ** (°C)	Typical operation conditions, rated COP (if available)
Broad Air	20 kW single- and double-effect	No data	No data
Colibri / Stork	100 kW NH_3 / H_2O single-effect	> 90	Example: $T_{cooling\ water}$ 27/32°C, $T_{chilled\ water}$ < 2°C: COP = 0.64
Coolingtec	70 kW R-134a/organic materials single-effect	70 – 145	Example: T_{drive} 90°C, $T_{cooling\ water}$ 27°C, $T_{chilled\ water}$ 2°C: COP ~ 0.55
Dunham-Bush	327 kW single-effect	Steam 112	Chilled water 51 m^3/h, cooling water 105 m^3/h, steam 777 kg/h
EAW	15 kW single-effect	75 – 95	Example: T_{drive} 85°C, $T_{cooling\ water}$ 30°C, $T_{chilled\ water}$ 12°C: COP = 0.7, hot / chilled water 2 m^3/h, cooling water 5 m^3/h
Sanyo	105 kW single-effect	85 – 95	Hot water 26.5 m^3/h, chilled water 8°C, cooling water 29.4°C
Trane	380 kW single-effect	Steam 171, hot water 132	Chilled water 59 m^3/h, cooling water 92 m^3/h, steam 990 kg/h: COP = 0.63
Yazaki	35 kW single-effect	80 – 100	Example: T_{drive} 87°C, $T_{cooling\ water}$ 30°C, $T_{chilled\ water}$ 9°C: COP = 0.7, hot water 8.6 m^3/h, chilled water 6 m^3/h, cooling water 14.6 m^3/h
York	420 kW double-effect	> 116	Chilled water 65 m^3/h, cooling water 98 m^3/h

* all water / lithium-bromide unless otherwise indicated
** driving source: water, unless otherwise indicated

Table 2.1. Examples of commercially available absorption chillers suitable for solar-assisted air-conditioning (only smallest available size included). The list does not claim to be exhaustive.

2.2 Adsorption chillers

Instead of absorbing the refrigerant in an absorbing solution, it is also possible to adsorb the refrigerant on the internal surfaces of a highly porous solid. This process is called adsorption. Typical examples of working pairs are water/silica gel, water/zeolite, ammonia/activated carbon or methanol/activated carbon and other similar materials. However, only machines using the water/silica gel working pair are currently available on the market. In absorption machines, the ability to circulate the absorbing fluid between the absorber and desorber results in a continuous loop. In adsorption machines, the solid sorbent has to be alternately cooled and heated to be able to adsorb and desorb the refrigerant. Operation is therefore intrinsically periodic with time. The cycle may be described as follows (see Figure 2.3):

1. The refrigerant previousely adsorbed in the one adsorber is driven off by the use of hot water (left compartment in Figure 2.3);
2. The refrigerant condenses in the condenser and the heat of condensation is removed by cooling water;

3. The condensate is sprayed in the evaporator and evaporates under low pressure. This step produces the useful cooling effect;
4. The refrigerant vapour is adsorbed onto the other adsorber (right compartment in Figure 2.3). Heat is removed by the cooling water.

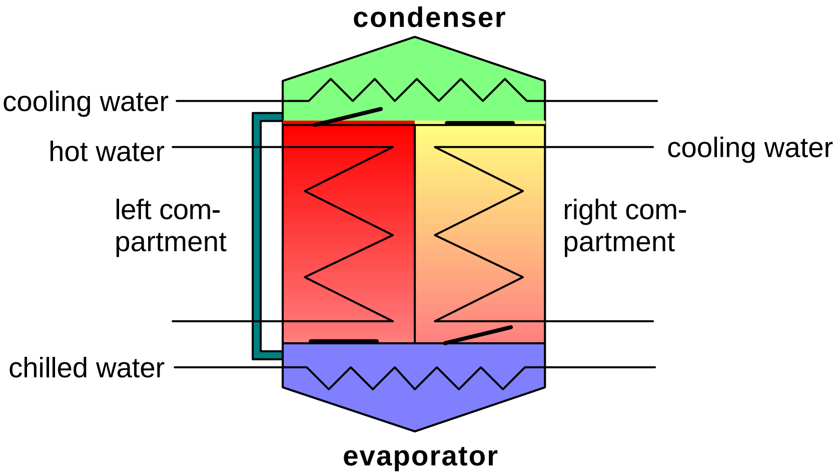

Figure 2.3. Schematic drawing of an adsorption chiller.

Once a compartment has been fully charged (saturation of the silica gel with water) and the other compartment fully regenerated, their functions are interchanged. In between, the two chambers may be directly coupled in order to achieve some heat recovery, since the hot chamber has to be cooled in the next step and vice versa. The time dependent temperatures in an adsorption chiller are shown in Figure 3.2; it can be seen, for example, that for this particular machine, a periodic change between the two compartments always takes place after about seven minutes. Adsorption chiller require for generation temperatures in the range from 60°C to 90°C and thus can operate at lower temperatures compared to absorption chillers.

Only a few manufacturers produce adsorption chillers. The performance characteristics of some commercially available adsorption chillers are summarised in Table 2.2. An example of a 70 kW adsorption chiller is shown in Figure 2.4.

Manufacturer	Chilling power	Driving temperature (°C)	Design conditions and rated COP
Mayekawa	70 kW water / silica gel	55 – 90	T_{drive} 75°C, $T_{cooling\ water}$ 29°C, $T_{chilled\ water}$ 9°C: COP = 0.60
Nishiyodo	67 kW water / silica gel	55 – 95	T_{drive} 90°C, $T_{cooling\ water}$ 29°C, $T_{chilled\ water}$ 7°C: COP = 0.65

Table 2.2. Commercially available adsorption chillers suitable for solar-assisted air-conditioning (only smallest available size included). This list does not claim to be exhaustive.

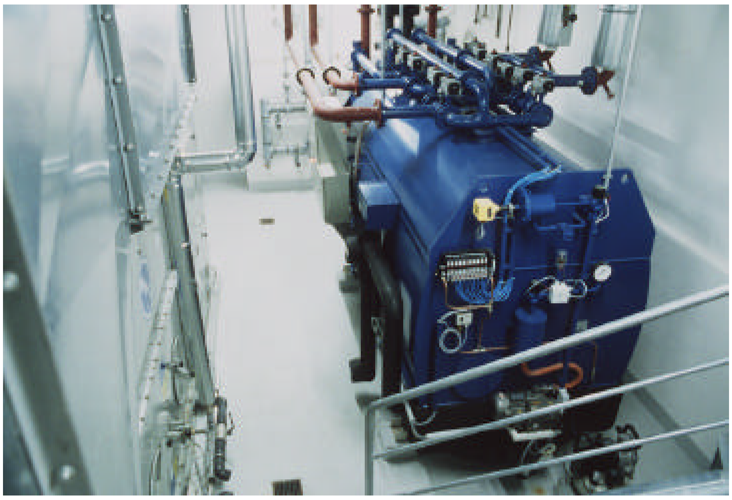

Figure 2.4. Adsorption chiller from the manufacturer Nishyodo; the machine is used for solar-assisted air-conditioning of a laboratory building at a hospital in Freiburg/Germany

2.3 Cooling towers

The implementation of an absorption chiller or of an adsorption chiller requires an additional cooling tower installation to allow the heat rejection at the intermediate temperature level. Since this system component consumes non-negligible amounts of electricity and water, a brief description of wet cooling towers will be added.

A cooling tower is a specialised heat exchanger where cooling water is brought into contact with ambient air to transfer rejected heat from the coolant to the ambient. For this purpose, two basic types of systems can be found: open-circuit systems, where there is direct contact between the primary cooling-water circuit and the air, and closed-circuit systems where there is only indirect contact between the two fluids across heat exchanger walls. Open-circuit systems are commonly known as ‘open cooling towers‘, ‘wet cooling towers‘ or just ‘cooling towers‘. A characteristic feature of all such systems is that they mostly use latent heat transfer where the coolant, which has to be water, is cooled by evaporating about 2%-3% of the coolant itself. This results in highly efficient cooling operation, even at coolant temperatures below ambient temperature, together with minimum investment cost, but it is accompanied by significant water consumption at any operational modes. Closed–circuit systems, on the other hand, show a great variety of types and operational modes. These range from dry air coolers, transferring just sensible heat to ambient air, to a second type of wet cooling tower, incorporating an auxiliary water circuit for spraying heat exchanger tube bundles at the air side and primarily utilising latent cooling. In addition, there are several hybrid systems which combine both cooling modes, latent cooling by evaporation of water and sensible cooling against ambient air, or which are able to switch between both cooling modes in dependence on the ambient conditions and on the cooling demand.

However, all closed-circuit systems generally show less efficient operation, increased electricity consumption due to larger fans and at least doubled investment costs in comparison to open-circuit cooling towers. Further here, only open-circuit cooling towers will be discussed. A schematic drawing of such a cooling tower is shown in Figure 2.5.

The basic function of a cooling tower is to ensure a good heat and mass transfer between the cooling water stream and ambient air. Thus, the hot water enters the upper part of the cooling tower, where it is evenly distributed across the tower by a spraying system. To increase the effective contact surface between water and air, there is additional filling material installed inside the cooling tower. At the bottom of the tower, the cooled water is collected again in a reservoir. To ensure sufficient air-flow through the tower, a fan is installed that either forces entering air into the tower or sucks discharge air at the outlet. Additional installations for water treatment and blow-down are required for all cooling towers to replace the evaporated cooling water and to prevent fouling.

The performance of a wet cooling tower mainly depends on the wet bulb temperature of the ambient air, while it is only slightly affected by the ambient temperature. The design limit for the temperature of the cooling water leaving the tower is about only 3-5°C above the wet bulb temperature, which typically is still below ambient air temperature. As a cooling tower operates with about 90% latent cooling even at low ambient temperatures, the water evaporation can directly be estimated from the cooling load; however at least 50% additional blow-down has to be considered to obtain the total water consumption. Since there is a highly non-linear relation between air temperature and water vapour saturation pressure, no simple equations can be given to describe the operational behaviour of a cooling tower at different operational states.

Typical design and performance figures for an open-circuit wet cooling tower are:

Air volume flow:	130 – 170 m^3/h per kW of cooling power;
Electricity consumption of the fan:	6 –10 W per kW of cooling power for axial ventialtors, 10 – 20 W per kW of cooling power for radial ventilators;
Control:	In order to save energy, it is recommended to equip the ventilator with a frequency control, so that the fan velocity can be adapted to the required cooling power.

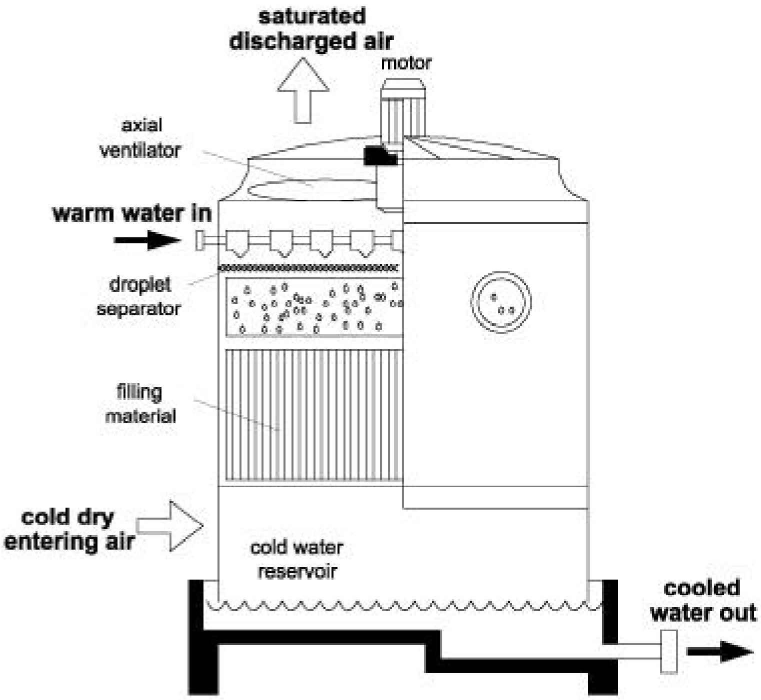

Figure 2.5. Schematic drawing of an open type wet cooling tower.

2.4 Vapour compression chillers

The most common refrigeration process applied in air-conditioning is the vapour compression cycle. Most of the cold production for air-conditioning of buildings is generated with this type fo machine. The process employs a chemical refrigerant, e.g., R134a. A schematic drawing of the system is shown in Figure 2.6.

In the evaporator, the refrigerant evaporates at a low temperature. The heat extracted from the external water supply is used to evaporate the refrigerant from the liquid to the gas phase. The external water is cooled down or – in other words – cooling power becomes available. The key component is the compressor, which compresses the refrigerant from a low pressure to a higher pressure (high temperature) in the condenser.

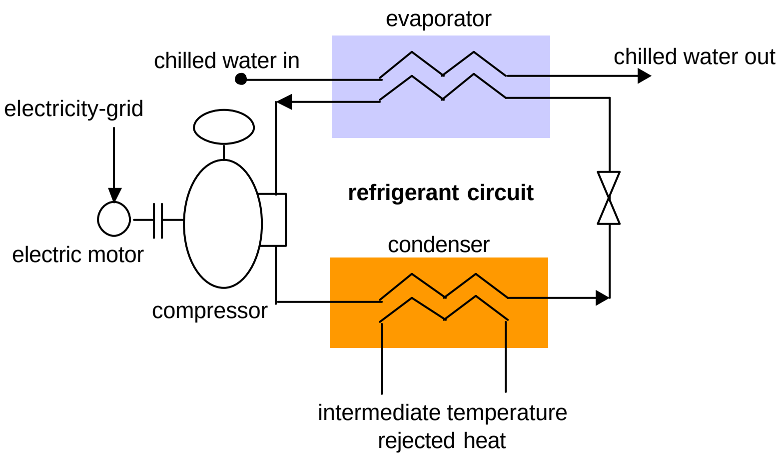

Figure 2.6. Schematic drawing of a vapour compression chiller.

Electrical energy is consumed by the motor used to drive the compressor. Thus, it is possible to reject the heat from the refrigerant at a higher temperature; for this purpose, either direct air cooling or a wet cooling tower is used. In the next step, the expansion valve throttles the pressure to the necessary pressure in the evaporator.

Typical COP_{conv} values and capacity ranges of the most common compression machines are:

Reciprocating compressors
COP_{conv} 2.0 – 4.7
Chilling capacity 10 – 500 kW

Screw compressors
COP_{conv} 2.0 – 7.0
Chilling capacity 300 – 2000 kW

Centrifugal compressors
COP_{conv} 4.0 – 8.0
Chilling capacity 300 – 30000 kW.

The COP_{conv} of vapour compression chillers depends on the pressure difference between evaporator and condenser and thus on the temperature difference between the evaporator and the condenser. Higher temperature differences lead to a reduced COP_{conv}. Concepts that make lower temperature differences possible are therefore beneficial since they reduce the energy consumption of the process.

3 Desiccant cooling systems

The use of sorption air dehumidification – whether with the help of solid desiccant material or liquid desiccants – opens new possibilities in air-conditioning technology. This can offer an alternative to classic compression refrigeration equipment. Alternatively, if it is combined with standard vapour compression technology, it leads to higher efficiency by an increase of the required evaporator temperature of the compression cycle.

Desiccant systems are used to produce conditioned fresh air directly. They are not intended to be used as systems where a cold liquid medium such as chilled water is used for heat removal, e.g., as for thermally driven chiller based systems. Therfore, they can be used only if the air-conditioning system includes some equipment to remove the surplus internal loads by supplying conditioned ventilation air to the building. This air-flow consits of ambient air, which needs to be cooled and dehumidified in order to meet the required supply air conditions. Desiccant cooling machines are designed to carry out these tasks.

Economic advantages arise for desiccant cooling equipment when it is coupled with district heating or heat supplied from a combined heat and power (CHP) plant. Of particular interest is the coupling with thernal solar energy. The components of such systems are generally installed in an air-handling unit and are activated according to the operation mode of the air-conditioning system. These operation modes implement different physical processes for air treatment, depending on the load and the outdoor air conditions. These systems are based on the physical principle of evaporative and desiccant cooling. Unsaturated air is able to take up water until a state of equilibrium, namely saturation has been achieved. The lower the relative humidity of the air, the higher is the potential for evaporative cooling.

The evaporative cooling process uses the evaporation of liquid water to cool an air stream. The evaporation heat that is necessary to transform liquid water into vapour is partially taken from the air. When water comes into contact with a primary warm air stream it evaporates and absorbs heat from the air, thus reducing the air temperature; at the same time, the water vapour content of the air increases. In this case, the supply air is cooled directly by humidfication and the process is referred to as direct evaporative cooling.

Indirect evaporative cooling involves the heat exchange with another air stream (usually the exhaust air), which has been previously humidified and thus cooled. In this case, the water vapour content of the primary air stream is not influenced.

These two techniques of evaporative cooling can also be combined in a process that is known as combined evaporative cooling.

Complementing combined evaporative cooling with desiccant dehumidification enhances the cooling capacity of the cycle and thus it is possible to reach even lower temperatures. This combined cooling process is referred to as *desiccant cooling*.

Using evaporative cooling, either direct, indirect or in a combined process, it is not possible to reduce the vapour content of the ventilation air. But, using a desiccant cycle, in principle lowering of the temperature and the humidity ratio of ventilation air is possible.

Fresh air conditions have a considerable effect on the amount of cooling that can be achieved. If the outdoor air is properly pre-treated, the ventilation air can be cooled to lower temperatures via subsequent indirect and direct evaporative cooling. For this purpose, the pre-treatment involved is the desiccant dehumidification process to enhance the potential of evaporative cooling without obtaining a disproportionate high humidity ratio.

The dehumidification process uses either liquid or solid desiccants. Systems working with solid desiccant materials use either rotating wheels or periodically operated, fixed-bed systems. Systems employing liquid desiccants use air-desiccant contactors in the form of packed towers or the like.

Regeneration heat must be supplied in order to remove the adsorbed (absorbed) water from the desiccant material. The required heat is at a relatively low temperature, in the range of 50°C to 100°C, depending on the desiccant material and the degree of dehumidification. Moreover, the solar desiccant cooling system, depending on the cooling loads and environmental conditions, will use one of the above mentioned cooling modes, i.e., direct evaporative cooling and/or indirect evaporative cooling and/or desiccant cooling, with the aim of providing comfort conditions in the building. The most commonly used desiccant cooling process, which is based on the use of desiccant wheels, works as follows (see Figure 3.1 and 3.2):

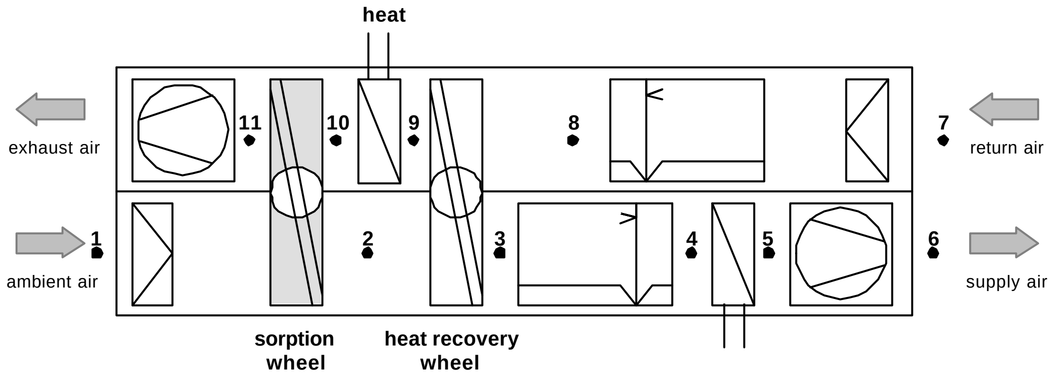

Figure 3.1. Schematic drawing of a desiccant cooling air-handling unit.

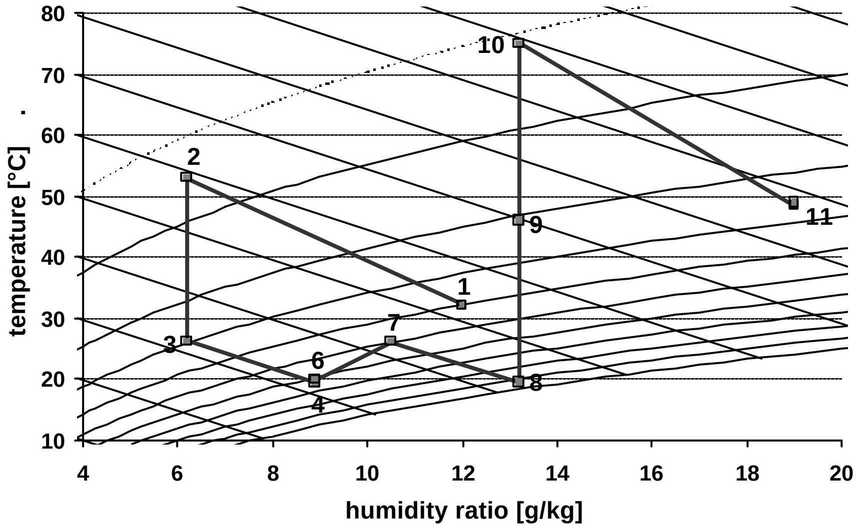

Figure 3.2. Typical desiccant cooling process in the T-x-diagram.

The ambient air (1) is dehumidfied in a desiccant wheel, causing the air temperature to increase; the process is nearly adiabatic (2). The regenerative heat recovery leads to cooling of the air inlet to the humidifier, by means of indirect evaporative cooling (3). Depending on the air inlet temperature and humidity supplied, the temperature is reduced by direct evaporative cooling in the humidifier, with a simulataneous increase in humidity up to condition (4). The coil in the supply stream is in operation only for heating conditions. The fan releases heat, leading to an increase in the temperature of the supply air to condition (5). An increase in temperature of up to 1°C is usually expected. A proper design of the fan is recommended so as to minimise the heat added to the supply air.

The return air from the room is in state (6). The air is then humdified as close as possible to saturation (7). This state is the one which guarantees the maximum potential for indirect cooling of the supply air stream through the heat exchanger for heat recovery. The heat recovery from (7) to (8) leads to an increase in the temperature of the air, which is then used as regeneration air. The air is subsequently reheated in the coil until it reaches state (9). The temperature of the latter is adjusted such as to guarantee the regeneration of the sorption wheel (9 to 10).

Figure 3.3. Example of a desiccant air-handling unit with desiccant wheel (nominal air-flow of 4500 m³/h).

It is important to mention that in many desiccant systems a bypass is installed which allows that some of the air coming from the heat recovery unit bypasses the regeneration air heater and the desiccant wheel. Depending on the actual conditions, more than 20% of the air can go through the bypass thus saving regeneration heat and also electricity, because of the reduced pressure drop along the desiccant wheel.

The $COP_{thermal}$ of a desiccant cooling system is defined as the ratio between the enthalpy change from ambient air to supply air, multiplied by the mass air-flow, and the external heat delivered to the regeneration heater, Q_{reg}:

$$COP_{thermal} = \frac{\dot{m}_{supply}\,(h_{amb} - h_{supply})}{\dot{Q}_{reg}} = \frac{\dot{m}_{supply}\,(h_1 - h_6)}{\dot{Q}_{reg}} \qquad \text{(Eq. 3.1).}$$

The value of $COP_{thermal}$ of a desiccant cooling system depends strongly on the conditions of ambient air and supply air. Under normal design conditions, a $COP_{thermal}$ of about 0.7 is achieved and the cooling power lies in the range of about 5-6 kW per 1000 m³/h of supply air. An example of a desiccant air-handling unit with a configuration such as in Figure 5.1 is shown in Figure 3.3.

The sorption dehumidification unit is a central component in a desiccant cooling system, which is not implemented in most of the standard air-handling units. For this reason, Table 3.1 provides a list of desiccant wheel manufacturers, along with a short description of the available products.

Company	Country of Origin	Desiccant	Wheel Size	Disposition
Munters USA	US	SiGel, AITi, Silicates, New Proprietary	0.25-4.5 m	Own use
Munters AB	Sweden	SiGel, AITi, Silicates, New Proprietary	0.25 - 4.5 rn	Own use
Seibu Giken	Japan	SiGel, Am, Silicates, New Proprietary	0.1 - 6 m	Own use, export: US, South America & Europe
Nichias	Japan	SiGel, Mol. Sieves	0,1 - 4 m	Export
DRI	India	SiGel, Mol, Sieves	0.3 - 4 m	Own use, Export
Klingenburg	Germany	Al oxide, LiCl	0.6 - 5 m	Export, to OEMs
ProFlute	Sweden	SiGel, Mol Sieves	0.5 - 3 m	to OEMs
Rotor Source	US	SiGel, Mol Sieves	0.5 - 3 m	to OEMs
NovelAire	US	SiGel, Mol Sieves	0.5 - 3 m	to OEMs

Table 3.1. Manufacturers and product description of sorption dehumidifiers. The list does not claim to be exhaustive.

Desiccant cooling with liquid sorbent

Liquid sorbent agents can also be used for the dehumidification of conditioned air. The liquid desiccant system is essentially an open-cycle absorption system, where water serves as the refrigerant. However, wheras a large number of working fluid pairs are available for closed absorption refrigerating machines, there is only a small number of suitable materials for open liquid-based systems which can be used for the conditioning of ventilation air. This is due to the strict limitations that apply to aqueous, hygroscopic solutions, since they come in direct contact with the environment. The solutions used should be non-toxic and environmentally friendly, and should not contain any volatile material other than water. In practice, liquid sorbent agents which consist principally of salts dissolved in water are mainly used, e.g., lithium chloride or calcium chloride. These hygroscopic salts lower the vapour pressure of water in solution sufficiently to absorb humidity from the air. In contrast to the case of the solid sorbents, the water bonding mechanism is not adsorption, but absorption.

The sorption systems used for the drying of air consist basically of an absorber and a regenerator, as shown in Figure 3.4. These are air-solution heat and mass exchangers, normally in the form of packed towers, where air and solution come into contact in counter-flow or cross-flow. Both types of equipment may be identical in structure, i.e., they have the same type of exchange surfaces and usually differ only in terms of their relative dimensions. Humidity is absorbed from the process air into the hygroscopic solution in the absorber. Then the salt solution is regenerated so that the same initial concentration is always available when drying the air. In order to remove the absorbed water out of the dilute solution, heat at a relatively low temperature level is required; temperatures of about 60°C to 70°C are sufficient.

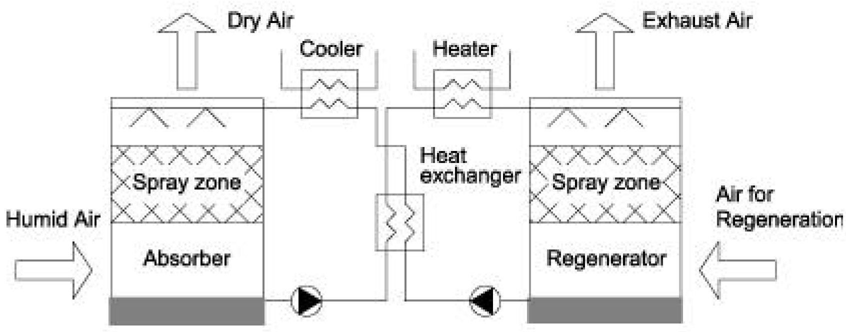

Figure 3.4. Schematic drawing of a liquid sorption system

An advantage of liquid desiccant systems is the ability to store cooling capacity by means of the regenerated desiccant. Thus, hygroscopic salt solution may be concentrated when solar energy is available, and used to dehumidify process air later, when needed. The dehumidification process can be operated as long as a concentrated desiccant is available and is independent of the availablility of driving heat for regeneration at the same time. This form of cold storage is the most compact, requires no insulation and can be applied for indefinitely long periods of time.

The concentration difference between concentrated and diluted solution can be increased by cooling the absorption process and using a special design of the absorber. To cool the absorption process, either an air-cooled or water-cooled absorber may be employed. This feature of high concentration difference increases the use of energy storage separating concentrated and diluted desiccant.

Only a few manufacturers offer liquid sorption systems at present. However, the commercially available systems are not well adapted to the use of solar energy for desiccant regeneration. Pilot plants of solar-driven systems are in operation in several demonstration projects.

4 New chiller technologies

A new development for solar assisted air conditioning is a chiller based on the steam jet cycle. Currently, this type of chillers is preferably used in industrial processes with high chilling power demand and continuous operation. In pilot projects and pre-investigations, this chiller principle is adapted to small size units (20-200 kW cooling power) to be combined with solar thermal systems. As an advantage in comparison to absorption and adsorption chillers, COP values at part-load operation may exceed the value 1.0. Only water is used in the fluid cycle and the construction principle is simple without moving parts. Chilling power control is obtained by switching up to four steam jet units stepwise according to the load demand. The driving temperature of the proposed steam jet cycle chillers is typically 200°C and thus, concentrating collectors with tracking systems are required. As a result of current studies the system can be economically competitive compare to other solar cooling technologies. Further cost reductions are likely with series manufacturing of adapted steam jet systems.

Further development has been made on absorption chillers, using Ammonia-Water as working fluid. This type of chillers is usually manufactured with large capcacities for refrigeration temperatures below 0°C, requiring high generation temperatures above 100°C and running with a comparatively low thermal COP around 0.5. In pilot projects, promising NH3/H2O chillers have been adapted to work with low generator temperatures between 65°C-80°C and thus can be heated by solar thermal systems. The evaporation temperature in these developments is approx. 5°C and the COP exceeds 0.6.

5 Application of solar assisted air-conditioning systems

As there are many different solutions in the planning of a solar assisted air-conditioning system are possible, a precise knowledge on the load structure plays a key role, to support the decision on a specific type of a solar assisted air conditioning system.

The load structure of a building or of a particular area of a building depends on the physical properties of the building and of the climate conditions at the location and thus on the thermal and radiative solar gains, and of course on the usage of the building: occupancy and frequency of occupancy, and internal additional loads due to the technical equipment of the rooms. High latent loads have to be removed with high air exchange rates of an air handling unit typically in case of lecture or seminar rooms, to give an example. Here desiccant cooling technology might be an appropriate choice. But also in case of a low occupation density (office rooms), the supply air may have to be cooled and dehumidified at locations with high humidity of the hot ambient air, as it may be found necessary in some mediterranean areas. It is evident that a clear picture on the load structure can be achieved only by considering all items in a comprehensive load calculation, carried out using cooling load calculation tools (like the standard VDI 2078 in Germany) or in a building simulation program. Further more, the cooling demand can be decreased using energy saving equipment and applying passive techniques, such as night ventilation in combination with the thermal inertia of the building.

Since a high exploitation of the solar system is desired, the heating demand of the building should be integrated into the calculation as well.

Other questions may arise from a construction point of view or from economic considerations: is it always possible to implement the required air ducts or is it more favourable to implement a chilled water network? Which system matches at best the existing technical infrastructure of a building, which is not newly designed? For instance, centralized air handling systems using supply and return air require a certain level of air tightness in order to work efficiently.

However, the following Table 5.1 provides a simplified decision scheme on possible applications of solar assisted air conditioning. It has to be kept in mind that the following tasks cannot be considered in this condensed presentation:

- Necessity of a backup system for the cold production or to allow solar autonomous operation of the solar assisted air conditioning system;
- Flexibility in comfort conditions, e.g. to allow certain deviations from the desired air states;
- Economical issues;
- Availability of water for humidfication of supply air or for cooling towers;
- Comfort habits for room installations: fan coils have lowest investment cost, but allow dehumidification only when connected to a drainage system; chilled ceilings and other gravity cooling systems require for high investment cost, but provide high comfort.

It is not decided here, whether an adsorption chiller or an absorption chiller is applied. This is more subject to the heating system (e.g. type of collectors). If a DEC system is applied, the additional required chiller for peak-load cooling may be an electric driven compression chiller for economical reasons. The solar heat is then used for the regeneration of the desiccant wheel.

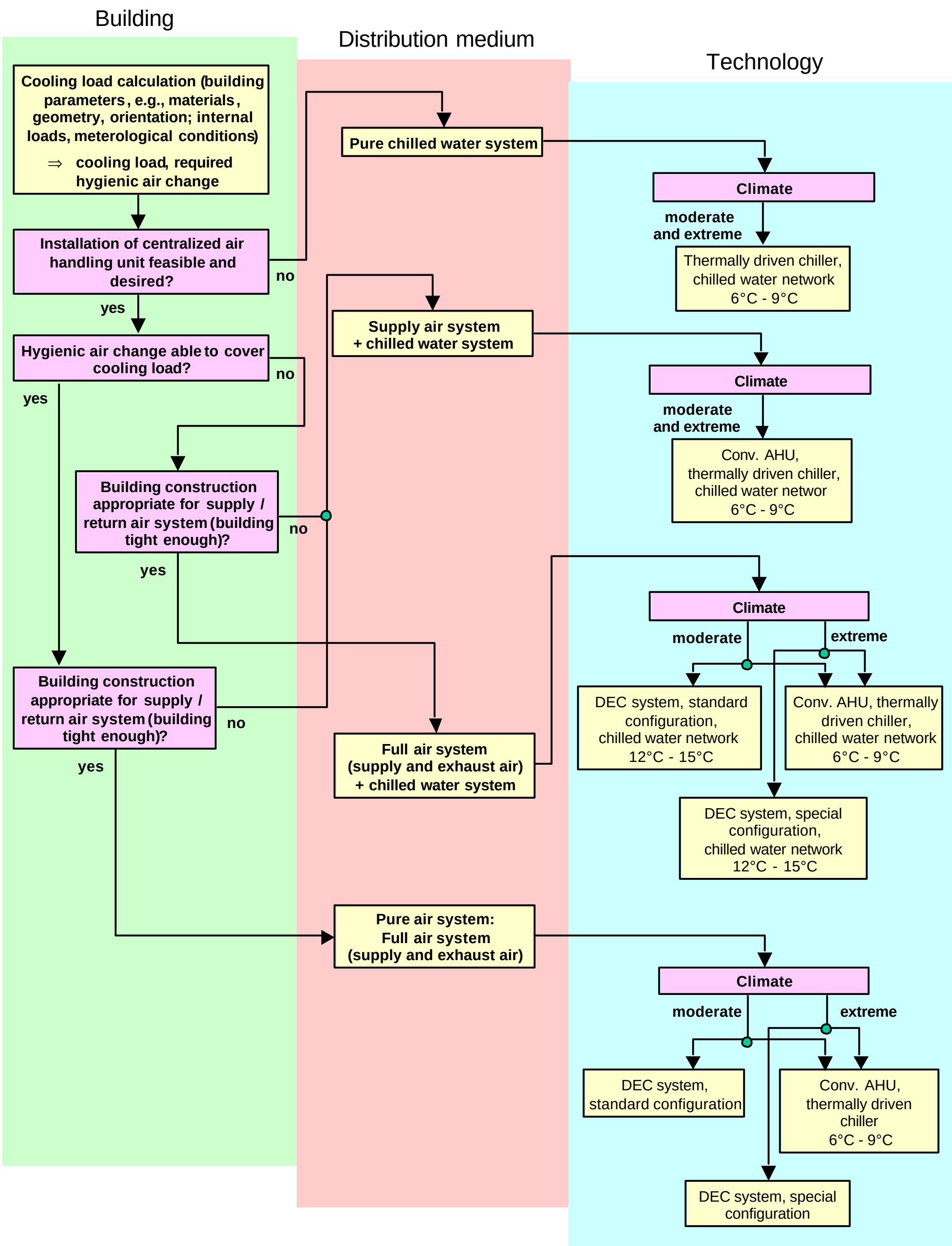

Figure 5.1. The figure shows a simplified decision scheme for solar assisted air conditioning technologies. A basic assumption is that both, temperature and humidity of indoor are to be controlled. Finally, each decision results in a solution which includes use of solar thermal energy for conditioning of indoor air. The starting point always is a calculation of cooling loads based on the design case. Depending on the cooling loads and also according to the desire of the users/owner, either a pure air system, a pure water system or hybrid air/water systems are possible for extraction of heat and humidity out of the building. The basic technical decision is whether or not the hygienic air change is sufficient to cover also cooling loads (sensible + latent). This will typically be the case in rooms/buildings with a requirement of high ventilation rates, such as e.g. lecture rooms. However, a supply/return air system makes only sense in a rather tight building, since otherwise the leakages through the building shell is to high. In cases of supply/return air systems both thermally driven technologies are applicable, i.e., desiccant systems as well as thermally driven chillers. In all other cases only thermally driven chillers can be used in order to employ solar thermal energy as driving energy source. The lowest required temperature level of chilled water is determined by the question whether air dehumidification is realized by conventional technique, i.e., cooling the air below the dew point or whether air dehumidification is realized by a desiccant process. In the latter case the temperature of chilled water - if needed at all - can be higher since it has to cover only sensible loads. Application of desiccant technique in

extreme climates, i.e., climatic conditions with high humidity values of the ambient air, special configurations of the desiccant cycle are necessary in order to be able to employ this technology.

Short cuts: DEC = desiccant cooling; AHU = air handling unit.

6 Primary energy saving potential of solar assisted air-conditioning systems

With respect on environmental issues, some pre-conditions in the planning and sizing phase may be derived from simple rules and calculations. It is evident that solar assisted air-conditioning systems should be designed in order to save primary energy consumption, compared to a conventional system solution. For this reason a basic energy balance can help to assess the energy saving potential.

The specific primary energy consumption of a solar/fossil fueled hybrid system is defined by expressing the amount of consumed primary energy per produced kWh cold:

$$PE_{spec,\,sol} = \frac{(1 - SF_{cool})}{e_{fossil} \cdot COP_{thermal}} + PE_{aux} \quad ,$$

with SF_{cool} being the solar fraction (1 = total coverage of the cooling load by the solar system, 0 = no contribution of the solar system to the cooling load coverage); ε_{fossil} being primary energy conversion factor of a burner using fossil fuels, and PE_{aux} denotes the specific primary energy consumption of auxiliary components like for example a cooling tower and other fluid pumps.

Figure 6.1 shows as an example the specific primary energy consumption as a function of the solar fraction. The different slopes of the curves reflect different average $COP_{thermal}$ values of the process from 0.6 to 1.2. In this calculation, a primary energy conversion factor for heat from fossil fuels of 0.9 (kWh of heat per kWh of primary energy) is assumed. Furthermore, a primary energy conversion factor for electricity of 0.36 (kWh of electricity per kWh of primary energy) is applied. Additionally, the figure shows two horizontal lines: these are the specific primary energy consumptions of conventional electrically driven compression chillers with a) a COP_{conv} of 2.5, and b) a COP_{conv} of 4.5, whereas the latter represents an high efficient large size compression chiller.

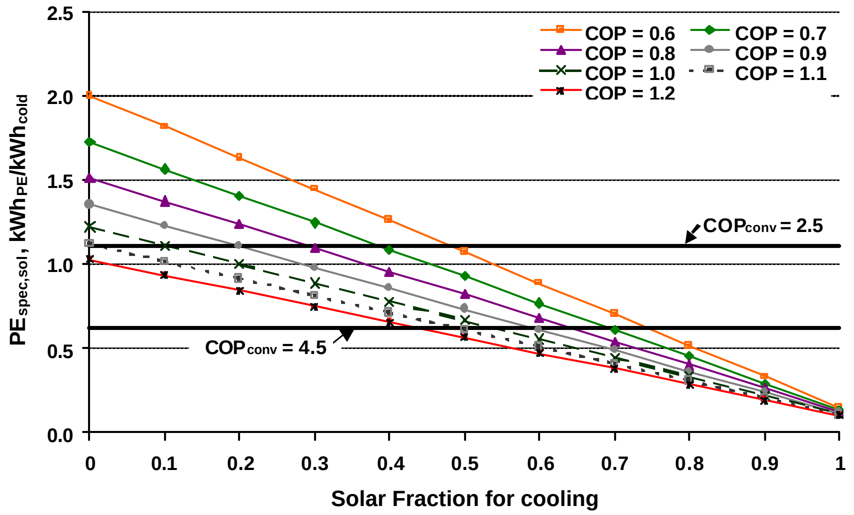

Figure 6.1. Specific Primary Energy consumption of solar assisted cooling systems as a function of the solar fraction for different values of $COP_{thermal}$. Additionally, the primary energy consumption for conventional eletrically driven compression chillers with two different values of COP_{conv} is shown (horizontal lines).

Assuming for example a thermally driven chiller system with an average $COP_{thermal}$ of 0.6, the system has to be designed for a solar fraction of approx. 0.5, to save primary energy compared to a conventional chiller system with a COP_{conv} of 2.5. If the conventional chiller works with a high COP_{conv} of 4.5, primary energy savings are achieved at solar fraction of > 0.7. The following conclusions can be drawn from this considerations:

- A thermally driven cooling system with a comparatively low COP and a fossil fuel heat source as a backup, requires a high solar fraction in order to achieve significant primary energy savings. This has to be guaranteed by a properly design of the system, e.g. a sufficient large solar collector arera, sufficient large storages and other measures in order to maximise the use of solar heat.
- Alternatively, a conventional chiller as backup system may be used. In this concept, each unit of cold provided by the solar thermally dr iven chiller reduces the cold to be delivered by the conventional unit. This design allows some primary energy savings even at low values of solar fraction. The solar system then serves mainly to reduce the electrical energy consumption.
- When a heat backup using fuels is applied, any replacement of fossil fuels by fuels from renewable sources such as biomass will increase the fossil fuel conversion factor ($\varepsilon_{fossil} \rightarrow \infty$ in case of 100% biomass fuel, only PE_{aux} is then contributing to the specific primary energy consumption) and thus decreases the primary energy consumption of the thermally driven system.
- Solar thermally autonomous systems do not require any other cold source and therefore always work at the limit with a 100% solar fraction.
- Systems with a thermally driven chiller with a high $COP_{thermal}$ may be designed with a smaller solar fraction even if a fossil fuel heat backup source is applied. The reason is that the heat from the fossil fuel burner is also converted at a high $COP_{thermal}$, competitive with a conventional system from a primary energy point of view.
- In any case, the use of the solar collector should be maximised by supplying heat also to other loads such as the building heating system and/or hot water production.

Additional recommendations on the design of solar assisted air conditioning systems may be found in the Guidelines for planners, installers and other professionals at the homepage of the EU project SACE – Solar Air Conditioning in Europe (http://www.ocp.tudelft.nl/ev/res/sace.htm).

7 Economic aspects of solar assisted air-conditioning systems

Most of todays realized solar air conditioning projects are of research or demonstration nature and still a lot of additional design and planning effort is necessary in the implementation phase of such a project. This, and the production of particular components currently below the level of industrial large series manufacturing, causes investment cost clearly above the investment cost of a conventional system solution, although the solar driven cooling system can support environmental protection by saving primary energy considerably and thus contributes to the goals in reducing greenhouse relevant emissions.

Figure 7.1 shows as an example the investment costs for different chiller types as a function of the installed chilling power. For desiccant cooling systems using a sorption wheel for dehumidification, average investment cost range from 5 € to 10 € per m^3/h of nominal supply air-flow (without solar thermal system). The price strongly depends on the nominal air-flow rate of the system which is directly proportional to the cross-sectional area of the wheel.

For an estimation of the total costs, the running costs of the entire system have to be included into the calculation, to determine the annual costs. Within the EU project SACE – Solar Air Conditioning in Europe[1] -, the energy-cost performance of solar assisted cooling systems was analysed to identify

- the performance of most relevant solar air conditioning technologies with respect to different load structures and at different European sites;
- configurations, leading to a minimum in cost with regard to environmental aspects;
- economic and technical conditions, to increase the energy-cost performance of the systems.

All solar assisted systems have been compared to defined reference systems, which are the conventional solutions without any solar assistance.

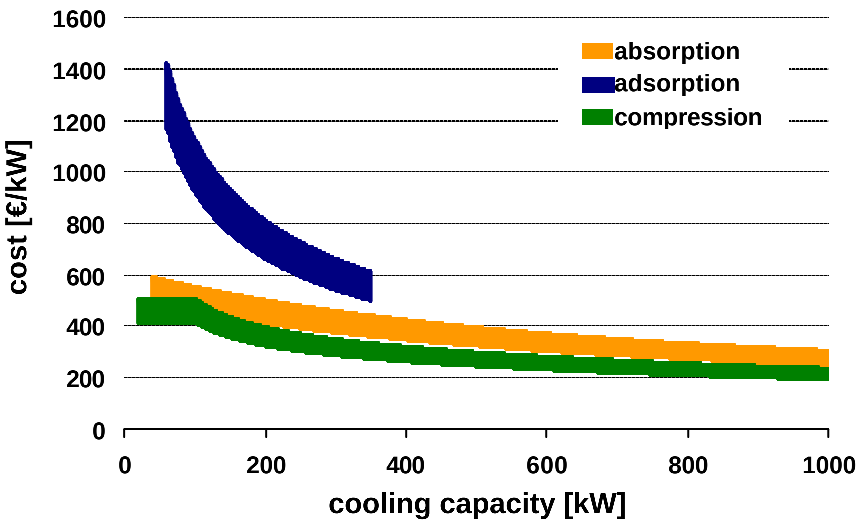

Figure 7.1. Specific cost ranges for different chiller types as a function of the cooling power (cost figures include heat rejection device, i.e., air cooling or cooling tower, but does not include installation cost).

The European sites Madrid, Athens, Palermo, Perpignan and Freiburg were selected to include different meteorological areas from moderate continental climate to mediterranean humid climate into the survey. Three typical load structures (lecture room, office building and hotel) were applied and beside different technical approaches in the configuration of the cooling system, different types of solar collectors were investigated. The key figures, reported in the study, are:

- Annual cost, compared to the annual cost of a conventional designed system without solar assistance;
- Saved primary energy;
- Cost of saved primary energy;
- Net collector efficiency.

The results of the survey can be summarised as follows:

- The potential in saving primary energy is high for solar assisted air conditioning systems using thermally driven chillers (up to 50% using high efficient solar collectors) and moderate for desiccant cooling systems (up to 30%)

1 Project web page: http://www.ocp.tudelft.nl/ev/res/sace.htm

- The annual cost are in general distincty higher for systems with thermally driven chillers, compared to conventional systems using an electric driven vapour compression chiller (up to 190% of the annual cost of the reference system, depending on the load structure, on the site and applied technology)
- The annual cost are in general moderately higher for desiccant cooling systems, compared to conventional air handling systems with an electric driven vapour compression chiller (up to 115% of the annual cost of the reference system)
- Collector type:
  in desiccant cooling systems are mostly common flat plate collectors sufficient;
  in systems using thermally driven chillers, the most promising collector type depends on the load pattern and on the chiller type. Often, evacuated tube collectors lead to a slight increase in annual cost only, but with high positive effect in primary energy saving and on the net collector efficiency.
- Backup system:
  in most considered systems with thermally driven chillers, an electric compression chiller as backup chilling device system leads to lower cost than a heat beckup;
  in desiccant cooling systems, a heat backup is more advantegous
- The use of the solar collector system in both, the cooling and heating system is mandatory in order to maximise its use and thereby improving the economic conditions

It has to be mentioned that a survey of this nature can not substitute a detailed analysis of a particular system under consideration, since in the study necessarily ‘flat rates’ for installation costs, system control etc. have been applied which do not match the real cost of a specific system and are subject to large variations. Furthermore, a solar autonomous operation of the cooling system was not investigated. Thus, the study presents very general the actual trends in the energy-cost performance of solar assisted air conditioning.

It is expected that with a moderate decrease in component cost of desiccant cooling systems (nearly within the range of negotiations with distributors and within the uncertainty of the survey), these type of solar assisted air conditioning systems may be already cost competitive to conventional solutions in some applications.

For systems using thermally driven chillers, more actions are necessary to improve the cost performance. Although large future cost reductions of the adsorption chillers and of evacuated tube collectors are expected, additional efforts in an increase of the technical performance (COP) of the chillers are required. A raised experience of manufacturers, planners and installers of these type of systems may additionally result in a decrease of planning, installation and control costs. With these measures, the systems may achieve step by step a cost range close to conventional reference systems, but always saving considerable amounts of primary energy and thus avoiding environmentally hazardous emissions.

8 Thermal collectors for solar assisted air-conditioning systems

In solar assisted air conditioning systems, the difference in the operation of the solar collectors compared to solar thermal collector systems for hot water production is the high temperature level, at which the useful heat has to be provided. For thermally driven chillers, the driving temperature is mainly above 80°C, lowest values are 60°C. For desiccant cooling systems, the driving temperature is above 55°C up to 90°C. Due to the high volume flow rates in the heat supply cycle, an ideal stratification in the hot water storage is difficult and the return temperature to the solar collector is relatively high as well. This causes some restrictions in the selection of the collector type.

Figure 8.1 shows typical efficiency curves for different solar thermal collectors. Consequently, standard flat-plate collectors and solar air collectors may be implemented with most benefit in solar assisted desiccant systems. In configurations using an adsorption chiller or a single-effect absorption chiller, the use of selectively coated flat-plate collectors is limited to areas with high irradiation availability. For other areas and for chillers requiring higher driving temperatures, high efficient collectors are to be implemented, e.g. evacuated tube collectors. Highest temperatures may be achieved with fixed mounted evacuated tube collectors using optical concentration or even with tracking collectors using high optical concentration at regions with sufficiently high amounts of solar beam radiation. This is an interesting option for solar assisted air conditioning system using high efficient absorption chillers (2-effect) or new technologies such as steam jet cycle chillers.

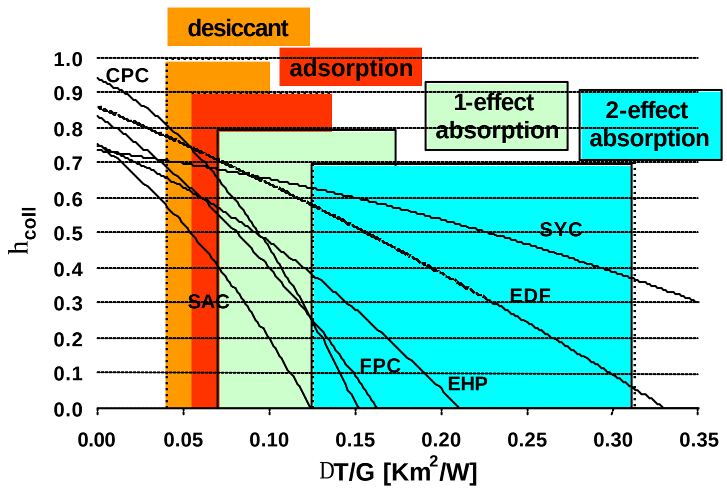

Figure 8.1. Typical efficiency curves of different solar thermal collectors as a function of the difference in the operation temperature of the collector and the ambient temperature, ΔT, divided by the radiation in collector plane, G.
Meaning of the abbreviations: SAC = solar air collector, FPC = flat plate collector, CPC = stationary compound parabolic collector, EHP = evacuated tube collector with heat pipe, EDF = evacuated tube collector with direct flow, SYC = concentrating evacuated tube collector.
The marked areas characterize the typical operation area of the different thermally driven cooling / air conditioning processes. Whereas all collectors are applicable for dessicant cooling systems (but economically senseless for expensive high efficient collectors), only direct-flow evacuated tube collectors and concentrating systems can be combined with 2-effect absorption chillers.