Adriacold

"Diffusion of Cooling and Refreshing Technologies using the Solar Energy Resource in the Adriatic Regions"

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Hands-on guidelines useful to help decision makers to adopt the solar cooling system

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1. Introduction

Increase of energy efficiency along with harnessing renewable energy play the most important role in achieving the goal of significant reduction of fossil fuel energy consumption in buildings in near future. Achieving a ‘nearly zero energy’ standard in all new buildings by end of 2020, as prescribed in the Energy Performance of Building Directive, would open possibility to cover minimized energy demand by renewables, especially by solar energy. In this regard, solar energy can be employed not only in active solar systems for heating and domestic hot water preparation, but also for cooling in thermally driven cooling systems during summer time. Such combined use of solar energy for heating and cooling has increased economic viability relative to the otherwise separate systems solutions. Nowadays, solar air conditioning market has a remarkable growth due to increased living standards and occupant comfort demands as well as due to trends in building architecture towards more extensive use of transparent surfaces in the building envelope, which is all accompanied with higher cooling loads. Increased summer cooling demand is in phase with solar radiation seasonal variation, which makes use of solar energy in thermally-driven chillers technically desirable solution. Today, only about 70 solar cooling systems are installed within Europe, which indicates that this technology is still technically and commercially developing. More solutions are available at larger capacities >50 kW, where different types of thermally driven cooling devices are available in the market and can be driven by solar collectors. Besides high investment costs, the main barriers for wider implementation, are the lack of practical experience in design, installation, operation, control, commissioning and maintenance.

On the other hand, small scale systems have not been available on the market for many years. In past few years, first commercial systems with capacity range 5 - 50kW appeared on the market. Further development of small capacity cooling and air conditioning systems and increased market share is expected in a close future. Development of solar cooling systems components and systems started more than 30 years ago, but many of the activities stopped due to lack of financial support. Activities in this field are restarted research and demonstration projects in this field were reactivated several years ago thorough co-operative projects such as Solar Heating and Cooling Programmed of the International Energy Agency (IEA). Yet, there is a lack of standardized design guidelines and lack of common practices for design and construction. Experience from realized projects and collected field data under real operating conditions indicated problems with the system’s hydraulic design, and with the controls. Also, the calculated energy performances and savings were proved not to be realistic.

Solar cooling systems require more engagement at design stage compared to a conventional system. Design process should include simulations of several different system configurations within a whole year in order to find an optimum solution from energy and cost perspective. Increase of economic feasibility highly depends on performance improvements of thermally driven chillers. Also, combination of small capacity chillers with existing solar combined systems for space heating and domestic hot water preparation gaining increased market shares is expected to open new market segments. Application of small thermally driven chiller in such system enables solar collectors to be fully exploited during summer days. Such systems are especially suitable for climates with mild winters and sunny summers such as in the Mediterranean zone.

This handbook aims at providing practical information about application of solar cooling systems in buildings. First part comprises basics on solar energy and its utilization in hot water systems for conversion into useful thermal energy, since solar cooling systems can also be used for domestic hot water and space heating purposes. Second part deals with thermally driven cooling systems including all auxiliary components and design guidelines based on knowledge and experiences gained from design and monitoring of installed systems.

The main purpose of this handbook is to provide investors, designers and potential users with practical and reliable information on suitability of solar cooling systems for particular application during the decision-making and the design processes as well to promote usage of such systems which can significantly reduce primary energy consumption in buildings.

2. Solar radiation

2.1 General characteristics

The sun is the ultimate source of the most of useful renewable energy on the earth, Fig. 2.1.

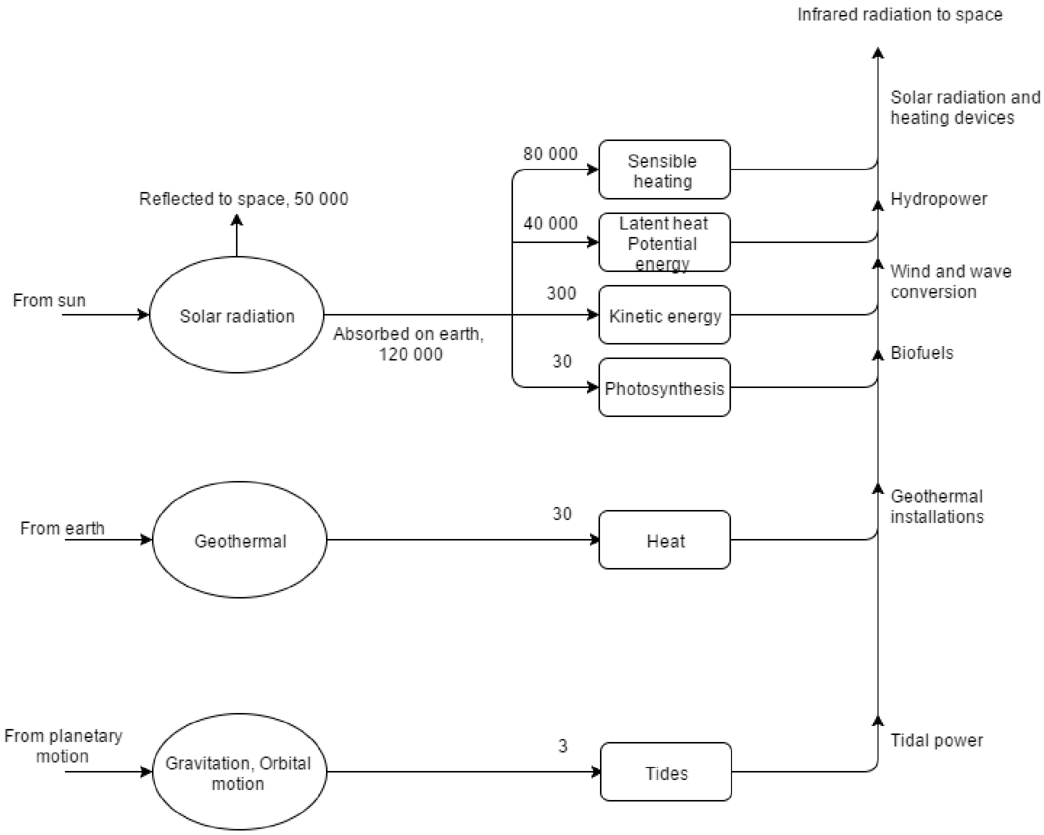


Figure 2.1 Natural energy currents continuously passing through the earth, dominance of solar radiation, (units terawatts – 10^{12} W) [1].

The energy produced by means of a nuclear fusion on the sun ($380000 \cdot 10^9$ TW) is radiated from its surface (photosphere temp. 5777 K) to the earth (170000 TW) in the form of electromagnetic waves over the wave length band $\lambda=(0.3 \div 2.5)\mu m$ - short wave radiation. The amount of solar energy absorbed on the earth annually (120000 TW) is equivalent to 6000 times the world's annual primary energy consumption (19 TW). However, solar energy is distributed over the whole earth's surface which results in a relatively small energy flux (< 1 kW/m^2) when compared to fossil fuel appliances (e.g. in gas furnaces the heat can be transferred at 100 kW/m^2). The solar constant equals 1367 W/m^2 (±1.5% measurement uncertainty) and represents the solar radiation power that reaches the earth's atmosphere, referred to the surface unit area perpendicular to the rays. 30% of the incoming radiation is reflected immediately back to the space, mostly from clouds, while a small proportion from the earth's surface (especially snow and ice). The remaining 70% (up to ~1 kW/m^2) is partially absorbed in the atmosphere (O_3, H_2O, CO2) causing heating there, while the remaining part proceeds to the surface either as a direct (beam) radiation or in the form of diffuse radiation (Fig. 2.2) which is a result of scattering on clouds and/or by air molecules (Rayleigh scattering), water and dust (Fig. 2.3). In winter months diffuse component account in average for 60% of total radiation for the area of Zagreb. In summer months this percentage is 30%.

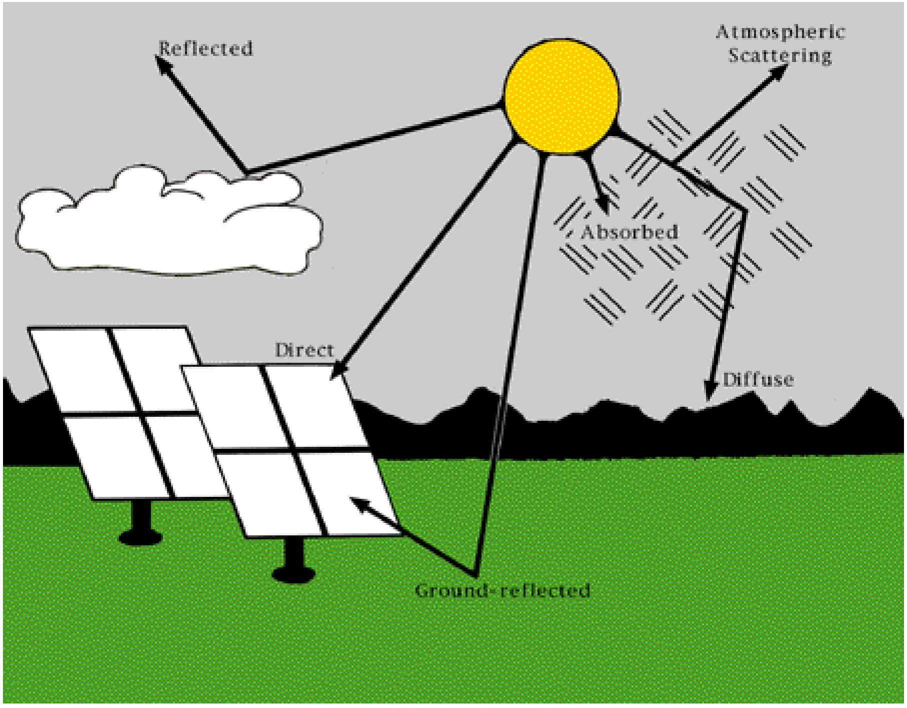


Figure 2.2 Origin of a direct beam and diffuse radiation [2].

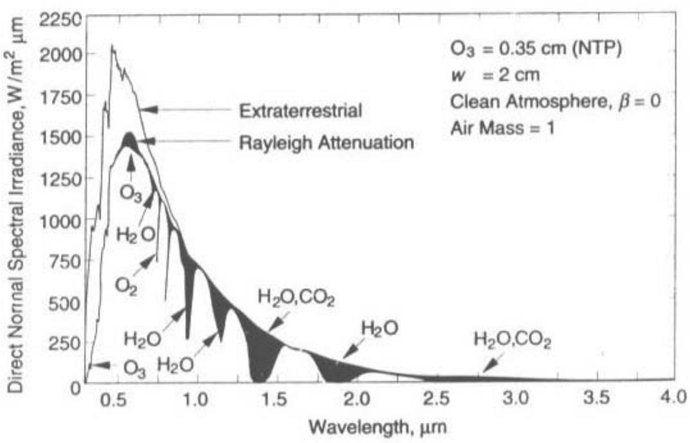


Figure 2.3 Effects of Rayleigh scattering and atmospheric absorption on the spectral distribution of beam irradiance [1].

One part of the solar energy that reaches the earth's surface is converted to warm up the air, soil and oceans while the rest is used for evaporation of the water. A relatively small part (<1%) is both absorbed by plant in a photosynthesis and used to drive winds and waves. All solar energy received on the earth's system is ultimately radiated to the space (-273°C), both from the earth's surface (av. temp. ~15°C) and from the upper layers of atmosphere (the effective atmospheric temp. appearing to the space is about -20°C) over the wave length band λ=(5÷25) μm - long wave (infra red) radiation, Fig. 2.4.

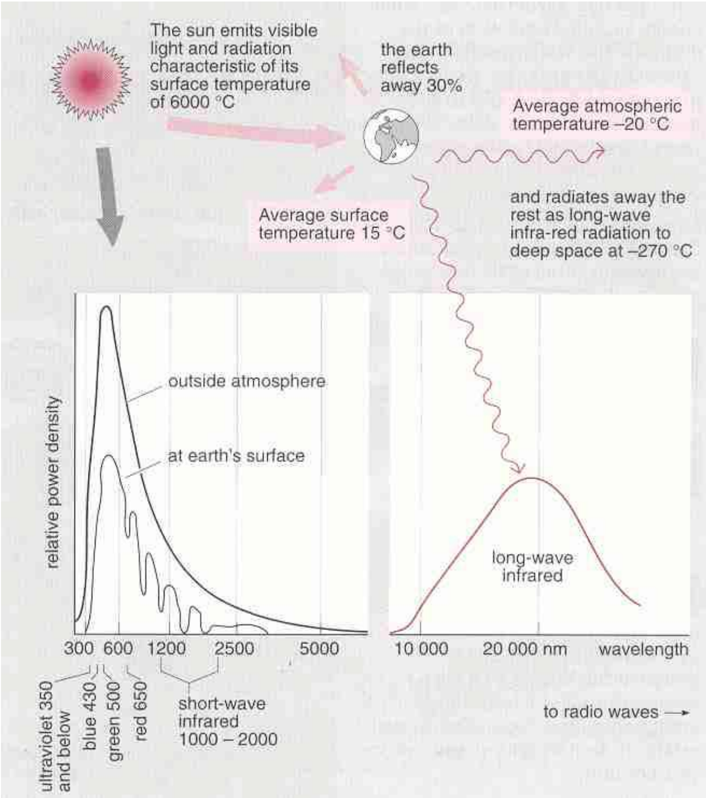


Figure 2.4 Radiation to and from the earth [3].

Most of the long wave radiation emitted by the earth's surface, atmosphere itself and clouds cannot pass directly through the atmosphere to the space as it is absorbed by the atmosphere gases such as CO_2, water vapour, methane and reemitted in all directions, including towards the earth's surface. This natural process is known as the greenhouse effect. Greenhouse gases act as natural insulators increasing, therefore, the overall heat transfer resistance from the earth system to the space. Consequently, the temperature of earth's surface and atmosphere is also increased provided in the thermal equilibrium all received solar energy by the earth's system is radiated back to the space. In the complete absence of the greenhouse gases the earth's surface average temperature would be -

20°C instead of 15°C. Nowadays present increase in the concentration of greenhouse gases in the atmosphere, mainly due to an extensive use of fossil fuels, is expected to cause the additional rise of the earth's surface and atmosphere temperature.

2.2 Seasonal variation

A seasonal variation of solar radiation intensity occurs due to the change of the earth's axis angle between the equatorial plane and sun rays direction, whereas the sun rays strike less or more perpendicularly particular surface area on the earth, which influence the amount of delivered energy (Figs. 2.5, 2.6). Furthermore, a consequent increase or decrease of the solar rays length path through the atmosphere influences the amount of scattered, reflected and absorbed radiation in the atmosphere and therefore the total energy flux incident on the earth's surface. The declination δ provides a convenient measure of the seasonal changes and varies smoothly from δ=-23.5° to δ=23.5°.

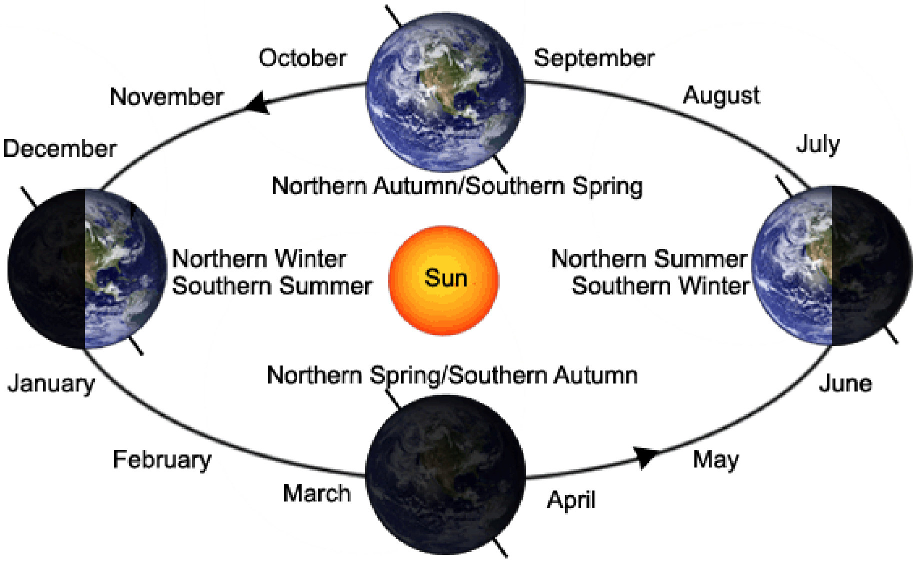


Figure 2.5 The tilt of the earth's axis during revolution around the sun [4].

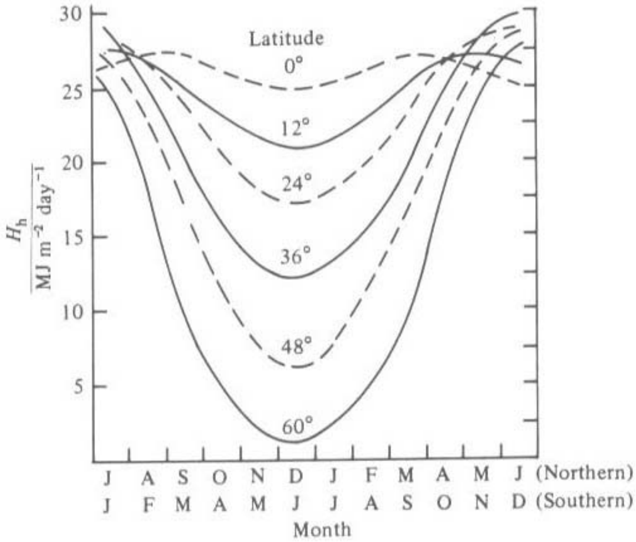
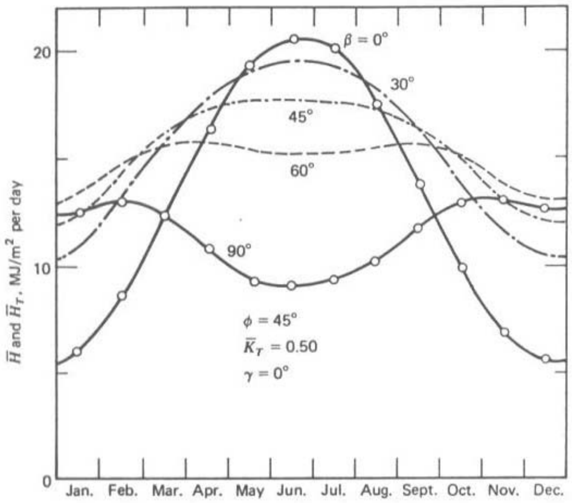


Figure 2.6 Seasonal variation of solar energy received on the horizontal surface on a clear day in dependence of the latitude [1].

The annual amount of solar energy received on a horizontal surface in Zagreb is 1200 kWh/m^2 while in Split 1600 kWh/m^2. 75% of this energy is received in the warmer half of a year i.e. from April to November. The diagram in Figure 2.6 shows that the solar energy received for the latitude of 45° (Croatia) in January is 5 times as low as in June. Influence of surface tilt β on received solar radiation can be seen from Figure 2.7.



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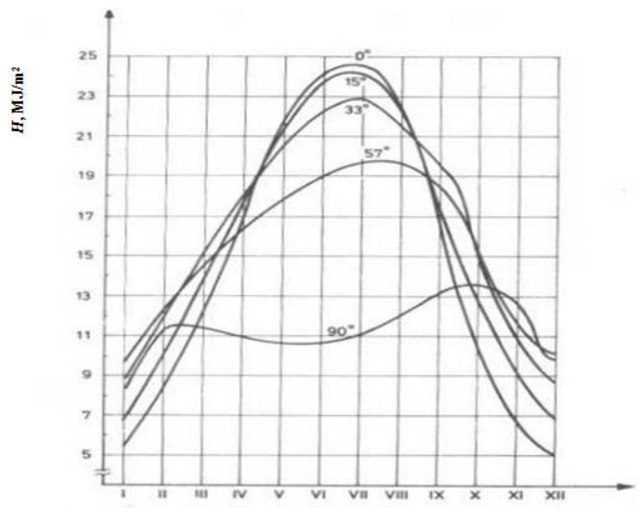


b)

Figure 2.7 Influence of surface tilt β on received solar radiation for a latitude of 45° and $\overline{K_T}$ =0.5, azimuth angle 0° a) calculated values [5], b) measured values, Croatia [6].

From the diagrams on Figure 2.8 it can be seen that the influence of the azimuth angle i.e. deviation from the south on received solar energy is lower than the influence of the slope β. No significant decrease of the received solar radiation on collector plane is exhibited if the deviation from the south is kept within 30°.

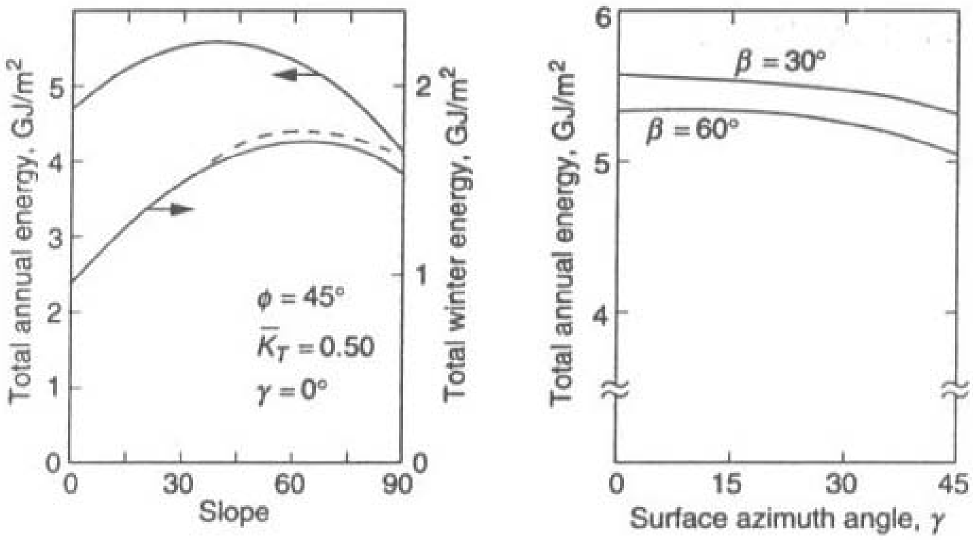
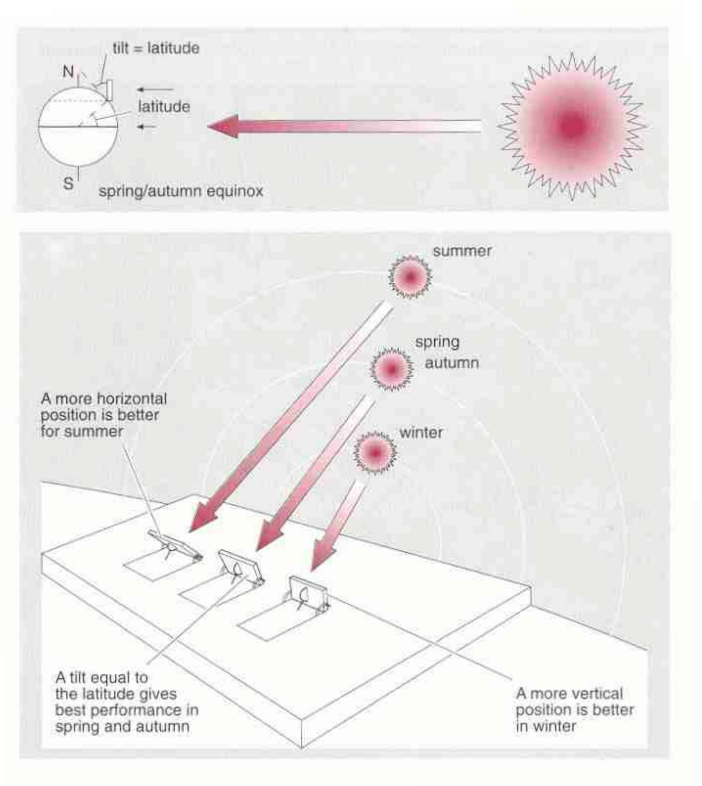
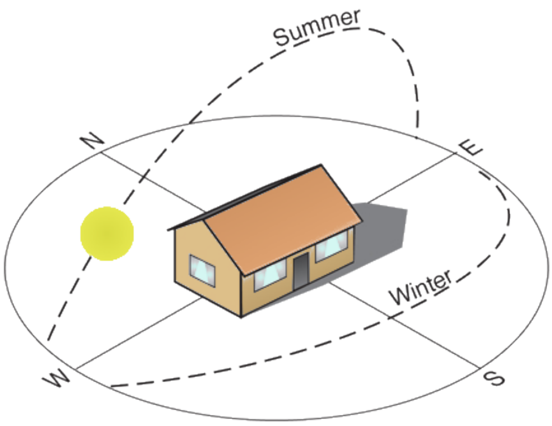


Figure 2.8 Variation of total annual solar energy received on sloped surface in dependence on β and azimuth angle γ for a latitude 45°, $\overline{K_T}$ =0.5 [5].

2.3 Optimum collector tilt angle

Solar collector should be mounted under the tilt to the horizontal which in a period of use (a year, summer months) enables the highest amount of solar radiation received. Optimal tilt angle depends on latitude, considered period of a year (Fig. 2.9), and on an application of the solar system. Normally, optimal tilt angel is close to the value of altitude, hence it is e.g. 37° for Croatia (av. latitude cca 45°). If solar system application is such that as much as possible solar energy is to be collected in summer months (e.g. apartments, hotels in tourist season) then the optimal tilt angle is (10÷20) °. If solar system is also intended to cover part of heating demand in winter months, then the optimal angle can be >45°.





a) **b)**

Figure 2.9 a) Optimal collector tilt angle in different seasons [3], b) relative Sun orbit seen from the Earth within seasons [7].

Optimal tilt angles for Croatia in different months are provided in Fig. 2.10. When the tilt angle is adjusted every month to its optimal value then the yearly insolation is 1900 kWh/m^2, while tilt angle of 45° would gain 1800 kWh/m^2 i.e. 6% less. This small difference is due to the influence of diffuse component that is less sensitive to the tilt change than the direct component.

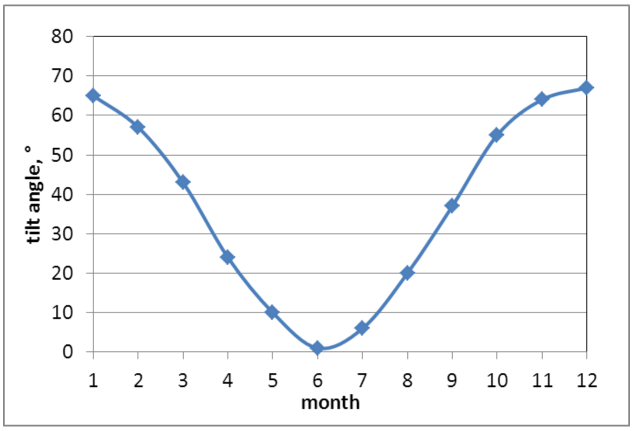


Figure 2.10 Optimum monthly values of collector tilt angle β within a year in Split, Croatia [6]

3. Solar collectors

3.1 Flat type collectors

The most common utilization of solar energy is water heating for domestic water purposes and for low temperature space heating. Besides solar cooling, other uses of solar energy comprise direct air heating, swimming pool heating, electricity production by photovoltaics and concentrating collectors, crop drying, desalination. The separate category is passive solar systems intended for space heating. In this Chapter only those types of solar collectors will be dealt with that can be used in solar cooling systems. Active solar hot water systems, which dominate on solar market, basically consist of a flat type solar collector, hot water storage tank and differential control, Fig. 3.1.

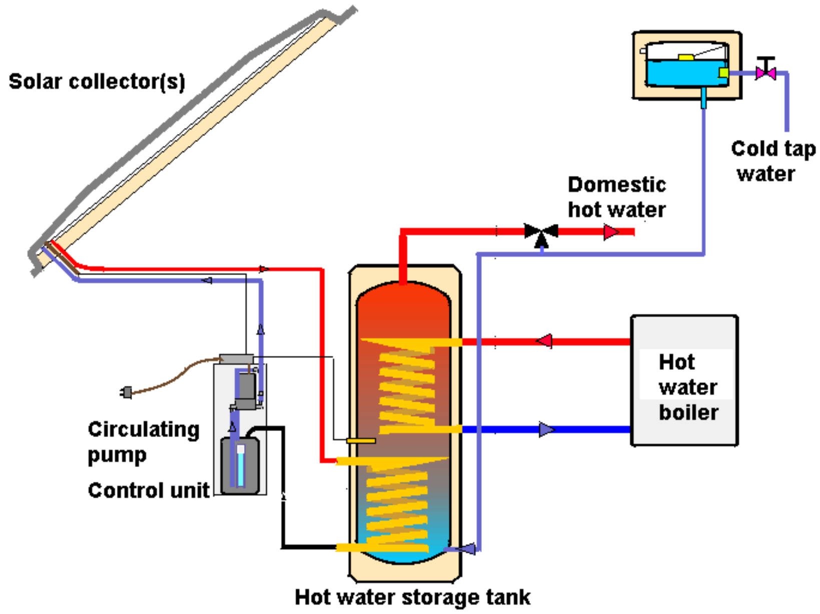
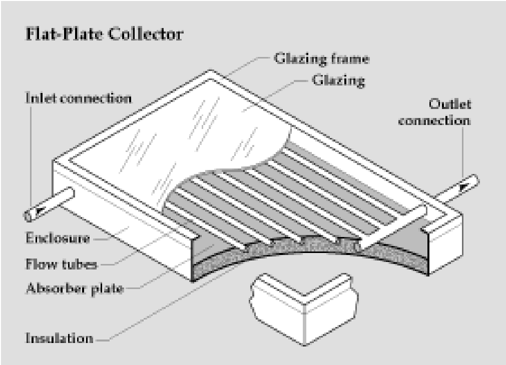


Figure 3.1 Solar water heating system.

The essential part of the solar system, flat solar collector, was first patented in 1909 in California, and became very popular during the oil crisis in 1970s. Since then, it has been significantly improved in terms of its thermal efficiency thanks to development of new coatings and new technologies for attaching tubes to absorber plate. All that led to a larger number of solar hot water system installations in 1990s and consequent lowering of the system components prices, which have been reduced up to 50% in past two decades. Solar collectors can be found in many forms, some of which are shown on Figs 3.2 and 3.3.



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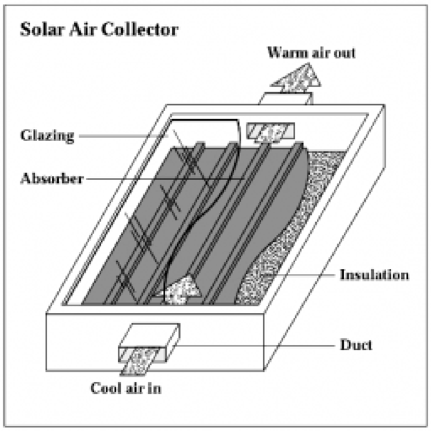

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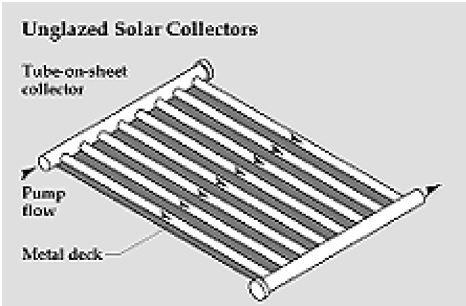


c)

Figure 3.2 Different types of plate solar collectors: a) flat plate (water) collector [8], b) flat plate (air) collector [9], c) unglazed solar collector [10].

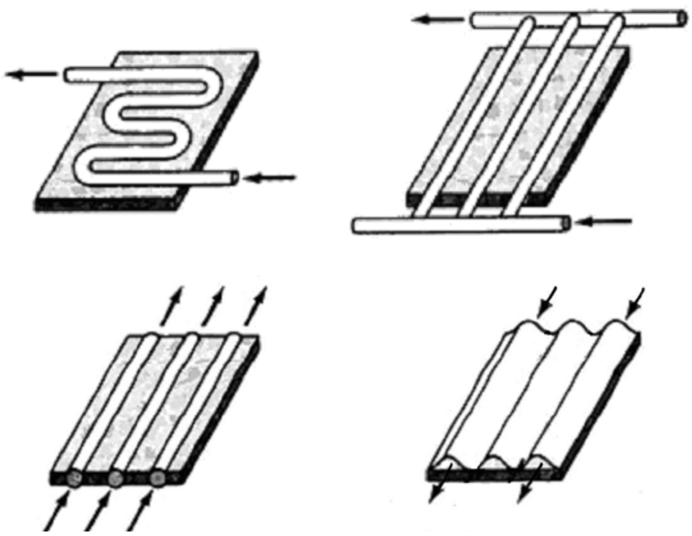
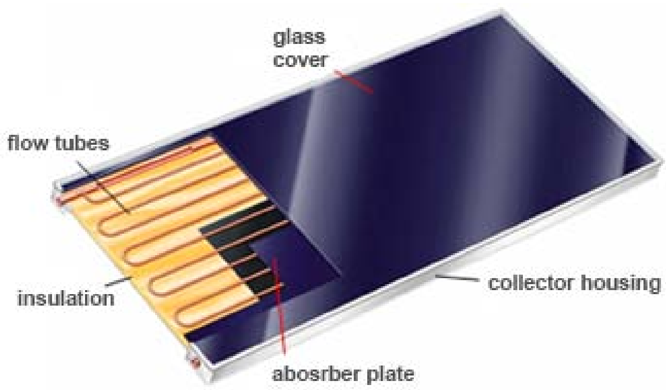


Figure 3.3 Some models of plate type solar collectors [11]

The prevailing form on the market is a single glazed plate solar collector (Fig. 3.4) consisting of absorber plate with attached tube sheet, glass cover and back insulation, all fitted in (aluminium) casing.



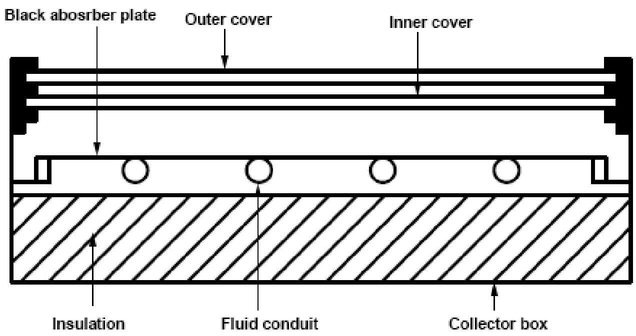


Figure 3.4 Flat type solar collector with single or double glass cover

The absorber plate is normally coated with a selective coating which has high absorptance properties for short wave radiation (a =0.9 - 0.96) and low absorptance i.e. emittance ε =0.06 - 0.2) for long wave (infra red) radiation. Similar characteristics possesses the glass cover in terms of its transmittance at short waves (τ =0.9 - 0.95) and long waves (τ < 0.02). All that increases the ultimately absorbed portion of the incident solar radiation and decreases radiant heat losses which occur due to the high absorber plate operating temperatures. Back insulation (mineral wool, PU foam) is normally 30 - 50 mm thick and it can also be combined with a side insulation to further reduce the heat loses (which are normally <5% of the total losses). The tube sheet can be arranged in a number of parallel running tubes with diameters ranging from 8 to 12 mm, with two somewhat larger distributive and collecting tubes (ϕ 12 - 14 mm), or in the form of a serpentine as shown on Fig. 3.3. The parallel tube arrangement is intended to sunnier and warmer climates (e.g. Mediterranean countries) since the temperature rise from the collector inlet to outlet is still high enough to allow for use of a moderately sized heat exchangers in the storage tank. The serpentine arrangement is more appropriate for middle to colder climates (such is mid-west and northern Europe) as it enables sufficiently high fluid outlet temperatures to be obtained per fluid pass through the collector at lower solar radiation intensities, indeed, with the lower mass flow rate.

3.2 Collector efficiency

The collector efficiency is defined as a ratio of the useful energy gain to the maximum available solar energy from the incident radiation

$$\eta_c = \frac{P_{coll}}{G \cdot A_{coll}}$$

The collector efficiency is directly linked to the collector heat losses that occur due to difference between the fluid i.e. absorber temperature and the temperature of surrounding air and objects. The higher the fluid temperature the lower is the efficiency. Also the lower is ambient temperature and solar radiation the efficiency is also lower (typically in winter periods).

A convenient way of presenting the collector efficiency is to relate it both to the fluid /air temperature difference and also to the solar radiation, as shown on a diagram, Fig. 3.5. Such plots are based on test data and enable an accurate calculation of the collector efficiency at arbitrary weather (G, T_{air}) and working (T_f) conditions.

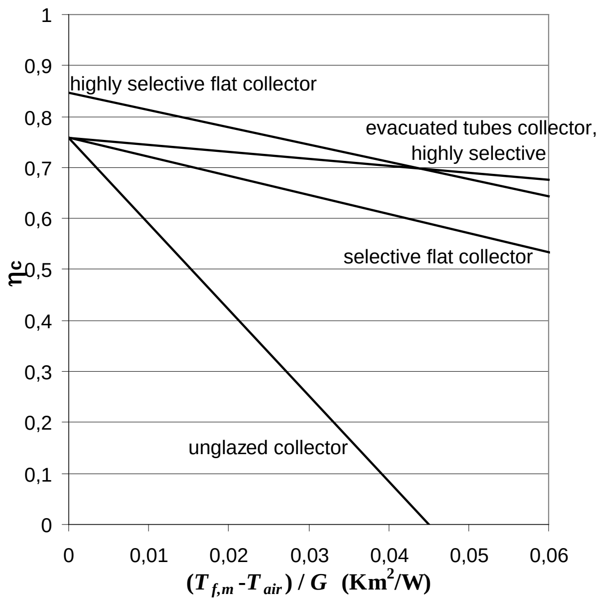


Figure 3.5 Measured collector efficiency as a function of fluid/air temperature difference and incident solar radiation for different types of solar collectors [12].

Hot water collector thermal performances expressed in terms of the collector efficiency η_c are determined according to the standard EN 12975-2. The results can be presented in the form of a second order polynomial

$$\eta_c = \eta_o - a_1 \frac{(T_{f,m} - T_{air})}{G} - a_2 G \left[\frac{(T_{f,m} - T_{air})}{G} \right]^2$$

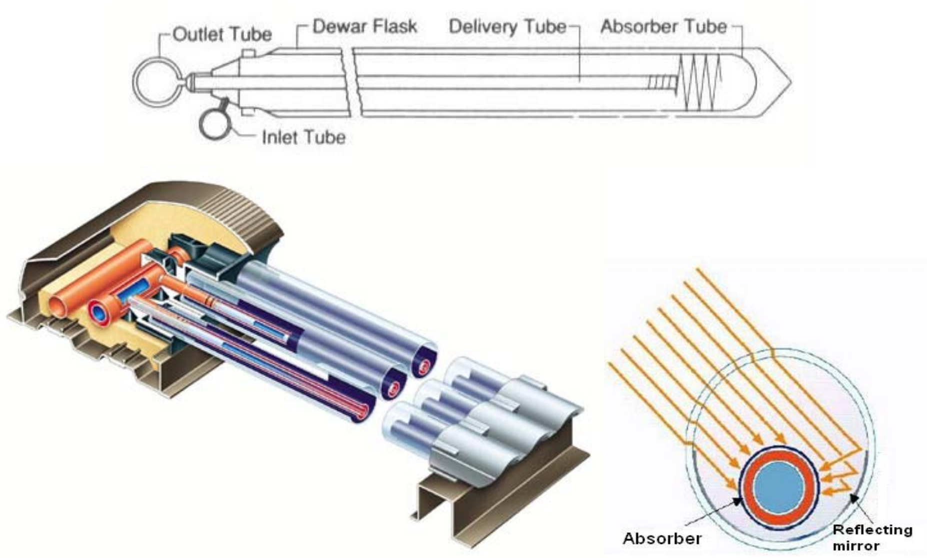
Knowing the collector efficiency from test data, the useful energy gain at arbitrary operating and weather conditions is simply obtained from

$$P_{coll} = \eta_c \cdot G \cdot A_{coll}$$

3.3 Evacuated collectors

Evacuated collectors are nowadays widely spread on a solar market as an alternative to plate collectors in areas with lower solar radiation and where higher fluid temperatures are needed, as case in solar cooling applications. The basic idea behind their introduction on the market was to reduce the convective heat losses between the absorber and cover by evacuating the space in between them.

In order to sustain the vacuum caused stress in a cover material, evacuated collectors consist of evacuated tubes (Dewar type) with, in some types, concentrating mirrors which focus both diffuse and direct radiation onto a selective absorber surface to which is attached an inlet and outlet tube. Main types are shown on Fig. 3.6. In one case (Fig 3.6a), the delivery tube is inserted in the coated absorber tube around which is the vacuum jacket. Inlet fluid comes from the delivery tube to the absorber tube space. Such construction with added reflecting mirrors is also available on the market. Another construction (Fig. 3.6b) has a fin absorber rolled and inserted into a vacuum tube. Its thermal characteristics are shown on Fig. 3.7 together with the characteristics of plate collectors and constructions without reflecting mirrors. In some models the absorber can be also found in a form of a flat plate fin (Fig. 3.6c). Similar to the model from Fig. 3.6a, the fluid is here also delivered through the inner tube inserted in a large diameter tube (’’tube-in-tube’’ arrangement) to which is attached the absorber plate.



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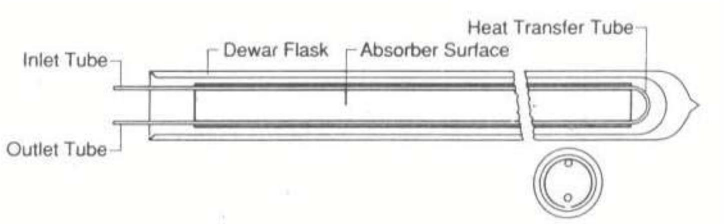

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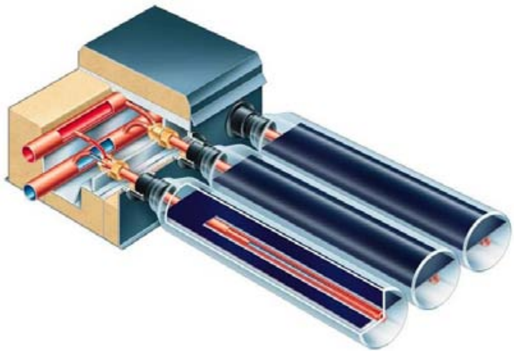
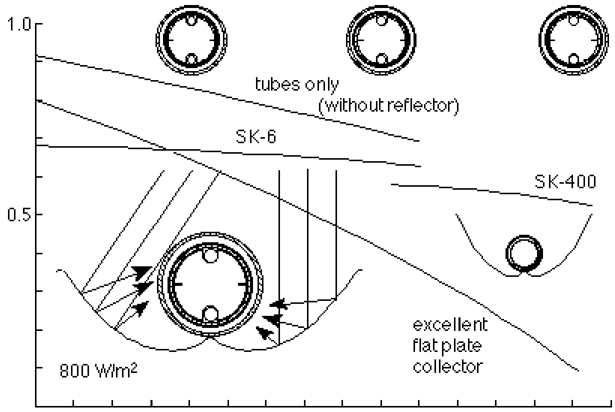
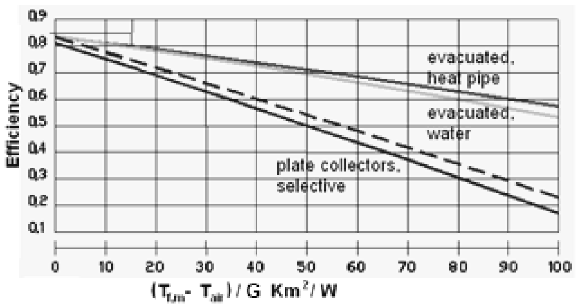


c)

Figure 3.6 a) Construction with absorber rolled around inner ’’tube-in-tube’’(with and without the reflecting mirror), b) Construction with U-tube and attached fin (without reflecting) mirror, c) Construction with plate fin attached to inner ’’tube-in- tube’’ [12].





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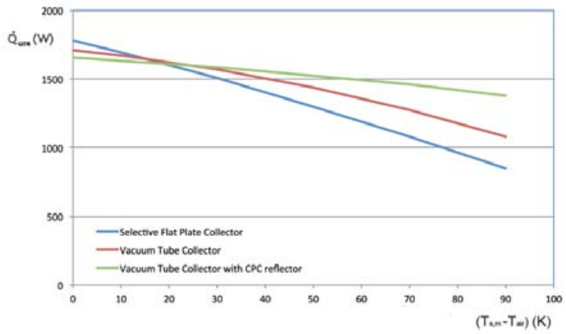
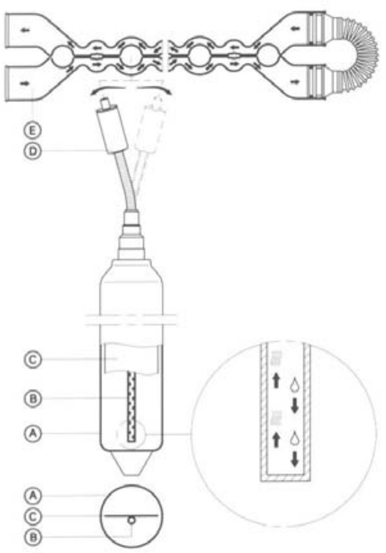
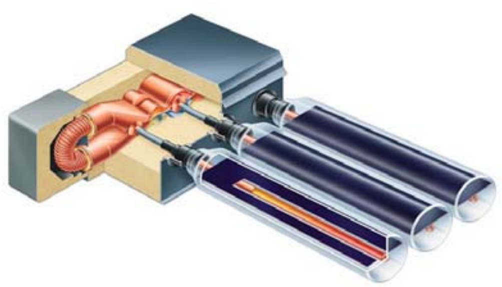


Figure 3.7 a) Thermal characteristics of Dewar tube with fin, b) comparison of heat pipe model with fin vs. standard evacuated tube collector with fin and water as a working fluid [12].

A special type of evacuated collector consists of only single tube acting both as an evaporator and condenser at the same time (heat pipe), whereby the fluid flows in two directions along the opposite sides of tube (Fig. 3.8).Typical working fluids used are alcohols or freons.





A- evacuated glass tube
B- heat pipe
C- absorber
D- condenser
E- coaxial heat exchanger (tube in tube)

Figure 3.8 collector with fin (without reflecting mirror) working as a heat pipe [12].

3.3.1 Evacuated flat plate collectors

Evacuated flat plate collectors are produced with special frames and covers permitting them to maintain a high vacuum level (10^{-3}Pa) despite a flat geometry, Fig. 3.9. Their performances are comparable to evacuated tube collectors. Stagnation temperatures can reach more than 300°C. Such a collector does not require a tracking system and can be mounted into a flat roof of e.g. industrial buildings for process heat. Some other commercial products are built with a lower level of vacuum (vacuum to be achieved on the roof after installation) at a lower cost.

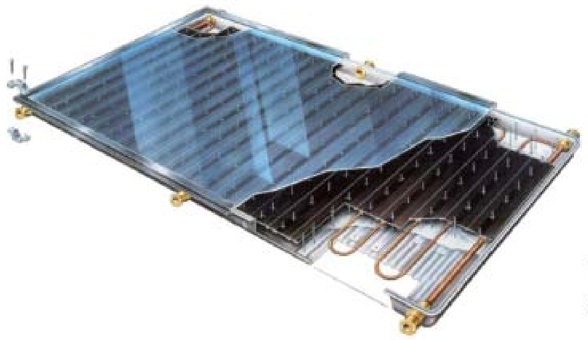


Figure 3.9 Evacuated flat plate collector [13].

As already mentioned, evacuated collectors have reduced the convective as well the overall heat losses compared to plate type collectors. On the other hand, due to the construction features (tubes instead of plate) the effective absorber area is relatively small compared to the collector projected (gross) area. Since the efficiency (and the price) is referred to the projected area, the evacuated collectors appear in the efficiency diagrams less effective than some plate collectors at some working regimes This in practice means that more roof area is needed for the same energy gain if evacuated collectors are used instead of plate ones at the identical absorber and glazing properties. Yet, the efficiency curves of evacuated collectors drop slower with the increased temperature difference collector fluid-ambient compared to plate types and lower solar irradiance, which means that at higher operating temperatures (>70°C) commonly encountered in solar cooling systems and/or in winter period of a year, evacuated collectors operate with higher efficiency and power than plate ones.

3.4 Concentrating collectors

There are a number of solar applications that require higher temperatures than those achievable by flat type solar collectors (up to 80°C). Typical application includes solar heat conversion to a mechanical work and/or electricity production but also recently use in double effect solar cooling systems operating at temperatures >130°C. Concentrating collectors (CPC) concentrate only beam solar radiation (which requires sun tracking) to the relatively small absorber (receiver) in order to reach the temperatures as high as 500°C - 2700°C. The concentration ratio R is a ratio of the mirror aperture area (opening to sun rays) to the absorber area, and it can be theoretically as high as R=45000, but in practice it never exceeds 10000 due to high precision needed in the manufacturing to obtain the high concentrations. Solar energy in such collectors is most commonly focused to the line (parabolic trough concentrators) or to point (parabolic bowl concentrators) (Figure 3.10). Parabolic trough concentrator is a parabolic mirror (aluminium or silver deposited on a glass or plastic, reflectance 90 - 94%) with the absorber running along its axis which is aligned east-west. The trough is then rotated around this axis to follow the sun elevation (zenith angle θ_z). The absorber is shielded from the above to decrease the radiation losses (although it protects the absorber from a direct solar radiation). The concentration ratio of trough concentrators goes usually up to R=50 which enable the temperature up to (400 - 500) °C to be reached. Such concentrators are often used for the electricity production in a large solar power stations (Fig. 3.11) where the collectors heat up working media (e.g. thermal oil), which is then used to produce a steam via heat exchanger and finally the electricity with the overall conversion efficiency of 14 - 22%.

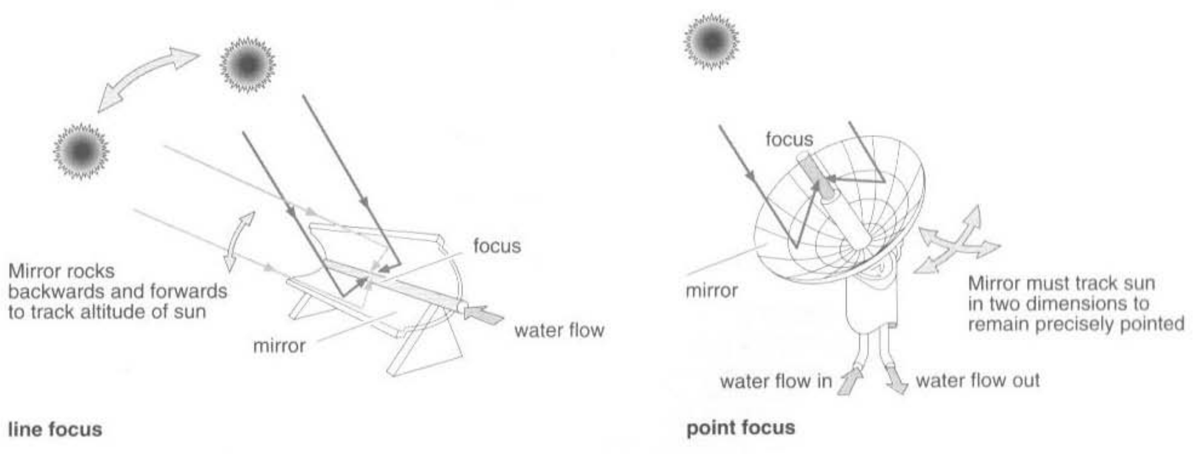


Figure 3.10 Solar concentrators [3].

Figure 3.11 Luz solar power station for electricity production, southern California [3].

Parabolic bowl concentrators track the sun in two dimensions-following the azimuth and zenith angle. As the concentrator fully tracks the sun and the concentration ratio can be as high as R=10000, the maximum achievable temperature is higher than in a trough concentrator and in practice goes up to 2700°C. The high temperatures allow direct production of a mechanical work. Figure 3.12 shows the Stirling engine that is placed right in the focus of parabolic bowl concentrator which run at temperatures (700 - 1000)°C producing electricity with the high average overall conversion efficiency of 30%.

Figure 3.12 Stirling engine driven by solar heat [14].

Stationary CPC collectors used for the heating of a liquid fluid at low (50 - 70 °C) and medium temperatures (80 - 110 °C) have two orthogonal axes of symmetry and are designed with acceptance angles greater than 30º to avoid the need for tracking the sun, Fig. 3.13. As a result they usually have concentration factors lower than 2. Practical design limitations, such as the height of the collector box, can also impose the use of truncated CPC's. This means that the upper part of the mirror is cut and sunrays with incident angles higher than the acceptance angle can still impinge directly on the absorber, but when they strike the mirror, they are reflected out of the collector.

An aspect of CPC's is that they collect beam, diffuse sky and ground-reflected solar radiation. This has to be taken into account when determining the collected solar energy for the different collector trough orientations, typically east-west for forced circulation systems or north-south for thermosiphon systems. The efficiency of stationary CPC collectors is determined using the same standards as for flat plate collectors. The thermal performance will depend on the collector's design and especially on the concentration ratio. Manufacturers of stationary CPC collectors use selective surfaces for the absorbers. As a result these collectors can have lower heat losses than flat-plate selective collectors, depending on the concentration ratio they achieve.

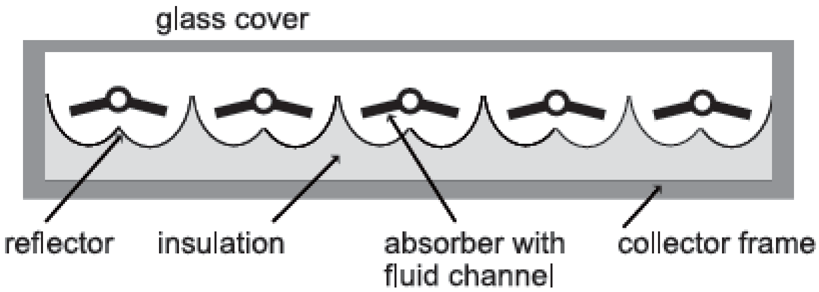
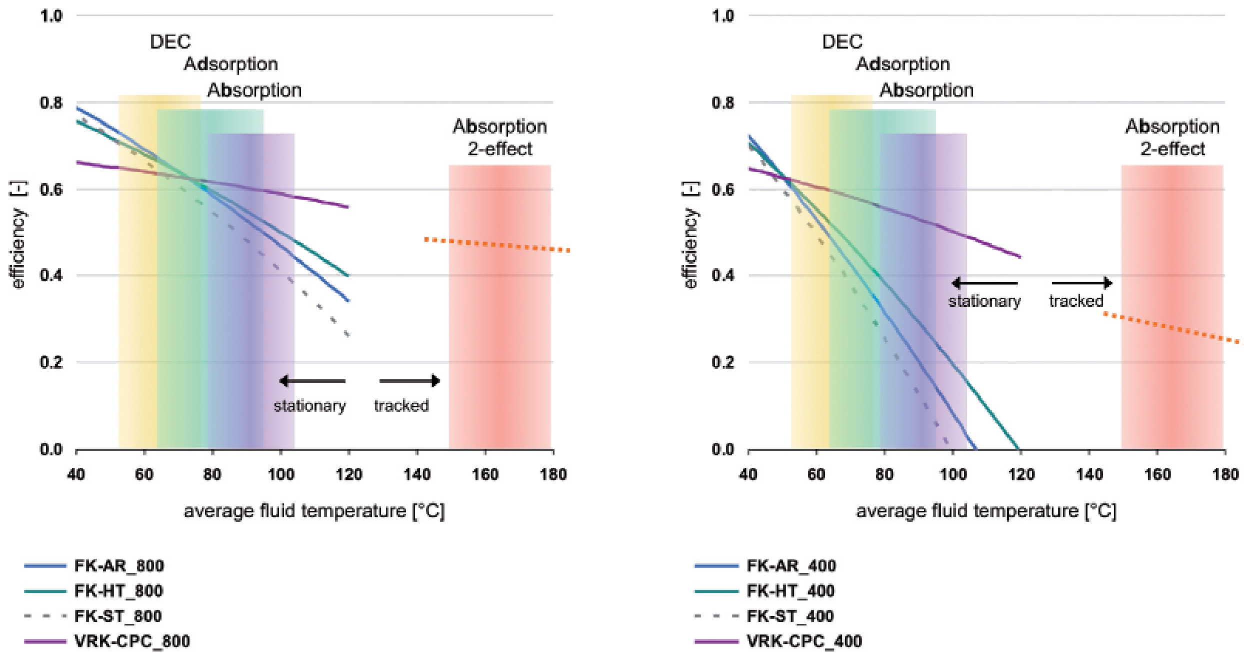


Figure 3.13 Cross-section scheme of a stationary CPC collector

Figure 3.14 Example of an installation [15].

The heat losses in solar concentrators occur mainly due to radiation at elevated temperatures. The higher the concentration ratio, the higher the efficiency, as well as the price. Usage of selective coatings on the reflecting mirrors significantly raises the efficiency at low increase in price, so the concentrators with low *R* can be used instead an expensive collector with higher *R* and non selective surface. In general, concentrating collectors are expensive due to complex mechanisms needed to ensure sun tracking and due to requested accuracy in positioning the absorber in focus, as well as in obtaining a parabolic form of reflecting mirrors, which is especially heavy task for larger concentrators. Mirror surfaces are subject to oxidization and corrosion processes as they are operated under elevated temperatures. In addition, they should be maintained free of dirt so as to ensure a high reflectance to the absorber surface. All this so far restricted the utility of concentrating collectors, but new materials and construction developments, such as sheets with aluminized plastic film as a substitute for heavy glass mirrors, are recently making them a promising solution, especially, for large scale application. As a summary, Fig. 3.15 shows efficiency curves for different types of flat type, evacuated and CPC collectors with indication of operating range for particular solar cooling technique.



FK-ST Flat-plate collector, standard product
FK-AR Flat-plate collector, 1-cover glass, anti-reflective coated
FK-HT Flat-plate collector, 1-cover glass, convection barrier foil, improved insulation
VRK-CPC Evacuated tube collector, direct mass flow, Sydney type with external CPC-reflector

Figure 3.15 Efficiency curves for different types collectors and solar radiation (800 W/m^2 and 400 W/m2), referred to the collector aperture area [13].

3.5 Solar air collectors

The solar air heating systems offer some advantages against solar hot water heating systems. A problem with freezing and overheating (boiling) is eliminated as well as are corrosion problems. A high degree of stratification possible with the pebble bed entails lower collector inlet temperatures, thus, the higher collector efficiency. Disadvantages would include a relatively high air pumping costs, larger storage volumes needed, difficulties with proper sealing and related losses of the heated air in the collectors and ducts. The air solar collectors are operated with the lower efficiency (25 - 35%) than the liquid fluid ones, due to poorer heat transfer capabilities of air, Fig. 3.16. This is usually compensated by increasing the effective absorber area by means of porous materials or finned constructions. Air heaters are relatively cheap provided they are built of light materials, with no high pressure resistance demands put on liquid fluid collectors (6 - 10 bar). In deciding on a solar air collector not only has the thermal efficiency to be considered but also the electric energy consumption for ventilating the air through the collector.

Air collectors using evacuated tubes have been introduced to the market recently, Fig. 3.16. Air flows in parallel through several absorber tubes. An annular gap type absorber tube is used to direct and redirect the air through it. The evacuated tube is fully made of glass and has a selective coating on the inner glass tube inside the vacuum. This collector is able to produce heat at temperatures above 100°C with sufficient efficiency. Thus the collector seems appropriately designed for the operation of heat driven cooling systems and in particular desiccant cooling systems (see chapter 5).

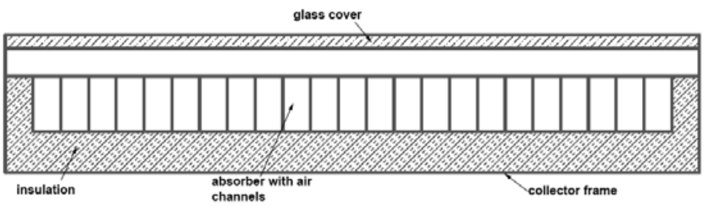


Figure 3.16 Solar air collector and its schematic cross section [16].

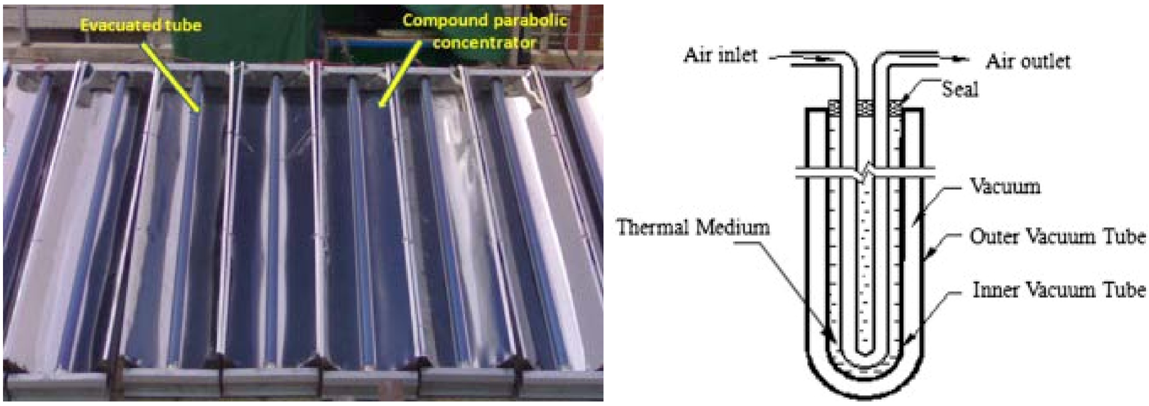
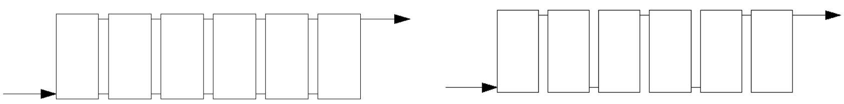


Figure 3.17 Solar air collector with evacuated tubes [17]

3.6 Arrangement of collectors

Collectors are assembled either in the parallel or serial arrangement (Fig 3.18). The parallel arrangement is favourable from a pressure drop point of view, but it requires more piping and larger pipe dimensions due to larger flow rates. The serial arrangement produces larger temperature rise per each fluid pass and higher outlet temperature since the flow rates are lower, but still the pressure drop through one array (bank) can be substantially higher than in the parallel connected array. Due to the higher working temperatures, collectors connected serially operate with lower efficiency, but sometimes such arrangement is unavoidable, especially in the areas with lower solar radiation available through a year. In this case, the collector fluid outlet temperatures can be high enough to assure the collected heat is effectively transferred to the water in a storage tank by the heat exchangers of a reasonable size.



a) **b)**

Figure 3.18 a) Parallel arrangement, b) Serial arrangement

Typical values for the flow rate are 40 - 80 l/h per m^2 of the absorber active surface area. The test flow rates are usually about 70 l/m^2h (according to HRN EN 12975-2), which means that collectors operating with lower flow rates will have somewhat lower efficiencies than those calculated from the test results which are commonly provided by a collector manufacturer (Fig. 3.19). Some manufacturers allow flow rates as low as 15 $l/(m^2 \cdot h)$ which might result in a remarkably disrupted thermal performances compared to those achieved during tests with the higher flow rates. Another issue that might impair collector performances is the mal-distribution in a collector array. Collectors in central part of the array have the smallest flow rates as the pressure difference between the collector inlet and outlet is also low there. Those collectors, however, will be hotter than the others having, indeed, consequently higher heat losses.

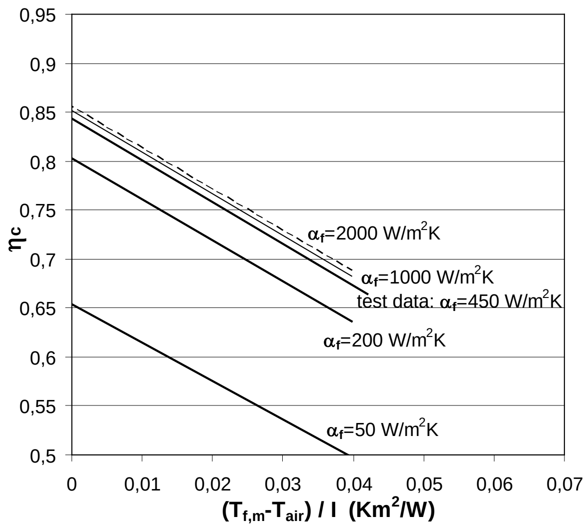


Figure 3.19 Influence of the heat transfer coefficient in tubes α_f on collector efficiency, based on CFD simulation in FLUENT [12].

For this reason, the number of collectors assembled in an array should not exceed 8 - 10. The most manufacturers recommend only 5 to 6 collectors to be connected in the parallel arrangement. In the serial arrangement the maldistribution problems do not occur, but a high number of collectors in the array will, however, entail too high pressure drop. Figure 3.20 shows some examples of the manner of connecting collectors which have been proved to yield fairly uniform flow distribution and temperatures.

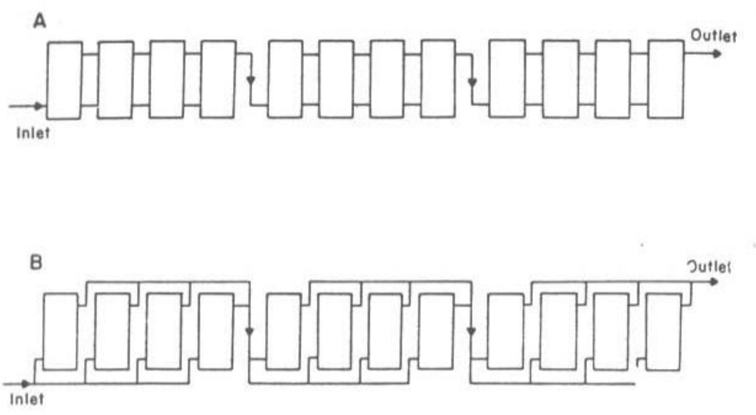


Figure 3.20 Example of alternative methods for connecting collectors: a) series-parallel, b) parallel-series [5].

4. Solar hot water systems

4.1 Types and characteristics

The most frequently encountered systems for utilization of solar energy are active solar hot water systems used in low temperature applications (40 - 60°C) for mainly domestic water purposes, space heating and swimming pool heating. As already mentioned before, they consist of a plate or evacuated tube collector which supply heated water to a storage tank. Circulation of a heat carrier (water or water/glycol mixture) between the collector and the tank can be obtained by a pump or by a natural circulation due to temperature gradients between the collector and the tank (thermosiphon systems-normally not employed in solar cooling systems).

Fig. 4.1 shows a system with forced convection and a single storage tank. Unlike the natural circulation systems, the forced circulation systems uses a thermostatic control to shut of the pump when the temperature in tank closely approaches the temperature at the outlet of collector (3 - 5°C). The pump is turned on when a sufficient large temperature difference is established again.

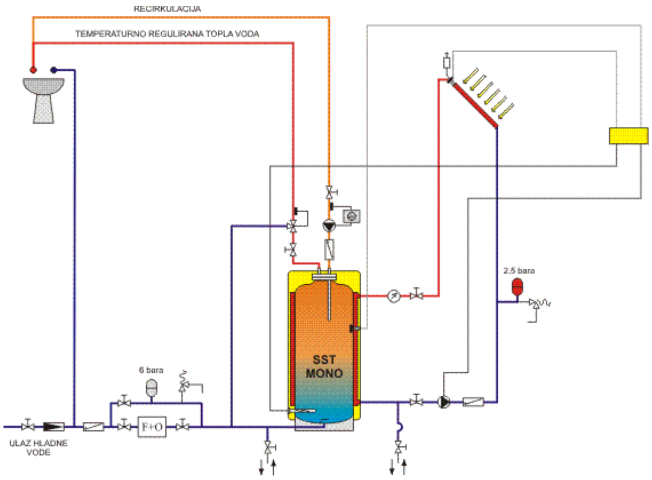


Figure 4.1 Forced hot water system with a single tank [12].

Higher system efficiency is achieved with the internal tank (Fig. 4.2). In this way, the upper layers in the larger tank with hot water are prevented from mixing with the cold tap water from the bottom of larger tank which is delivered directly to the internal tank. The water stored in the internal tank exchanges heat with the surrounding outer water which is heated by solar collectors and auxiliary heater. Also, this water is normally delivered to the heating devices within the object (radiators, floor heating systems etc.) which is an additional reason why it is separated from the domestic hot water in the internal tank. All this ensures the warmest water is isolated at the top of the tank and colder water is brought to the collectors increasing their efficiency i.e. the overall system efficiency.

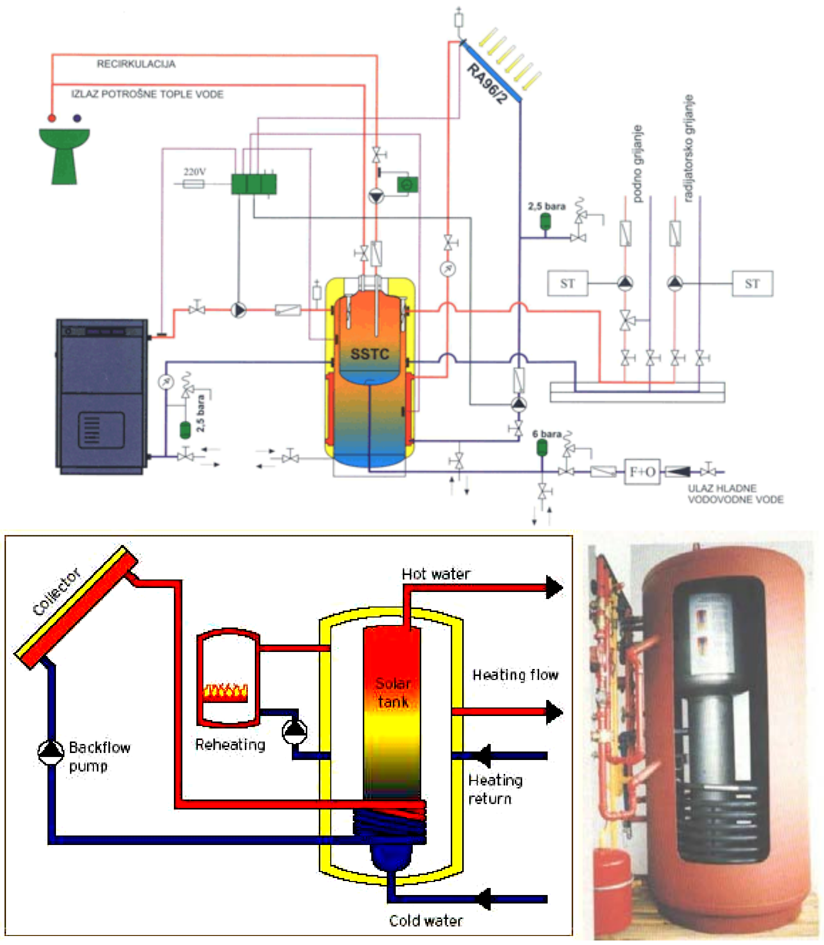


Figure 4.2 Examples of forced hot water system with “tank-in-tank” configuration [12].

Larger system might benefit from the configuration with two tanks (Fig. 4.3). In the two tank system the collectors first supply energy to one storage tank until a certain temperature is reached (e.g. 50°C). Then the collector fluid is directed by means of a three way valve to another tank which is heated until the temperature in the first tank falls below a selected value or when a certain selected temperature is reached in the second tank. Then the flow is again redirected to the first tank. The advantage of such systems is that in winter months only one tank can be heated to sufficiently high temperatures needed for domestic water purposes, while in the summer period the operation of both tanks will assure a maximum of solar energy is stored, at the same time preventing the system from overheating. Also, the desired water temperature in one tank is reached faster than it would in a single tank system with larger tank.

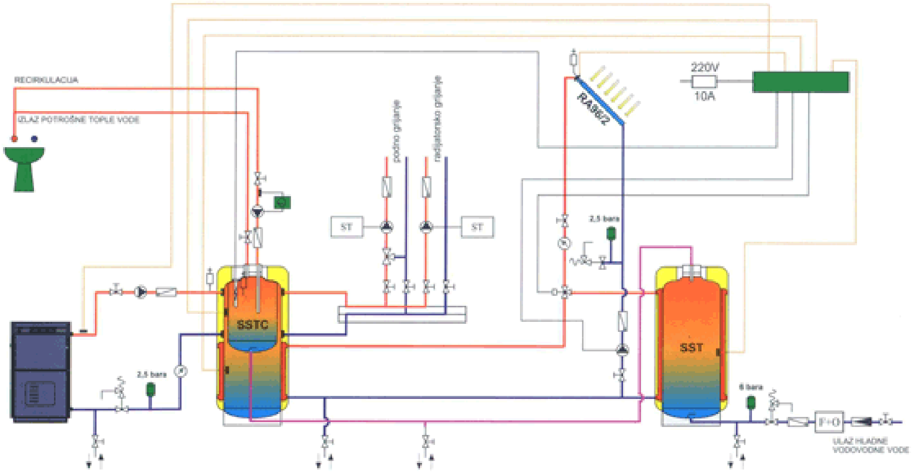


Figure 4.3 Two tank system

The best system performances are achieved with systems like those shown on Fig 4.4 In this case the domestic water is heated via an internal or external heat exchanger, by which is avoided mixing of hot water from the top of tank with cold tap water. The drawback is an additional cost for heat exchanger and especially the maintenance problems (cleaning).

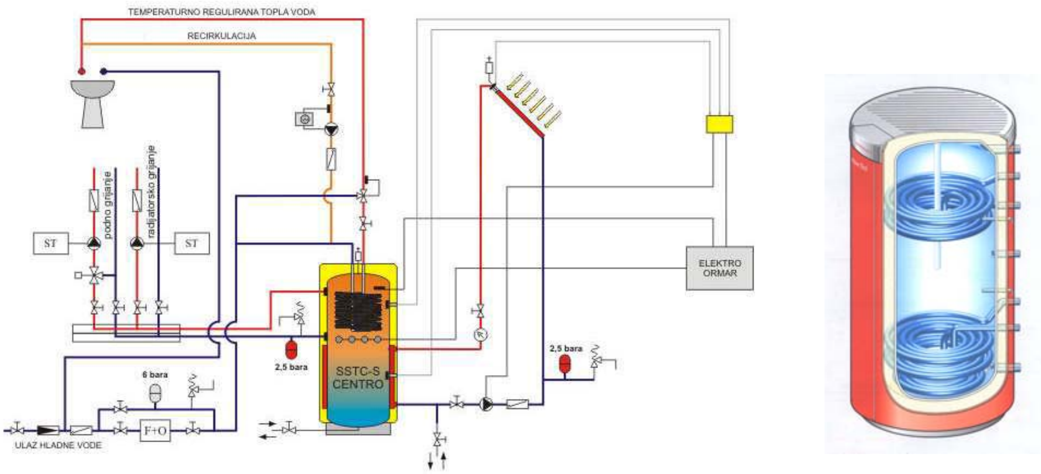


Figure 4.4 System with additional internal heat exchangers, recent design of a tank with two spiral heat exchangers [12].

In recent systems, collector loop fluid releases heat to the tank water through an internal heat exchanger, which is commonly found in the form of coaxial tube (especially in smaller household systems). In special constructions collector fluid is passed outside the tank (Fig. 4.4) or exchanges heat in an external plate heat exchanger (Fig. 4.5). The efficiency of these heat exchangers will also influence the overall system efficiency. Internal heat exchangers in the form of coaxial tube are simplest solution but they operate with a relatively low overall heat transfer coefficient because of the free convection heat transfer outside the tubes. A care must be exercised when dimensioning such heat exchangers to assure all heat collected in a collector is exchanged with the tank water using as less as possible the heat transfer area (i.e. the lowest cost heat exchanger).

Plate heat exchangers are good solution from the heat transfer point of view, but they are sensitive to lime, which entails periodical chemical cleaning, making them suitable for larger systems and industrial applications.

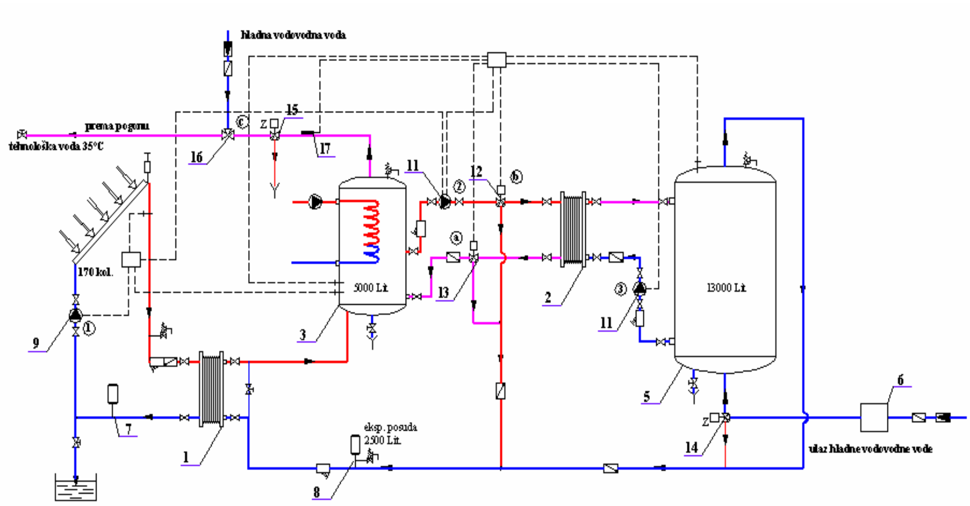


Figure 4.5 Large scale solar hot water system with external heat exchangers [12].

Although the most of solar water systems are used for heating of domestic water, very often they are used for low and middle temperature space heating. Hot water from the storage tank can be delivered to conventional radiators operating at temperatures regimes (40 - 60)°C or to the floor or wall heating system which operate at temperatures of about 30°C. Unfortunately, the space heating systems require large collector area, which causes problems with overheating in summer months when usually a less energy is demanded. Therefore, the space heating systems should be carefully sized in order to provide enough heat in winter periods for both domestic water purposes and space heating, but to avoid the overheating in summer. Very often such systems are sized to meet the domestic water demand in summer periods in weekend houses occupied seasonally, while in winter periods the collected solar energy is mostly used to keep house warm and dry. Another option is to use more collector area for space heating in winter and for swimming pool heating purposes in summer.

4.2 System control

The correct system control operation is crucial for an efficient operation of the whole system. The system control consists of temperature sensors which measure temperature at collector outlet and at different positions in storage water tanks. The electronic control decides: when the pump will be in operation, which tank will be collector fluid directed to, when an additional heating (electrical or from hot water boilers) should be turned on/off etc.

4.3 Components, equipment and installation tips

In general, pipes in solar systems should be as short as possible in order to reduce the heat losses. The pipes connecting collector and tanks are normally insulated with special UV rays resistant insulation (such as Armaflex UV). In larger systems the insulation can be protected from the UV rays and wearing by aluminium sheets (it also provides protection from birds). In smaller systems (up to 10 m^2 of collector area) connecting pipes are normally of dimension ranging from 10 to 20 mm. Collectors are mounted directly on roofs (see Fig. 4.6a), or they can be incorporated into a roof (Fig. 4.6b) which is a better solution from the economical and heat losses point of view (collector provides an excellent insulation to attic space of a house). They can be also fixed to the special carriers which allow them to be placed in e.g. backyards or on flat roofs, Fig. 4.7.

a) b)

Figure 4.6 a) collector mounted on the roof; b) collector incorporated into the roof [18]

Figure 4.7 Collector attached to the carrier [18]

In larger systems collectors are connected in banks, Fig. 4.8. A special care should be devoted to the sizing of the pipes in order to feed all banks with identical flow rates, which is additionally controlled by valves having the flow meter incorporated. It should be possible to disconnect each bank from the system in cases when some of collector in the bank is to be replaced (e.g. when it is broken, or it is leaking).

Figure 4.8 Large scale system with collector banks mounted on a flat roof [19]

5. Solar cooling systems

5.1 General

The most of solar cooling systems are nowadays used for comfort cooling in buildings and for food refrigeration. Due to the low efficiency and high investment cost (few times as high as cost of conventional refrigerating systems) they have been mainly applied in remote areas with no access to electric supply or in cases where a waste heat is available. Solar cooling is expensive, even more then solar heating. An appropriate design of new buildings or reconstruction of older ones are needed to minimize loads on the air conditioning and heating system, as those measures are far less expensive than providing additional cooling/heating. Furthermore, unlike in the case of solar heating, cooling demand and availability of solar radiation are in a phase (summer period). Most of solar systems designed for space heating are oversized in regard to the hot water demand in summer period, which causes problems with the system overheating. Combining such systems with solar cooling system improves the overall annual efficiency of the combined system. Solar cooling systems have been improved technically in past ten years and tested through a number of various pilot projects. Hence, solar cooling is expected to become more and more an alternative to conventional cooling systems, especially in Mediterranean region with high solar gains and high cooling demands.

The most important technologies applied in solar cooling are (see also Fig. 5.1):

- absorption cooling
- adsorption cooling
- closed cycles
- open cycles (desiccant cooling)
- PV driven vapour compression chiller
- solar driven Rankine-cycle (mechanical systems)

Conversion of solar energy into electricity by photovoltaics (PVs)to drive a classical motor driven vapour compression chiller is not further considered here, since the electricity produced by PV system today mainly feed into the public grid.

Systems with solar driven Rankine-cycle provide electrical energy to the conventional air conditioning systems. Such systems suffer from the low efficiency due to the relatively low temperatures available from the concentrating collectors and related low Rankine cycle performance. Due to high investment cost and low efficiency, they have minor market share compared to the other solar cooling systems. For these reasons, they will also not be treated in detail here.

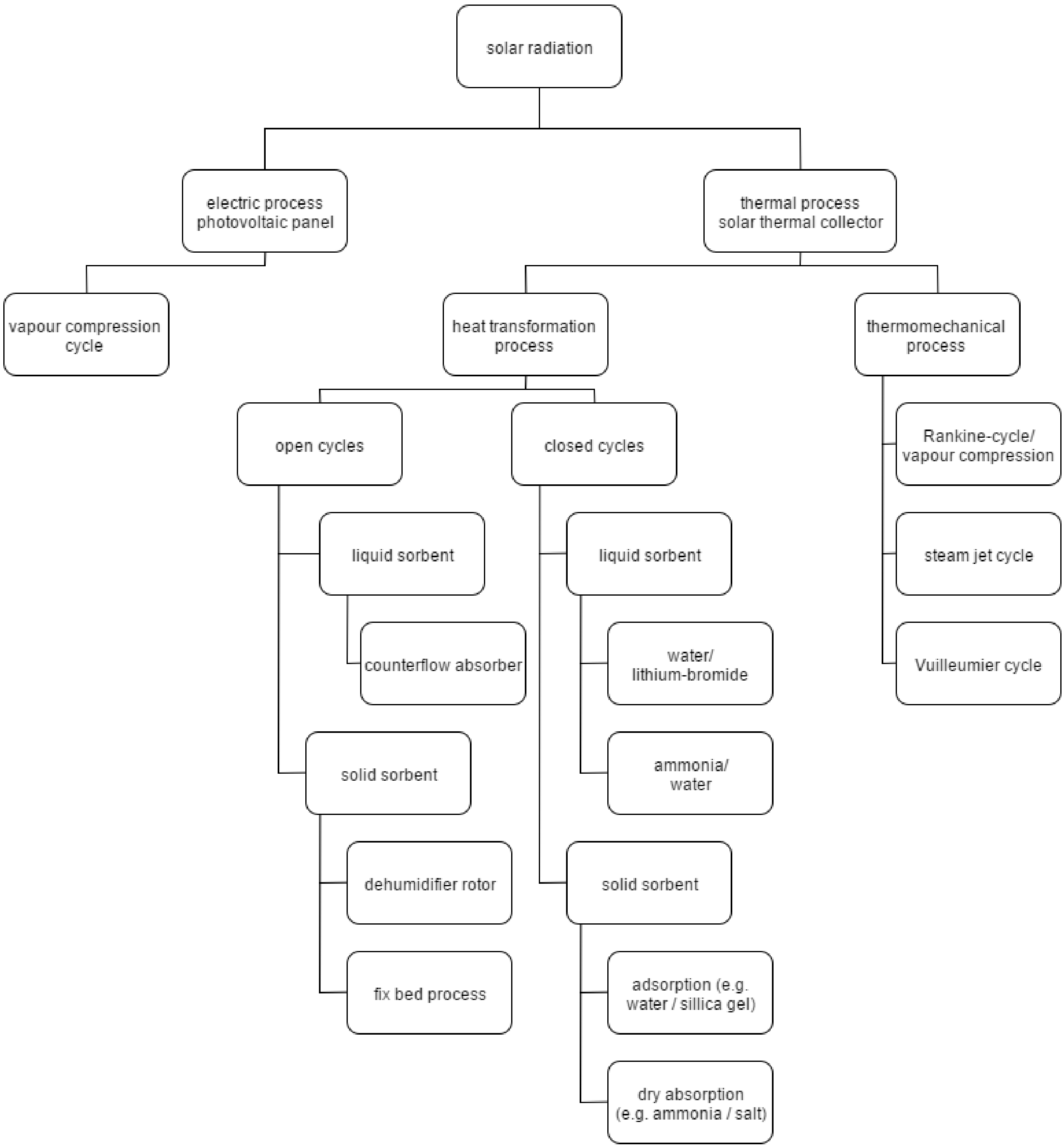


Figure 5.1 Overview of most important technologies applied in solar cooling

Solar cooling systems apply solar thermal collectors as a heat source to thermally driven cooling devices (chillers). They main components are: solar collectors, storage tank, air conditioning subsystem, including various forms of cold distribution, and auxiliary (backup) subsystems.

As shown on Fig. 5.2 solar cooling system can be divided into three subsystems:

1. hot side consisting of solar thermal system which provides heat to the generator G of the chiller
2. heat rejection of condenser (C) and absorber or adsorber (A) through the cooling water loop and cooling tower
3. cold side with the evaporator (E) which provides useful cooling via the chilled water loop

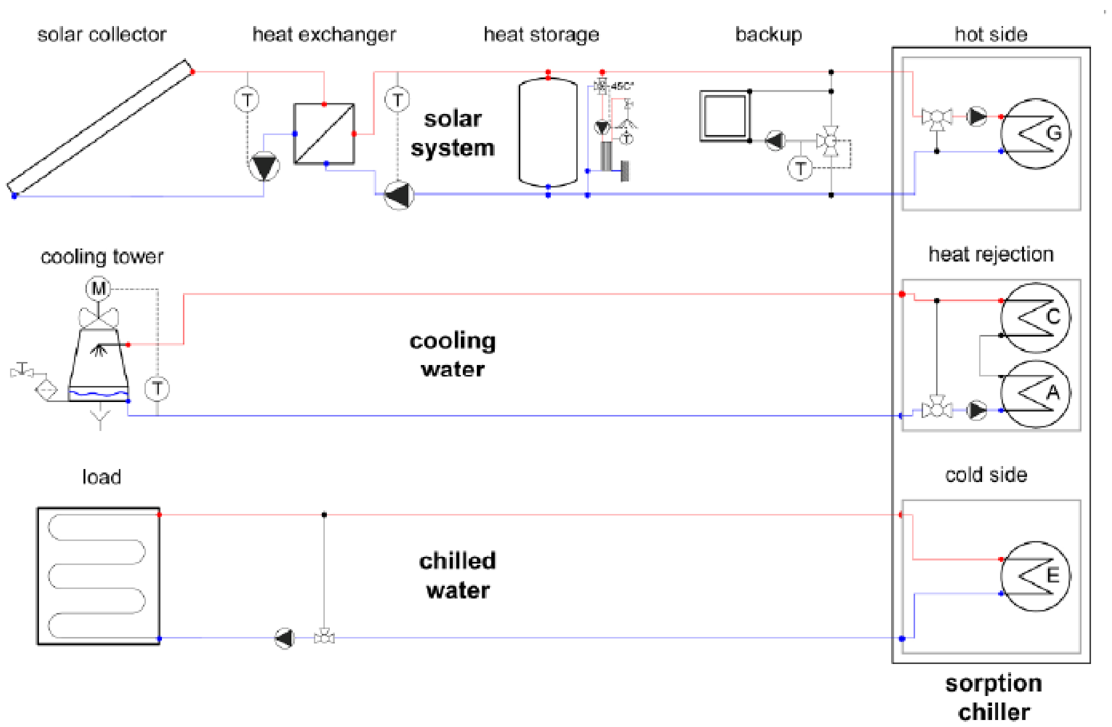


Figure 5.2 Thermally driven chiller connected to solar thermal system, cooling water loop and chilled water loop [20].

5.2 Energy performance: COP, SCPF, primary energy

The thermal coefficient of performance, COP_h, is a basic quantity to evaluate the quality of the conversion of heat into cold, defined as the useful cooling power Q_{cold}, per unit of driving heat Q_{heat}:

$$COP_h = \frac{Q_{cold}}{Q_{heat}}$$

Also, the electrical COP_{el} is used here, defined in the same way as for a standard electrically driven vapour compression chiller. Hence, COP_{el} is the ratio of cooling power Q_{cold} and driving electrical energy E_{el}.

$$COP_{el} = \frac{Q_{cold}}{E_{el}}$$

To evaluate the overall system (solar + chiller), the solar cooling performance factor (SCPF) is commonly used. It is defined as the ratio of cooling power to the solar radiation incident on the collector G

$$SCPF = \frac{Q_{cold}}{G}$$

As shown in Fig. 5.3, the efficiency of the solar collector decreases as the driving temperature increases, while the efficiency of the thermally driven adsorption chiller increases. Therefore, there is an optimum driving temperature, where $SCPF$ (COP_{sol}) is at maximum. This working condition can be achieved by advanced control for each set of input parameters.

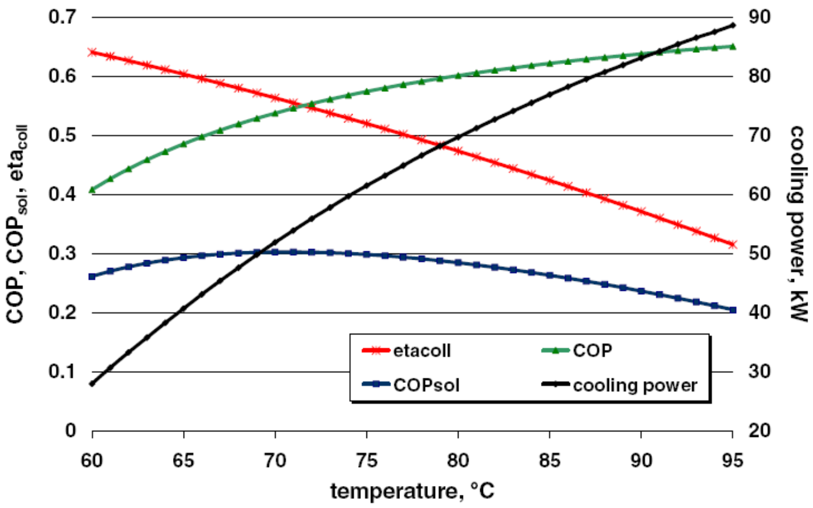


Figure 5.3 Characteristics of a silica gel adsorption chiller as a function of driving temperature, COP_h (COP), collector efficiency (etacoll), solar cooling performance factor SCPF (COP_{sol}) and cooling power [21].

Primary energy consumption is another important figure to evaluate the system performance. It can be obtained from

$$E_{prim} = \sum_i \left(f_{p,i} \cdot Q_{gen,in,i} \right) + \sum_j \left(f_{p,el} \cdot W_{aux,j} \right) \qquad \text{[kWh]}$$

$Q_{gen,in,i}$ – delivered non-renewable energy to the generator (kWh);

$W_{aux,j}$–auxiliary energy (kWh);

$f_{p,i}$– primary energy factor for particular source/fuel (national values);

$f_{p,\mathrm{el}}$– primary energy factor for electricity (national value).

5.3 Absorption cooling

The most frequently encountered systems on the solar cooling market are absorption systems, Fig. 5.4. The physical process of absorption cooling is based on use of mixture of two chemical components, one of them serving as the refrigerant and the other as the sorbent. More details on the operating principle can be found in [5] and [13].

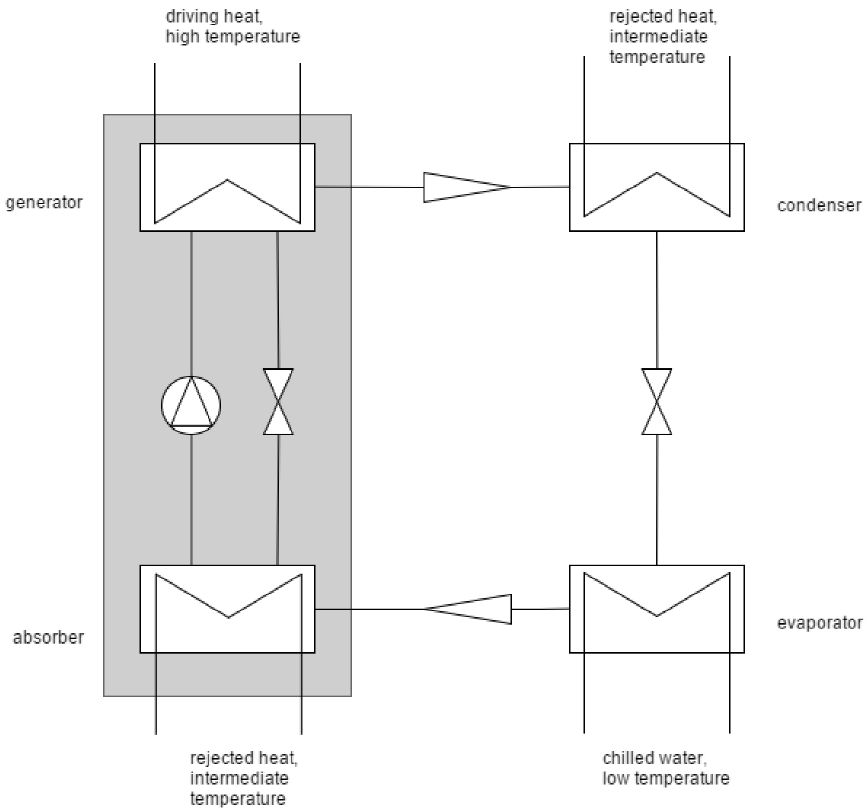


Figure 5.4 Absorption system

Absorption systems operate with driving temperatures at generator of (70 - 95)°C. The working media is water/lithium-bromide (LiBr) mixture (water is refrigerant), water/lithium-chloride (LiCl) (water is refrigerant), ammonia (NH_3) /water mixture (ammonia is refrigerant). Energy to the generator is provided by flat or evacuated collectors. The condenser and absorber can be cooled by water or air. A cooling tower is normally part of the system in the case of water cooling. The cooling capacity is greatly affected by a change in heat rejection temperature. It is therefore important to correctly size and design the heat rejection device (e.g. cooling tower).

LiBr and LiCl absorption chillers are mostly used for air-conditioning. Chilled water temperatures shall be above 6°C to avoid freezing of the refrigerant (water) in the evaporator.

The ammonia-water chillers can be used in refrigeration systems with chilled water temperatures below the freezing point, as the refrigerant ammonia has a lower boiling point than water. The thermal COP_h varies between 0.6 and 0.8. This low COP_h together with high collector operating temperature, leads to the rather low solar performance cooling factor SPCF = 0,1 - 0,15. These numbers are very often even lower during the transient periods between successive system start up and shut down, which is related to a variable weather conditions (solar radiation).

Beside the above described so called single stage (effect) machines, chillers using a double stage cycle are available. Two generators working at different temperatures are operated in series, whereby the condenser heat of the refrigerant desorbed from the first generator is used to heat the second generator. Thereby a higher COP_h in the range of 1.1 - 1.2 is achieved. On the other hand, driving temperatures between 140 and 160°C are required, which imposes need to use highly efficient and expensive concentrating collectors.

Absorption chillers are mainly available on the market at cooling capacities 30 kW - 10 MW. Also a few commercial low power systems, e.g., below 30 kW, are available (only single stage machines). They all produce chilled water that can be used in combination with any air conditioning equipment such as an air-handling unit, fan-coils, chilled ceilings, etc., Fig. 5.5. A number of large-scale solar cooling systems for large commercial building and industrial applications have been successfully demonstrated through pilot projects, which was a basis for introduction to the wide market. Also, there is a great potential for use of low power systems (especially for those < 10 kW) in residential and small size building applications, concerning the increasing cooling demand and expected price reduction of the equipment. Yet, for wider commercial use of small-scale solar cooling systems, more research and development is needed.

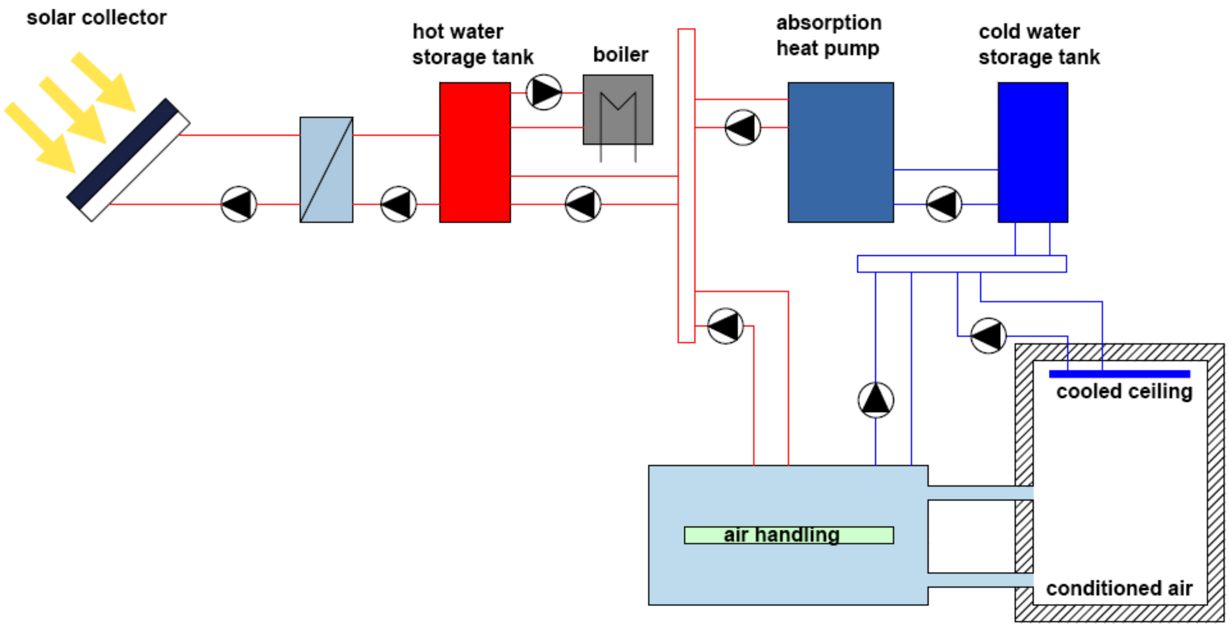


Figure 5.5 Application of solar cooling system in HVAC system [12].

5.3.1 Water-LiBr chillers

The most commonly used chillers in commercial applications are those working with water-LiBr mixture.

- Driving temperatures at generator are 70 - 95 °C for single-stage chillers
- Heat from absorber and condenser is released at a low temperature level from 27 - 30 °C
- Chilled water is available at temperatures between 6 - 15 °C
- Water/LiBr chillers operate under vacuum

Cooling capacity of water/LiBr chillers depends on working parameters as (Fig. 5.6):

a) 1 °C decrease of heat rejection temperature entails increase of cooling capacity by 7%

b) 1 °C increase of heat removal temperature entails increase of cooling capacity by 5%

c) 1 °C increase of driving heat temperature entails increase of cooling capacity by 3%

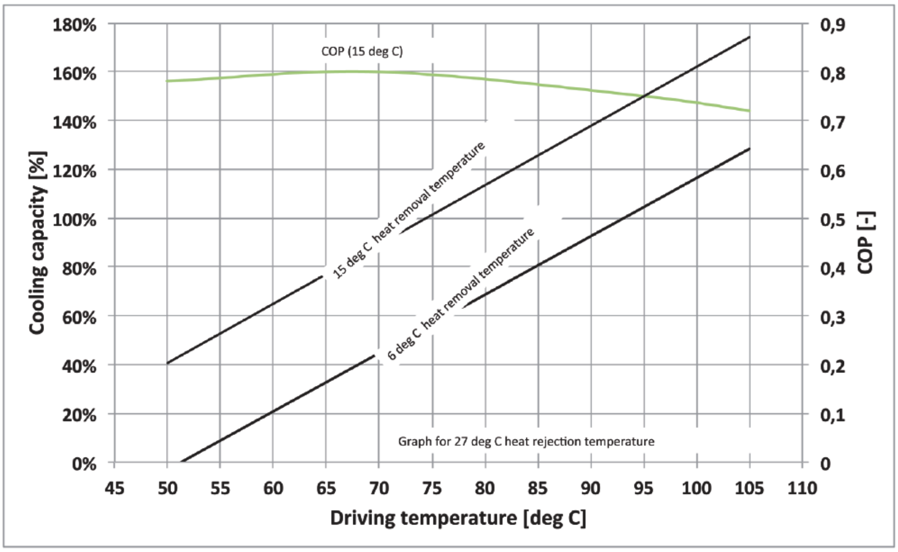


Figure 5.6 Influence of driving and heat removal temperature on cooling capacity of a single stage water/LiBr chiller [13].

The heat rejection temperature shall be kept above a minimum temperature and the driving heat temperature below a maximum temperature, due to risk of LiBr solution crystallization and consequent damage of chiller internal piping. Therefore, temperature monitoring of internal parts is often applied in commercial chillers in order to avoid the crystallization. Commercial single and double stage water/LiBr chillers are available on the market from small cooling capacities (6 kW) up to large scale units (11 MW). Table 5.1 shows some commercially available chillers up to 50 kW of nominal cooling power.

Table 5.1 Commercially available Li-Br/water absorption chillers up to 50 kW, examples.

Company	Product name	Sorbent-Refrigerant	Nominal cooling capacity [kW]	Nominal hot water temp. [°C]	Nominal chilled water temp. [°C]	Nominal cooling water temp. [°C]	Size [m]	Weight [kg]
Climate-well	CW 10	$LiCl/H_2O$	6	80	7	30	2.2x1.4x0.7	740
EAW	Wegracal SE 15, ESC15	$LiBr/H_2O$	15	90	10-15	27	1.75x0.76x1.75	700
Thermax	LT 0.5	$LiBr/H_2O$	17.5	85	6-8	30	2.0x1.0x0.9	1300
Yazaki	WFC-SC5, WFC18	$LiBr/H_2O$	17.5	88	6-8	31	1.82x0.74x0.6	420
EAW	Wegracal SE 30, ECS30	$LiBr/H_2O$	30	90	10-15	27	2.2x0.79x2.14	1400
Thermax	LT 1	$LiBr/H_2O$	35	70-110	6-8	31	2.1x1.6x1.6	2000
Yazaki	WFC-C10, WFC35	$LiBr/H_2O$	35	88	6-8	31	1.98x0.86x0.97	600
EAW	Wegracal SE 50, ESC50	$LiBr/H_2O$	50	86	10-15	27	2.31x1.10x2.95	2250

The regular annual service of water/LiBr chillers comprises its draining and eventual cleaning of the water boxes (typically every 3 - 5 years depending on the water quality). Also, de-airing of the system is occasionally needed due to operation under vacuum, which is automatically scheduled. In small scale units de-airing is done within the annual service. The lifetime of water/LiBr chillers is 20 years or even more as documented in some cases.

5.3.2 Ammonia-water chillers

The ammonia water cycle is similar to a water/LiBr cycle and employs the same components, except for an additional rectifier that is used here to separate ammonia vapour from the retained water after generator, Figure 5.7.

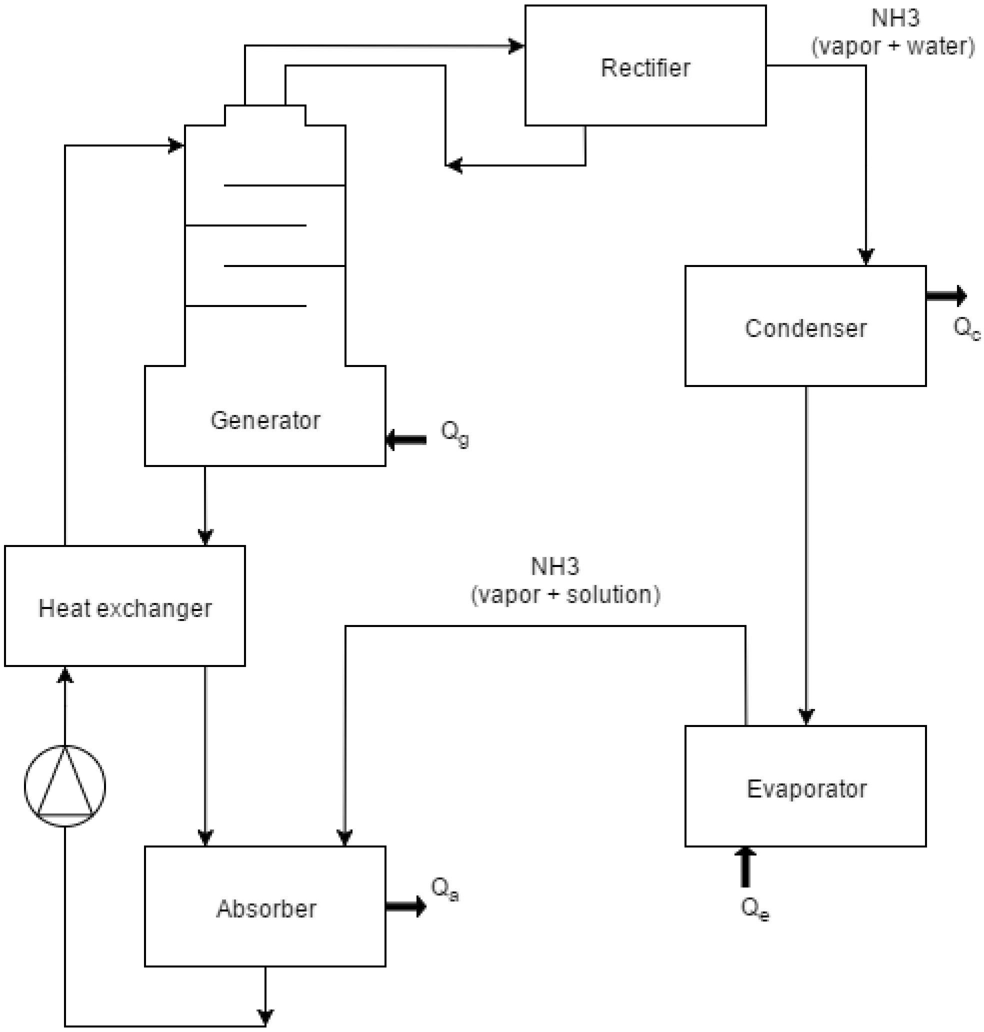


Figure 5.7 NH3/water absorption process (single-stage).

Unlike water/LiBr chillers, NH_3/water ones operate at positive pressure and hence do not require a regular vacuum purge. NH_3/water chillers can operate at higher heat rejection temperatures of around 30 - 40°C, even up to 65°C in some cases, which increases cooling capacity and allows for further use of this otherwise waste heat (e.g. for heating of DHW). Also, higher rejection temperatures allow use of a dry-cooling system instead of wet cooling towers.

On the other hand, chillers operating with highly toxic ammonia must be installed and operated according to special safety regulations.

Also, they require somewhat higher driving temperatures compared to water/LiBr chillers for the same cooling capacity. Similar influence on cooling capacity has heat rejection, removal and driving heat temperatures as in the case of water/LiBr cycles.

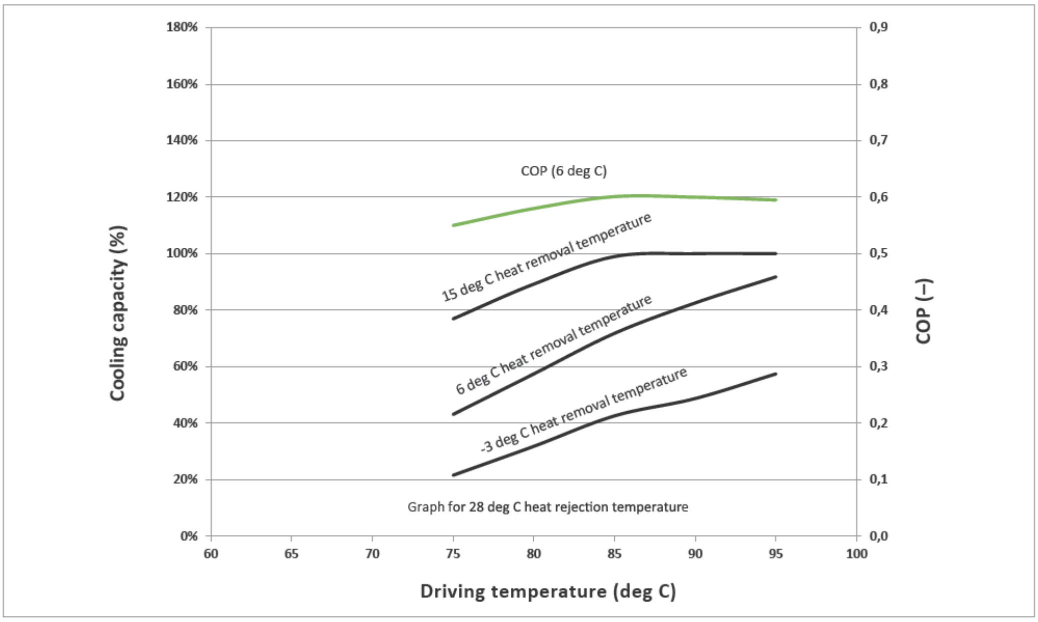


Figure 5.8 Influence of driving and heat removal temperature on cooling capacity of a single stage NH3/water chiller [13].

Today, ammonia-water chillers are also commercially available in the small cooling capacity range <50 kW, not only as large scale units as they used to be in past (100 kW to 1 MW). Strong development efforts have been undertaken mainly in Europe to develop small scale units as a result of constantly growing comfort requirements in residential sector. Some examples of commercially available ammonia-water chillers up to 50 kW are given in Table 5.2.

Table 5.2 Commercially available ammonia-water absorption chillers up to 50 kW - examples

Company	Pink	Solarice	AGO
Product name	Chilii PSC 19	AAC25	chillii®ACC50, congelo 50
Working pair	Ammonia/water	Ammonia/water	Ammonia/water
Cooling capacity [kW]	19	25	50
Heating temperature [°C]	85	95	115
Heat rejection temperature [°C]	24	24	25
Cold water temperature [°C]	6	-3	-10
COP	0.62	0.5	0.54
Dimensions (LxDxH) [m]	1.9x0.8x0.6 m	2.0x1.4x1.4 m	2.35x2.63x1.35 m
Weight [kg]	550	950	1600
Electrical power [W]	450	N/A	N/A

5.3.3 Water/LiCl chillers

The problem of crystallization in water/LiBr chillers was addressed in the foregoing text. Water/LiCl absorption chillers make use of crystallization. The concentration of lithium chloride raises being heated by e.g. solar collectors, until it finally crystallizes. During this heating process, the water (refrigerant) completely evaporates from the LiCl and condenses in a separate vessel. After the LiCl fully crystallizes free of water, the process is reversed. In this way, the concentrated LiCl salt can be used as a thermo chemical storage, which enables night cooling when no solar energy is available.

5.4 Adsorption cooling-closed cycles

In adsorption chillers the refrigerant (water) vapours adsorbed by a solid sorbent (e.g. silica gel or zeolite) accompanied by release of water latent heat on sorbent until it becomes saturated – refrigeration phase. The desorption phase consists of heating the sorbent at low temperatures of 55 - 90°C suitable for use of solar energy. Since the refrigerant is water, to allow evaporation at temperatures 9 - 15°C, the chillers have to be operated in a vacuum. Adsorption chillers consist of two parts (modules), where in each one the adsorption and desorption phase occurs in regular cycles intervals. When one of the modules is working in desorber phase, another one is working in adsorbing phase, and vice versa, Fig. 5.9.

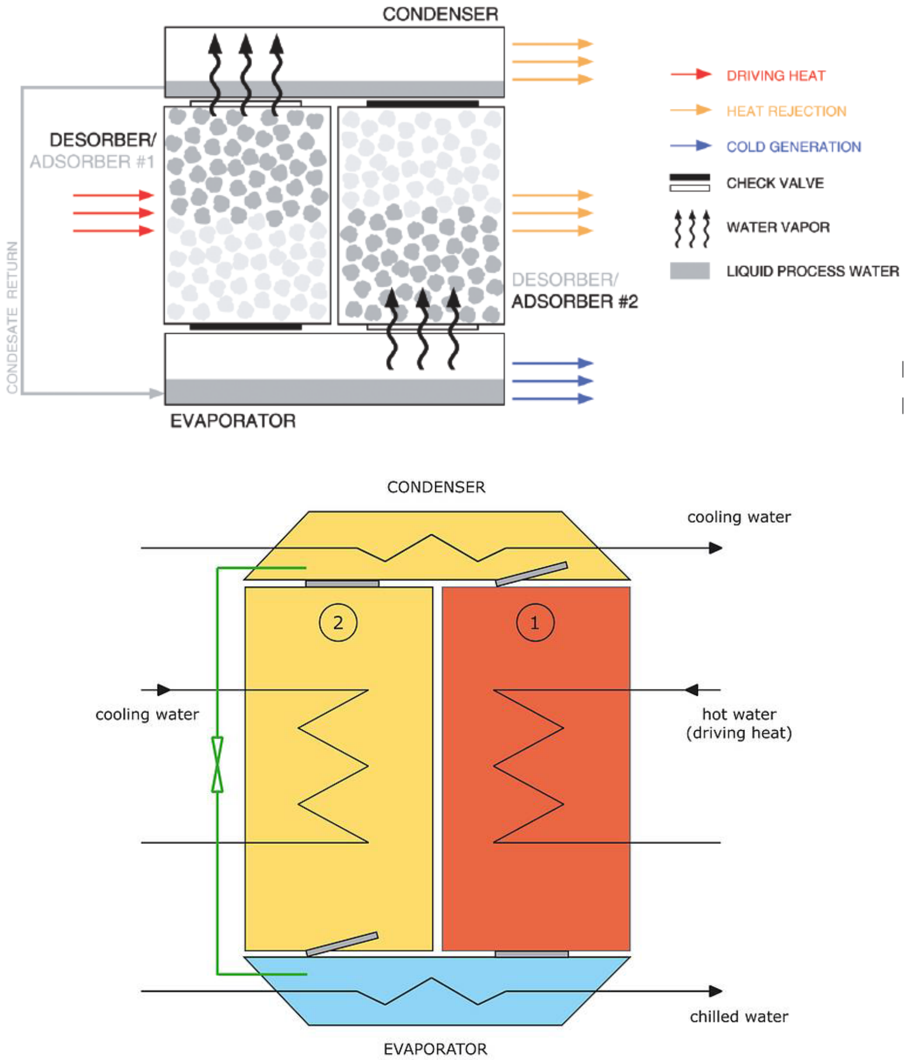


Figure 5.9 Schematic of adsorption chiller [13], [22] and [23].

Cooling capacity of water/zeolite chiller depends on working parameters as (Fig. 5.10):

a) 1 °C decrease of heat rejection temperature entails increase of cooling capacity by 4%

b) 1 °C increase of heat removal temperature entails increase of cooling capacity by 3%

c) 1 °C increase of driving heat temperature entails increase of cooling capacity by 2%

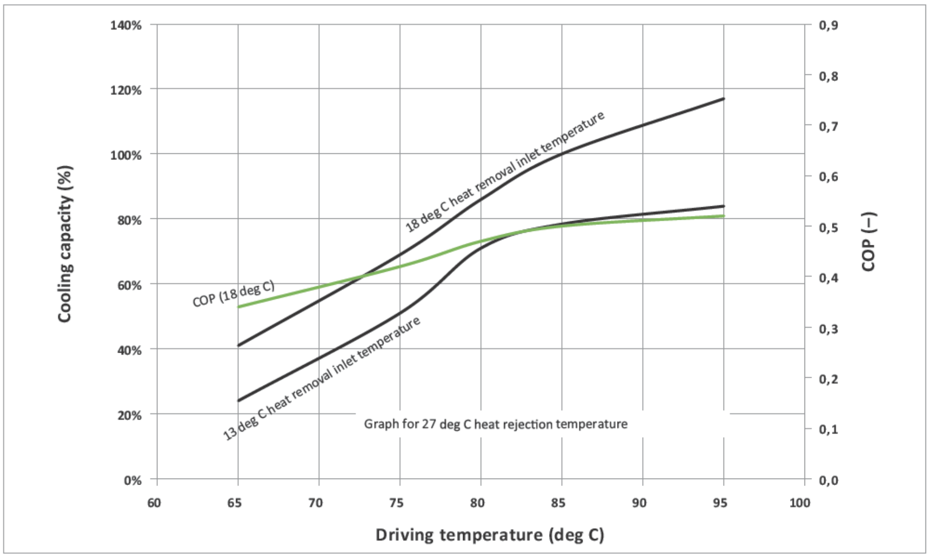


Figure 5.10 Influence of driving and heat removal temperature on cooling capacity of a water/zeolite chiller [13].

Adsorption chillers are commercially available in the range from small to medium scale units (7 - 500 kW of nominal cooling capacity). Table 5.3 shows some examples of commercially available chillers up to 50 kW of nominal cooling capacity.

Table 5.3 Commercially available adsorption chillers up to 50 kW - examples

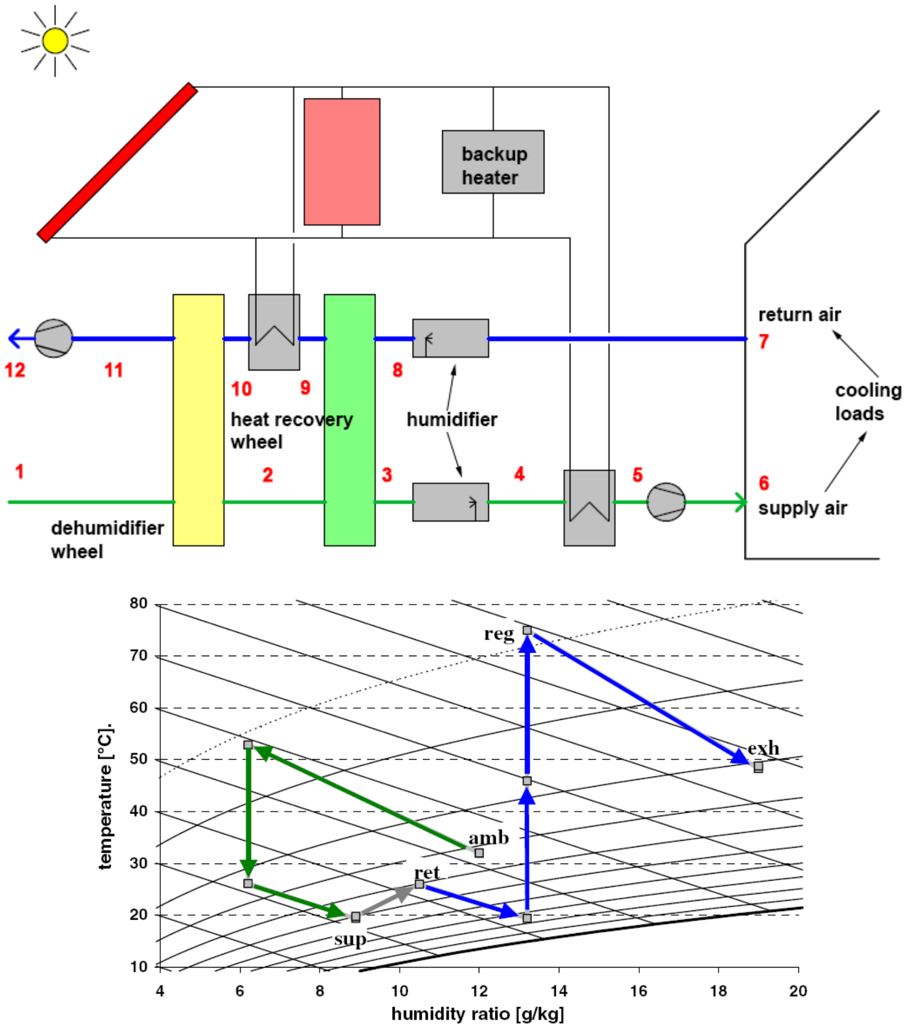
Company	Unit	InvenSor	SorTech	InvenSor	SorTech	Mayekawa
Product name		LTC 7	ASC 08	HTC 10	ASC 15	Mycom ADR-15
Working pair		water/zeolite	water/silica gel	water/zeolite	water/silica gel	water/silica gel
Cooling capacity	kW	7	8	10	15	50
Heating temperature	°C	65/60	72/65	85/77	72/65	75/70
Heat rejection temperature	°C	27/31	27/32	27/32	27/32	29/33
Cold water temperature	°C	18/15	18/15	18/15	18/15	14/9
COP at nominal conditions		0.54	0.60	0.50	0.60	0.60
Dimensions (LxDxH)	m	0.65x1.30x1.65	0.79x1.06x0.94	0.65x1.30x1.65	0.79x1.34 1.39	3.10x1.93x2.20
Weight	kg	370	295	370	590	5500
Electrical power	W	20	20	20	30	800

Typically, the thermal COP_h achieved is cca. 0.6, at the driving heat temperature of about 80°C. The advantage of adsorption chillers is simple construction and robustness. On the other hand, their construction is therefore relatively heavy. Since no internal pump for working mixture is needed, the electricity consumption is lower relative to absorption chillers. Also, there are no limitations regarding the heat rejection temperatures (no danger of crystallization). More improvements are yet expected in design of the heat exchangers in the adsorption compartments.

5.5 Adsorption systems-open cycles (desiccant systems)

Adsorption (desiccant) systems based on open cooling cycles directly produce conditioned air, unlike thermally driven chillers that produce chilled water furthermore used in other air-conditioning heat exchange devices. Here, open cooling cycle combines evaporative cooling with air dehumidification by a solid or liquid desiccant. Most often used solid desiccants are silica gel, titanium silicates, natural/synthetic zeolites, activated alumina. Liquid desiccants are water and ethylene glycol, LiCl, LiBr, and $CaCl_2$. In solar assisted desiccant cooling systems, solar energy is employed for desiccant dry out or regeneration. The most often, open cycle systems use rotating desiccant wheels, with silica gel or lithium-chloride as sorption material. All components are standard and have been employed in building air-conditioning applications for many years. Major advantage of this way of air conditioning is independent control of temperature and humidity. Also these systems run at lower operating costs and enable heat recovery. They generally improve indoor air comfort and reduce building maintenance due to high humidity. Hence, desiccant systems are normally used where the air change and/or the dehumidification of the indoor air are necessary or strictly prescribed. Examples are non-residential buildings like supermarkets, museums, and assembly halls with high occupancy.

Adsorption systems with liquid desiccants are less common on the market. They have physically separated absorber, where the air is dehumidified, and regenerator (desorber), where desiccant is regenerated. Such physical separation of the absorber and regenerator (desorber) enables regeneration of liquid desiccant at lower temperatures (at 60°C to 70°C), which increases solar collectors efficiency. The standard open cycle with a desiccant wheel is shown in Fig. 5.11 along with the changes of conditioned air states. Such systems operating are mostly used in temperate climates with moderate outdoor air humidity. In climates with higher air humidity e.g. in Mediterranean, an additional components shall be added. Fig. 5.12 shows the added sensible & latent heat exchanger which precooles and pre-dehumidifies ambient air by the exhaust indoor air, prior it enters the standard desiccant cycle. Fig. 5.13 shows system with additional cooling coils in return air supplied with cold water from a conventional chiller.



1-2 sorptive dehumidification of supply air;
2-3 pre-cooling of the supply air;
3-4 evaporative cooling of the supply air to the desired supply air humidity;
4-5 pre-heating of air (only in the heating season);
5-6 a small temperature increase is caused by the fan;
6-7 supply air temperature and humidity increased by means of internal loads;
7-8 evaporative cooling of return air;
8-9 the return air is pre-heated heat recovery wheel
9-10 regeneration heating;
10-11 desorbing of the desiccant material in dehumidifier wheel by means of the hot air;
11-12 exhaust air blown to the environment by the fan

Figure 5.11 Standard desiccant cooling cycle using a dehumidifier wheel with solar thermal systems and backup heater providing driving heat; change of the humid air states in T–x diagram during the conditioning process [21].

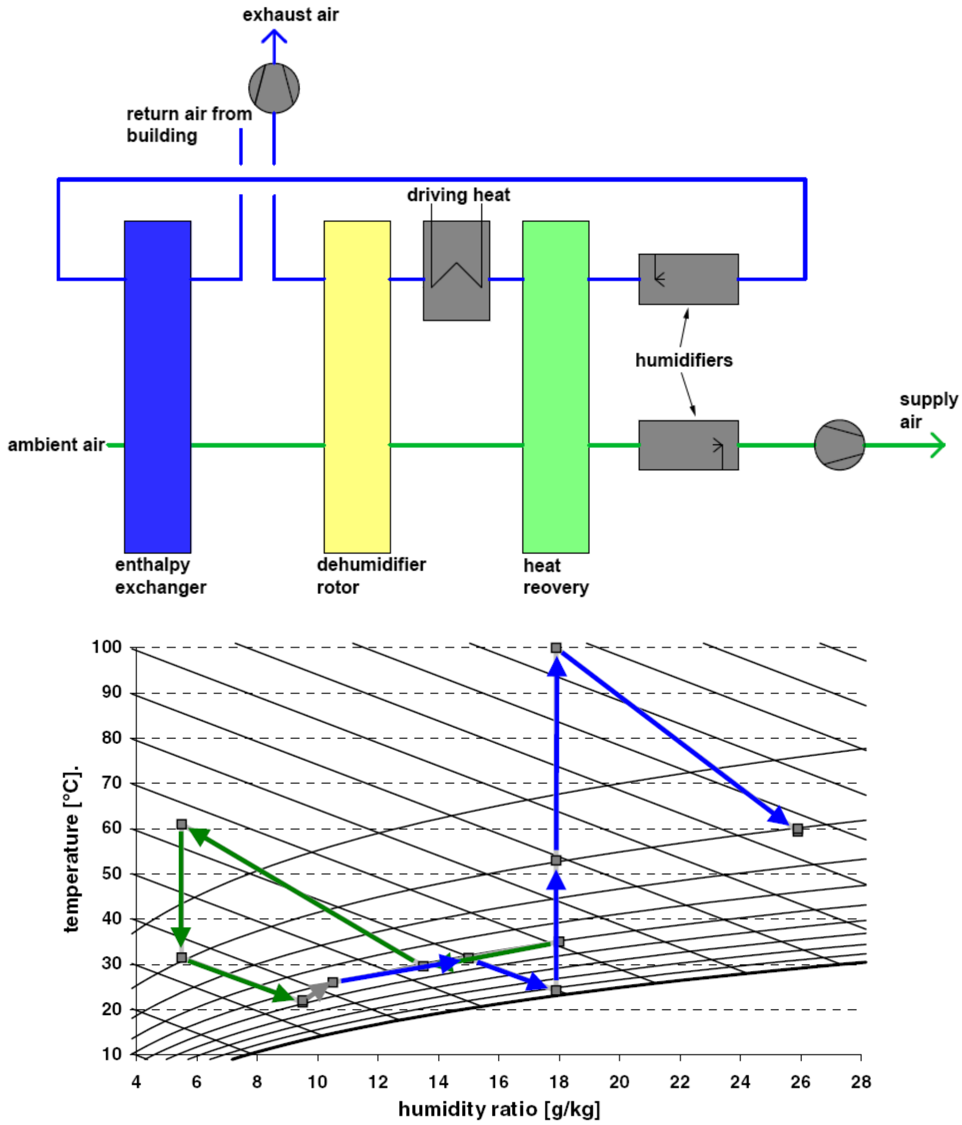


Figure 5.12 Desiccant cooling cycle for climates with higher humidity (e.g. Mediterranean); change of the humid air states in T–x diagram during the conditioning process [21].

Today, three types of solar desiccant cooling systems are most commonly used: solar-assisted and thermally autonomous, solar-assisted with back-up heating devices and solar-assisted in combination with a cold back-up (typically a vapour compression chiller).

Therefore, typical system configurations are:

- Solar-assisted and thermally autonomous desiccant cooling system with solar air collectors integrated
- Solar-assisted desiccant cooling system with solar liquid collectors, heat storage unit and back-up heat source
- Solar-assisted desiccant cooling system with solar liquid (or air) collectors and back-up electricity driven compression machine (heat pump or chiller)

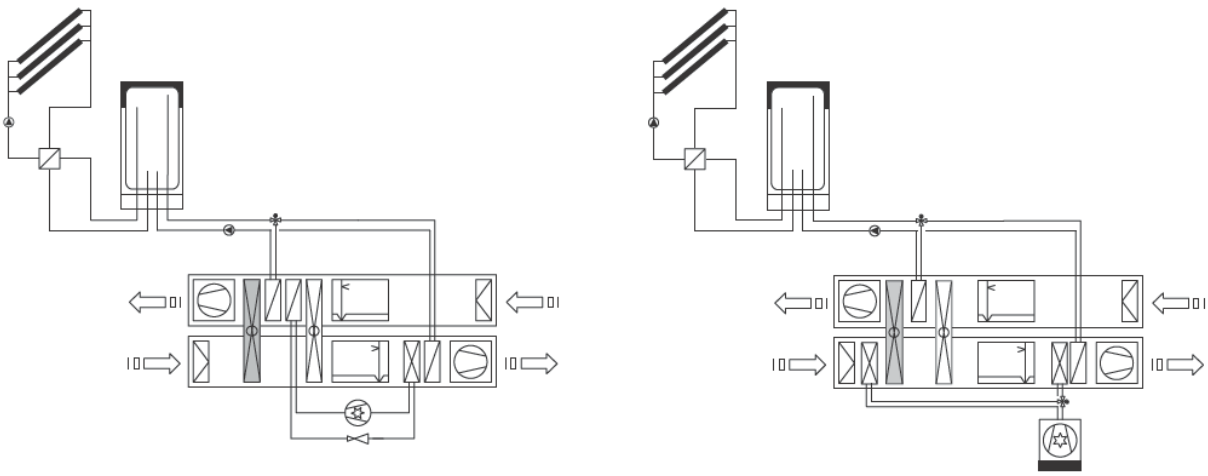
In solar desiccant systems solar liquid collectors are most frequently employed, but solar air collectors can be used as well. Liquid collectors are normally used with the storage tank. In systems using air collectors, either the air-handling unit needs to be modified or a duct diversion is added. Since no heat storage is available, solar air collectors shall be used in applications with a direct correlation between solar radiation and cooling load, or where night ventilation can be used for precooling of building thermal masses.

In systems with air collectors, energy consumption of the fans is often much higher than in water based systems, and therefore should be carefully determined at the design stage. Desiccant cooling system thermal COP_h depends strongly on the ambient air, supply air and return air. Depending on the ambient air conditions, the COP_h varies between 0.5 to 1. The higher are the regeneration temperature and the dehumidification required, the lower is the COP_h. The cooling power normally ranges from 5–6 kW per 1000 m^3/h of conditioned air.

Systems with desiccant air-conditioning shall have good control, which is crucial for economical operation in all regimes - air conditioning (cooling & dehumidification) in summer, heating and humidification in winter, and ventilation in periods without heating or cooling loads (e.g. during intermediate seasons).

In desiccant cooling systems with back up heater, it is desirable to control the water temperature to the regeneration air heater according to the current cooling load. In this way, the temperature in the solar storage tank volume can be kept lower in periods with lower cooling load, which also entails higher efficiency of solar collectors operation. Hence, such control, although more complex, increases the overall system efficiency, especially during the part load periods. Also, variable-speed fans should be used for better match of cooling loads (i.e. indoor temperature/humidity set values) and saving of driving energy for fans, if the system is designed to cover the maximum load. In case that the other air-conditioning systems are installed, i.e. desiccant system covers only a part of the load, fixed-speed fans can be used instead. Pressure drop in a standard desiccant air conditioning units (AHUs) is much higher (up to 2.5 times) compared to that in a conventional AHUs, due to presence of more components through which the air passes (the main additional pressure drop occurs in the desiccant wheel).

The examples of solar desiccant systems with conventional compression chiller as back-up and with liquid collectors are shown in Figs 5.13a,b. In one arrangement (Fig 5.13a), the compression chiller is used to cool the supply air and to heat up the exhaust regeneration air, acting therefore as a heat pump, which leads to the high overall heat recovery rate. In the second example (see Figure 5.13b), conventional compression chillers connected to cooling coils on the supply air side. Beside previously shown, a number of different air handling units can be arranged by combining different types of heat exchangers, humidification, free cooling and regeneration processes and devices.



a) b)

Figure 5.13 Examples of solar-assisted desiccant cooling system with conventional vapour compression chiller and liquid collectors: a) heat pump cools supply and heat up return regeneration air, b) chiller cools supply air [13].

5.6 Heat rejection

As shown before, chiller performances are greatly affected by the absorber/adsorber and condenser heat rejection temperature. The heat to be rejected is the sum of heat removed at evaporator and driving heat. Depending on the application, this heat is rejected mostly to the environment at temperatures between 24 - 40 °C, which is too low temperature level to be utilized directly for e.g. domestic water heating or swimming pool heating. In some cases the heat rejection temperatures can be as high as 65 °C (ammonia-water absorption chillers) and is hence suitable for the mentioned utilization.

The heat from a thermally driven chiller can be rejected to:

- Air via dry cooler, wet cooling tower, hybrid cooler- the most common
- Ground via vertical borehole or horizontal heat exchanger
- Water (ground water, river, lake, sea or swimming pool)

Wet cooling towers are based on cooling by evaporation of water, Fig. 5.14. They are available in open or closed design. Wet cooling towers can cool down the cooling water to a temperature below that of the ambient. They are compact and have low investment costs. On the other hand, the operation costs are high because of the water consumption. Also, there are hygienic issues due to fouling of the heat transfer surfaces by biological material and dust from the air stream passing by.

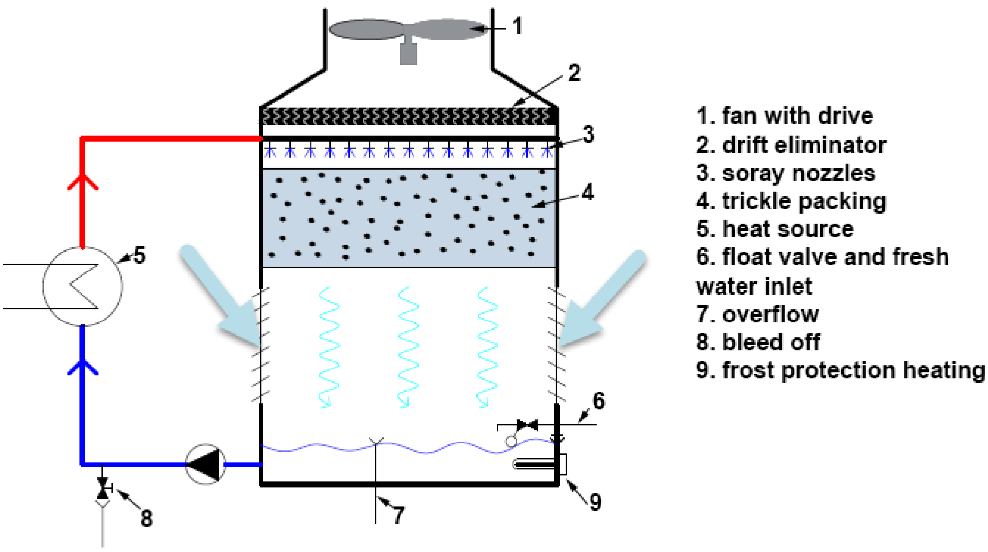


Figure 5.14 Open cooling tower [13].

This problem does not exist in closed wet cooling towers where the cooling water flows in pipes and therefore is being cooled by the water that evaporates outside the pipes, Fig. 5.15. The temperature of cooling water is higher than in open cooling towers, but lower than in dry coolers. The investment costs are higher due to more complex design, but the running and maintenance costs are lower.

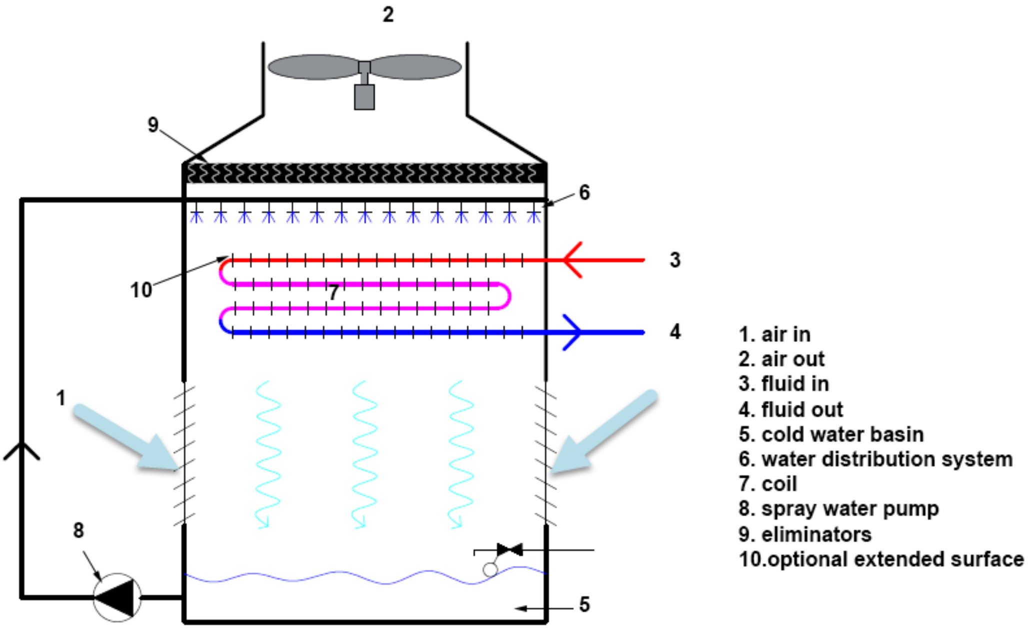


Figure 5.15 Closed cooling tower [13].

In dry coolers (Fig. 5.16), the cooling water cannot be cooled below the ambient temperature (higher heat rejection temperatures negatively affect chiller performances). They do not have hygienic issues as cooling water is separated in heat exchanger from the air. Also, the noise is low and installation simple. The operational and maintenance costs are low. On the other hand, the investment costs are much higher compared to wet cooling towers. The same applies to the energy consumption (mostly for the fan).

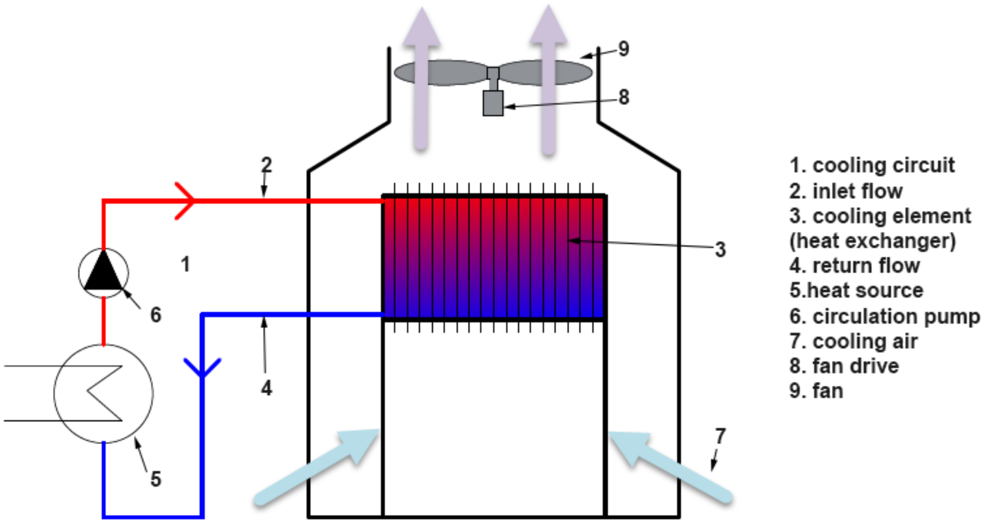


Figure 5.16 Dry cooler [13]

Size and cost of heat rejection devices, as well as electricity consumption for auxiliary equipment such are pumps and fans is considerably higher than in conventional mechanical vapour compression systems. This is due to much smaller COP_h of thermally driven cooling cycle and therefore higher (2 - 3 times) amount of reject heat for a given cooling capacity. As the heat rejection related cost for electricity consumption represent more than 60% of total electricity consumption, these costs need to be carefully assessed at the planning and design stage.

The hybrid cooler combines dry cooling and evaporative one used in cooling tower. The cooling water flows through a cross current air to water heat exchanger. Hybrid dry cooler can reach cooling water temperatures below the ambient air temperature. Also, it has higher capacity and lower energy consumption for fans compared to dry cooler. However, the investment costs are higher, as well as are the water consumption and maintenance costs, since there are hygienic issues present (like in the case of the wet cooling tower). Boreholes are ground coupled heat exchangers normally drilled to a depth of 100 m. Since they are more expensive than cooling towers or dry coolers, boreholes can only be interesting solution in cases where the surface area is limited. A horizontal ground heat exchanger is installed at depth of 0.5 to 2 m. This is also more expensive solution compared to cooling towers or dry coolers.

5.7 Cold storage

Cold storage is often part of cooling systems where chilled water is used as a working media. In a solar-assisted air-conditioning system, use of cold storage allows generation of more cooling power than needed during periods of high solar gains. The stored cold is then used during periods of low radiation when solar thermally driven chiller is not capable of meeting the current cooling demand.

The size of the storage tank depends on the following possible situations:

- Drops of the solar radiation due to clouds in periods of few minutes. In such cases, the cold storage size is smaller as it should cover the differences in load and generation only for a short time.
- The cooling load is not in phase with solar radiation, i.e. load pattern has peak in the afternoon or evening, while solar radiation at noon. In such cases, the storage system has to be largely sized to be able to cover the differences in load for several hours.

The main types of storages used are:

- Sensible heat storages (content-chilled water)
- Latent heat storages (content-ice, water salt solutions or other phase change materials(PCM) like e.g. paraffin)

Temperature change of sensible cold storages media is relatively small, so to store a unit cold, more volume (mass) is needed then in the case of hot water storages. In this regard, use of latent heat storages enables reduction of storages volume. Instead of storing driving heat, storing cold in a solar-assisted air-conditioning system has some advantages:

- The amount of cold to be stored to cover a unit of load is lower than it would be the amount of driving heat, when the COP_h< 1.0 (normally for all single stage thermally driven chillers)
- Lower are storage tank heat losses due to smaller temperature difference storage media-surrounding air, at same insulation thickness

On the other hand, the insulation thickness should prevent condensation on the tank walls. This also applies on pipes and their connections to the tank.

6. Design and sizing guidelines

In order to wide-spread use of solar cooling systems, a set of well explored system configurations need to be defined to enable easy selection of an appropriate system configuration for a certain application (cooling, DHW and space heating), type of building and demand. Such compact systems can be then installed by professionals from the heating and plumbing sector and there is no need for detailed planning procedure from case to case. Basic solar cooling system configurations are already presented in Chapter 5 with division on three sub-systems:

- solar thermal system with back-up heater
- cooling water loop
- chilled water loop

Apart from the previously shown configurations, (solar system + back up heater), during heating season winter period thermally driven chiller can serve as a heat pump delivering a low-grade heat from condenser and absorber for space heating (Figure 6.1). Heat source to the evaporator of this heat pump can be e.g. a ground water well, which can be in cooling mode utilized as a backup cooler.

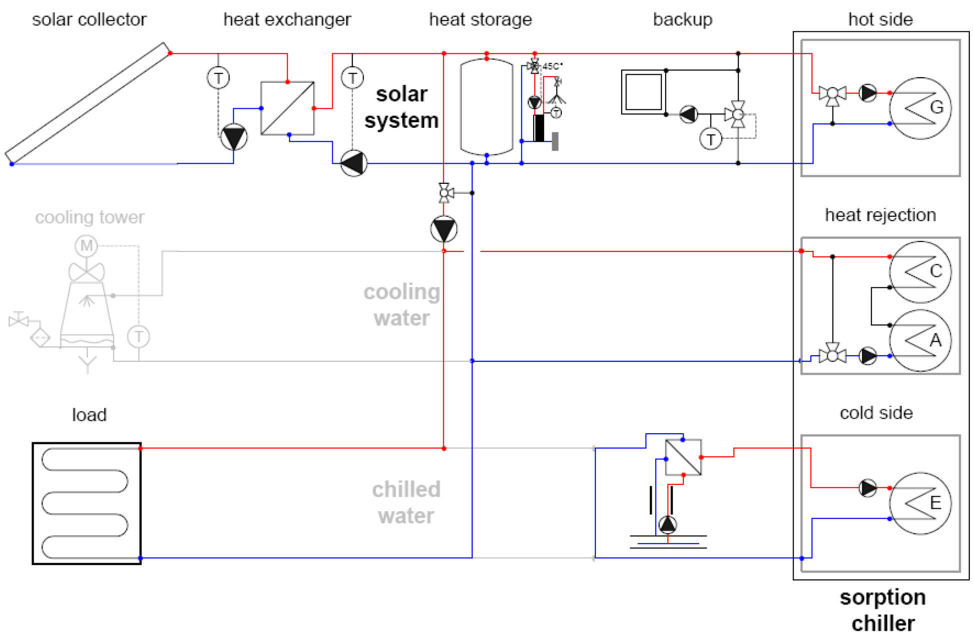


Figure 6.1 Thermally driven chiller operation in heat pump mode during heating season [20]

However, all three sub-systems can be further combined in a number of different configurations and sizes, depending on heating/cooling/DHW demand, climate, room-side appliances for heating and cooling.

In the following sections, selection and sizing guide for different components is provided.

6.1 Selection and sizing of solar collector area and chiller capacity

Size of solar collector area depends on whether the system is regarded as:

- solar autonomous cooling system or
- solar assisted (heating and) cooling system.

Solar autonomous cooling system has no back-up for additional cold production if available solar energy is not sufficient. Such systems are used when a certain indoor discomfort is acceptable, since the cooling demand will not be covered for a certain number of hours in a season and a set indoor temperature and humidity do not need to be assured all the time. In this case, a dynamic simulation of the building demand through a season(s) is performed in order to optimize the system components size. This optimization is important, as an oversized collector area and chiller capacity for too many hours entail lower overall system efficiency, as the system is likely to work under part load conditions at these periods. Solar assisted (heating and) cooling system do have a back-up for additional cold production if available solar energy is not sufficient. Then two cases can be distinguished:

a) when the back-up heater is installed within the solar system, the thermally driven chiller has to match cooling loads at design conditions. The back-up heater can be of lower capacity (power) if it is connected to the storage tank.
b) when the back-up is the vapour compression chiller, it has to be sized in the way it can cover the full cooling load together with solar driven chiller in periods of minimal solar radiation.

Since in solar assisted systems a back-up in combination with storage tanks can balance the cooling load not covered by solar driven chiller, designing process is more flexible. So, more parameters for collector area sizing can be included, beside the building energy needs. Such typical parameter is e.g. the available roof area for the installation of solar collectors.

In general, a proper sizing of the single components is crucial to achieve a high system COP_h. During designing process, the overall demand and its variation within considered periods shall be determined firstly. The diagram on Fig. 6.2 may serve for first selection of the system type in regard of the available driving temperature. As shown, the COP_h increases with the driving temperature. The other parameters to be considered are the system cost and required collector area per unit of installed chiller cooling capacity, available from Fig. 6.3. As can be seen, the H_2O/NH_3 systems require larger specific collector areas than $LiBr/H_2O$ or desiccant systems. As a result, the installations usually are more expensive relative to $LiBr/H_2O$ systems and only slightly expensive relative to desiccant systems. Based on these diagrams, the average specific solar collector area is 3.6 m²/kW, ranging from 0.5 to 5.5 m²/kW**.** Needed collector area in adsorption and absorption systems is between 2 m²/kW and 5 m²/kW. The average initial (investment) overall cost is 4000 €/kW, excluding the cost for distribution networks between the system and the application and the delivery units.

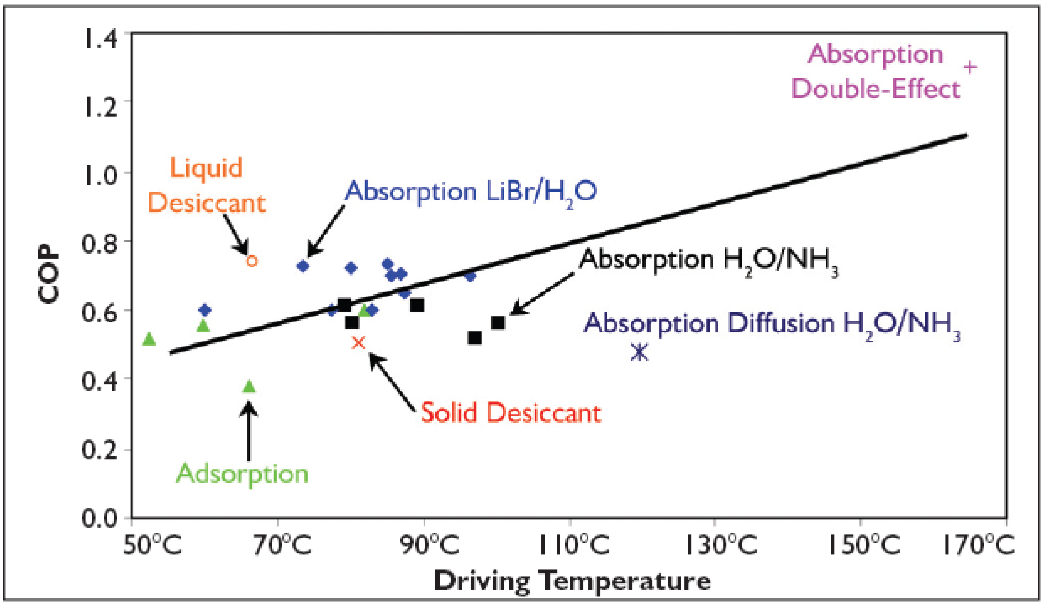


Figure 6.2 (COP) of different systems installations as a function of driving temperature [24].

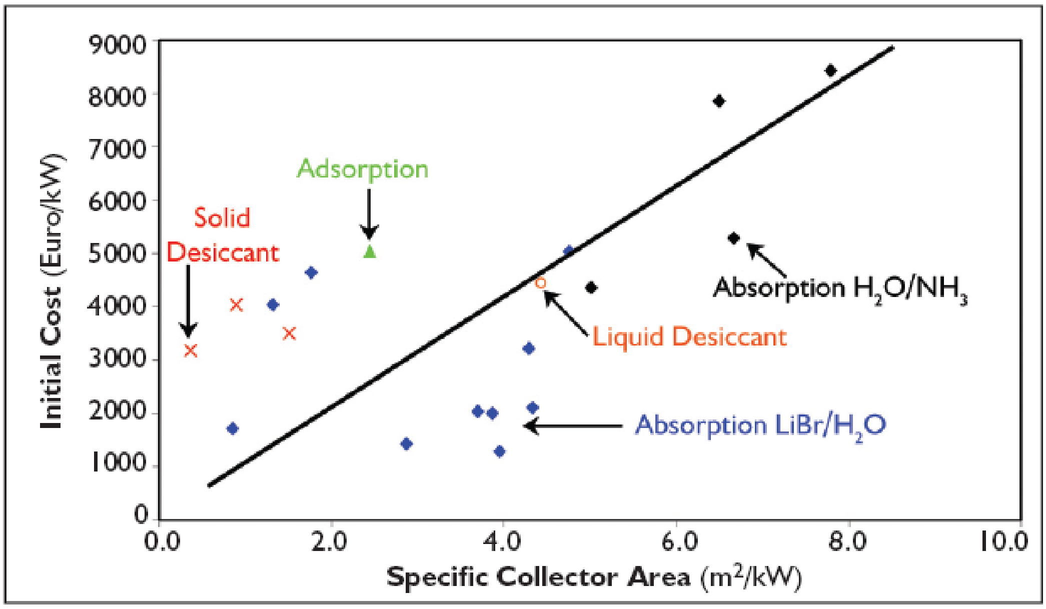


Figure 6.3 Initial (investment) overall cost for different systems in dependence of the specific collector area [24].

For systems operating at the driving temperatures of 60°C to 90°C, i.e. desiccant cooling systems, adsorption chillers and possibly single-stage absorption chillers, flat plate solar collectors are first option because of their low initial cost. Due to relatively high operating temperatures for this type of collectors, only highly efficient solar collectors with a selective absorber coating should to installed, in order to avoid excessive drop in their efficiency.

Evacuated tube solar collectors are used in systems operating at the driving temperatures of 80°C to 120°C. Compound parabolic concentrators can reach up to 165°C which makes them suitable for use in two-stage chillers. Detailed information on characteristics of different thermally driven systems is given in Table 6.1 along with driving temperatures and solar collectors types.

Table 6.1 Characteristics of commercially available thermally driven cooling systems

Type of system	**Water chillers(closed thermodynamic cycles)**							**Direct air treatment(open thermodynamic cycles)**	
Physical phase of sorption material	Liquid				Solid			Liquid	Solid
Sorption material	Water	Lithium-bromide			Zeolite	Silica gel	Lithium-chloride	Lithium-chloride	Silica gel, cellulose/lithium chloride
Refrigerant	Ammonia	Water	Water	Water	Water	Water	Water	Water	Water
Type of cycle	1-effect	1-effect	2-effect	3-effect	1-effect	1-effect	1-effect	Cooled sorption process	Desiccant rotor
COP range	0.5 - 0.75	0.65 - 0.8	1.1 - 1.4	1.6 - 1.8	0.5 - 0.75	0.5 -0.75	0.5 - 0.75	0.7 - 1,1	0.6 - 0.8
Driving temperature range	70 - 100°C	70 - 100°C	140 - 180°C	200 - 250°C	65 - 90°C	65 - 90°C	65 - 90°C	60 - 85°C	60 - 85°C
Solar collector technology	FPC, ETC, SAT	FPC, ETC	SAT	SAT	FPC, ETC	FPC, ETC	FPC, ETC	FPC, ETC, SAHC	FPC, ETC, SAHC

The average annual efficiency of the solar collectors in monitored systems varied between 12% to 32% [13] (see also Figure 3.14)

The following annually averaged values of required collector area per kW of installed chiller cooling capacity can be used at design stage for selection of the type of system:

- 3.4 m² /kW for flat plate type collectors
- 2.6 m² /kW for evacuated collectors

The required collector area for a solar cooling system can be obtained from:

$$A_c = \frac{1}{G \cdot \eta_c \cdot COP_h} \quad [\text{m}^2/\text{kW}]$$

where:

A_c (m^2/kW) is the specific collector area per installed kW of chiller cooling capacity

G (kW/m^2) is the incident solar radiation at design condition

η_c (–) is the collector efficiency at design driving temperature

COP_h (–) is the thermal COP_h of chiller at design conditions

6.2 Size of the hot storage volume

The hot water storage volume can be sized as 75 Lit / kW of installed chiller cooling capacity and 55 Lit/m^2 of collector area. The volume of hot storage also depends on whether a cold storage tank is installed to cover difference between solar radiation and cooling load or not. In case a cold storage is installed, cold water can be produced when an excessive solar energy is available relative to the current load and stored in a cold storage tank for later use. Such approach has not been applied in many installations by date, but it is recommended to be considered and evaluated at design stage. The size of the tank also depend on application i.e. usage profile. For example, in applications in tourist sector, such are hotels or apartments, the main DHW and cooling demand usually are shifted from mid-day to the late afternoon and evening hours. The storage tank has to be sized to accommodate all the solar energy collected during the mid-day which was not utilized at that time for use several hours later. The office buildings are the opposite example, where the peak load occurs in the morning and early afternoon hours.

Since the needed driving temperatures for operation of the thermally driven chiller are relatively high compared to the temperatures in storage tanks of conventional solar thermal systems, direct solar energy utilization is here preferred (avoid use of heat exchangers or mixing). Figure 6.4 shows the acceptable options for connecting hot water storage to solar loop for purpose of solar heat storage and domestic hot water preparation. The following solar tank solutions should be avoided:

- integrated heat exchanger
- tank-in-tank configuration (as 60°C is limit to avoid scaling and calcification)

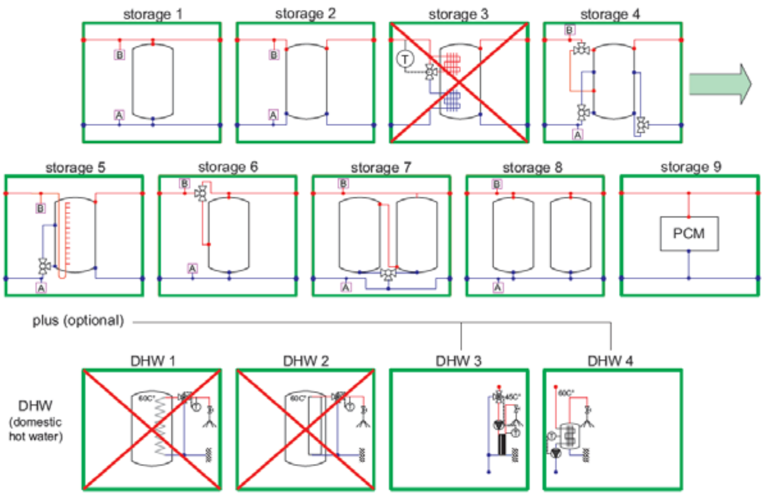


Figure 6.4 Hot water storage tank configuration for storing solar heat and DHW preparation [20].

6.3 Back-up heater/chiller

During periods of insufficient solar radiation solar system is backed up by heater in the same way as done in conventional solar heating systems. The back-up heater should provide driving heat for thermally driven chiller to assure production of cold independently from the currently available solar system output. Back-up heater can be either directly connected to the hot water loop leading to the chiller hot side, or it can be connected to the heat storage tank. From there, the hot water is further distributed to chiller hot side and other consumers. Back-up chiller(vapour compression chiller) is better option than back-up heater in cases when the most of cooling energy is to be delivered without solar energy input, i.e. in the evening and when solar cooling system serves only for cooling. This is so due to lower consumption of primary energy when compression chiller is used (thanks to its high COP) instead of conventional heater on fossil fuels. When the thermally driven chiller does not cover current demand, the vapour compression chiller is in operation until the desired temperature of chilled water supply is obtained - serial mode, or it provides additional cooling power at the given temperature difference -parallel mode. An alternative to using vapour compression chiller as a backup-chiller is a ground water well, a borehole or a ground heat exchanger.

6.4 Chilled water sub-system

Air-handling units (“AHU”) or fan coils are standard equipment that can be used for cooling and dehumidification of the indoor air. If sensible cooling can satisfy indoor climate requirements, higher chilled water temperatures should be used as this significantly lowers energy consumption of the chiller. In these cases, radiative cooling surfaces, like activated ceilings or floor heating/cooling are used. As mentioned before, during winter time the ground water well may be used as a geothermal heat source for the system operating in a heat pump mode.

6.5 Control

When solar combined-system intended for space heating and domestic hot water production during heating season is then used in cooling season for providing heat to the thermally driven chiller, fluid flow rate in collectors should be increased in order to avoid an excessive increase of temperatures in collectors and consequent drop in their efficiency as well as a risk of boiling of working fluid (and fatal crystallization in case of water-glycol mixture). This can be done by changing the pump setting (e.g. switching to higher pump speed mode) or by frequency speed control of the pump. In general, in solar cooling systems the solar collector loop should operate with a limited temperature lift and with a high volume flow of the working fluid. The chilled water supply temperature is a main controlled variable in the chiller output capacity control. The chiller capacity is controlled by mixing recirculating water leaving the generator with the hot water from the heat source (solar system). Also, the capacity can be controlled by a speed control of the hot water pump or by means of a two-way regulating valve. Chilled water return temperature should be controlled to a fixed value, as it entails an increase of chilled water supply temperature when the cooling load is decreased. This in turns allows lower driving temperatures (higher solar system efficiency) and reduced back-up heater or cooler energy consumption.

6.6 Auxiliary power demand

High efficient pumps with variable speed control should be used in order to reduce system auxiliary energy consumption. Also, control setup should prevent circulation in the solar loop in periods with too low solar radiation. Blower fan of the heat rejection, especially the one of the dry cooler, accounts for the most of the overall electricity consumption. The fan speed should be for this reason controlled in dependence of the cooling demand, enabling also the solar system to operate independently on the cooling demand.

The electricity consumption for auxiliary equipment (fans and pumps) can be roughly estimated as 225 W/kW cooling capacity.

6.7 Design procedure

During the design process of solar cooling system, the following issues are to be considered in order to optimize the system:

- technical features of the thermally driven chiller, solar system, heat rejection system, back-up and air conditioning equipment
- -investment and running costs for the system components
- -coupling of the solar system to the back up and thermally driven chiller
- delivered and primary energy savings
- maintenance

The procedure consists of the following steps (Table 6.2):

1. **Analysis of suitability of a considered building for solar cooling.** An assessment of the project's feasibility can be preliminary carried out using "**Check list**" method (Fig. 6.5) developed within the IEA-SHC Task38. This approach is only qualitative and compares the considered project to a benchmark based on the previous projects.

2. **Selection of suitable thermally driven chiller and solar system (collectors) pair for the particular application**. Here, only qualitative methods can be applied, such is the ‘’decision scheme’’ from Fig. 6.6. It allows for first selection of the most appropriate pair in regard to e.g. generation temperature, typology of distribution system, (Fig. 6.7), etc.

3. **Determination of the solar collector area and size of other major system components taking into account energy performances and investment cost.** At this stage, detailed computer simulations are performed with predetermined load profiles as an input. Unlike conventional HVAC systems, solar cooling systems are designed according to desired solar fraction. The simulations should, therefore, provide information on the annual delivered and primary energy consumption of the whole system.

4. **Cost analysis.** The last step is the initial (investment), operation and maintenance costs calculation, as well as calculation of savings (in terms of energy and lower costs)

Table 6.2 Main steps of the predesign methodology developed within the IEA-SHC Task 38 [20].

Step	Kind of method	Method	Purpose
1.	Qualitative	Check list (questionnaire)	Analyse suitability of using solar heating/cooling for a given application/building
2.	Qualitative	Decision free	Select the appropriate cooling/air-conditioning technology
3.	Quantitative	Annual simulation (energy balance)	Select and size the solar technology; select and size other key components
4.	Quantitative	Cost analysis (spreadsheet based on energy balance)	Calculate cost for selected system designs (investment cost, life cycle cost, ...)

Technical feasibility

To build-in a check list, the first topic is naturally the building, which is the target of the solar cooling system and its technical features

Please select an answer in the list for each quesiton.
If you do not know, let the answer empty.

		Help	Answer
Building	Climate of location	?	
	Area for solar collectors (R=building surface/available solar surface)	?	
	Space available for the technical premises	?	
	Heating and cooling distribution network adapted	?	
	Existing or planned adapted conventional heating/cooling system	?	
	Bio-climatic cooling solutions planned or installed	?	
	Are you planning to call for an installer and a engineering office with good solar cooling experience?	?	
Load	Correlation between daily production and thermal load	?	
	Correlation between yearly production and thermal load	?	
	Energy needs (cooling, hot water and heating) all year long	?	
	If air conditioning is planned. Is the solar system assisted with a back up?	?	

Figure 6.5 Check list method – spreadsheet for evaluation of “Technical feasibility”, IEA-SHC Task 38 [20].

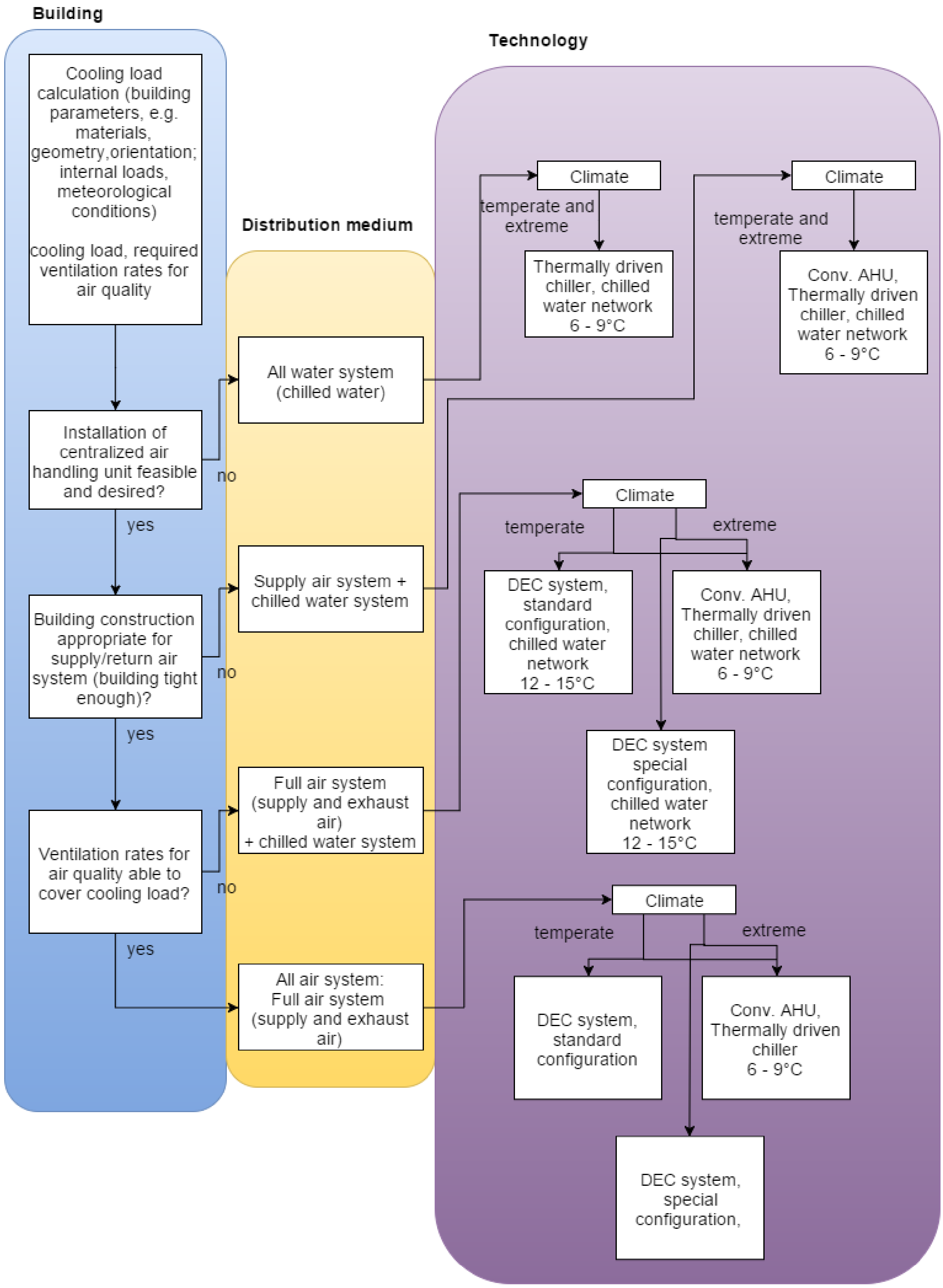


Figure 6.6 Decision scheme for selection of suitable thermally driven chiller and solar system pair, Tasks 25 and 28 of the IEA-SHC [13].

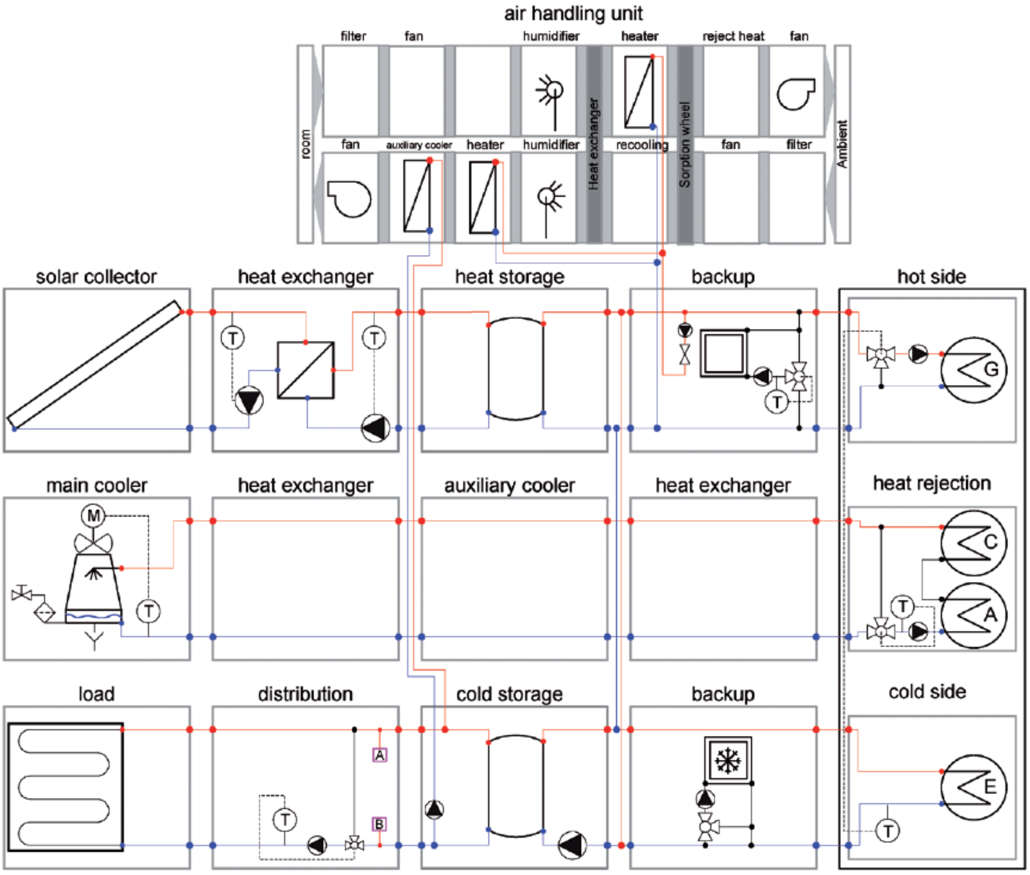


Figure 6.7 Example of a complete system scheme (desiccant system and thermally driven water chiller) [13].

A few simple pre-design software tools were developed within different EU Projects and are available for free download, Table 6.3.

Table 6.3 Examples of simple pre-design software tools

Software	Source
SACE - Solar Cooling Light Computer Tool	**http://www.solair-project.eu/218.0.html**
ODIRSOL	**Software available on demand by writing to TECSOL at info@tecsol.fr**

More detailed simulation tools are listed in Table 6.4. They enable transient simulation of solar heating and cooling systems and the building heat/cooling load

Table 6.4 Examples of detailed simulation software tools

Software	Source
TRANSOL 3.0	**http://www.aiguasol.coop/**
EnergyPlus	**http://www.eere.energy.gov/buildings/energyplus/**
PolySun	**http://www.velasolaris.com**
INSEL	**http://www.inseldi.com/index.php?id=21&L=1**

6.8 Economic aspects

An important part of designing process is an evaluation of economic viability of the particular solution considered. Corresponding cost calculations are rather complex and involve comparison with conventional heating and cooling systems.

The following factors should be taken into account:

- Climate data (solar radiation, temperatures, humidity)
- Technical specification of HVAC system (efficiency, energy consumption)
- Investment costs for the HVAC system
- Investment costs for solar system including additional costs for installation of solar panels on the building
- Operation (energy) costs including maintenance
- Energy price (growth) and interest rate prediction during payback period

The initial (investment) cost of solar cooling systems comprising planning, equipment and installation work is 2 - 5 times higher compared to a standard system, depending on climate, building load, system type and size etc. The average overall solar cooling systems initial cost are typically within the range 2000 - 5000 €/kW of installed cooling capacity.

The operational cost savings depend on the following conditions that shorten the payback time period:

1. High solar radiation – increases the capacity and efficiency of the solar system
2. Longer cooling season means more hours of the system use - higher cost savings per year
3. Use of the system for DHW production and/or space heating - increases the overall system efficiency
4. Higher prices of energy used in conventional system - higher cost savings when using free solar energy

In spite the fact that solar cooling systems have high initial cost relative to the conventional systems, their overall life cycle cost are lower in most cases. This means that solar cooling systems have shorter payback period (in some cases even < 10 years) than is their life time. In addition, an increasing trend of conventional energy prices is expected to further shorten payback period than can be estimated at the design stage. In order to assess economic viability of a solar cooling system, it is very important to determine the initial (investment) cost. A structure of initial costs for a desiccant and an absorption cooling system with flat plate solar collectors is provided in Figure 6.8.

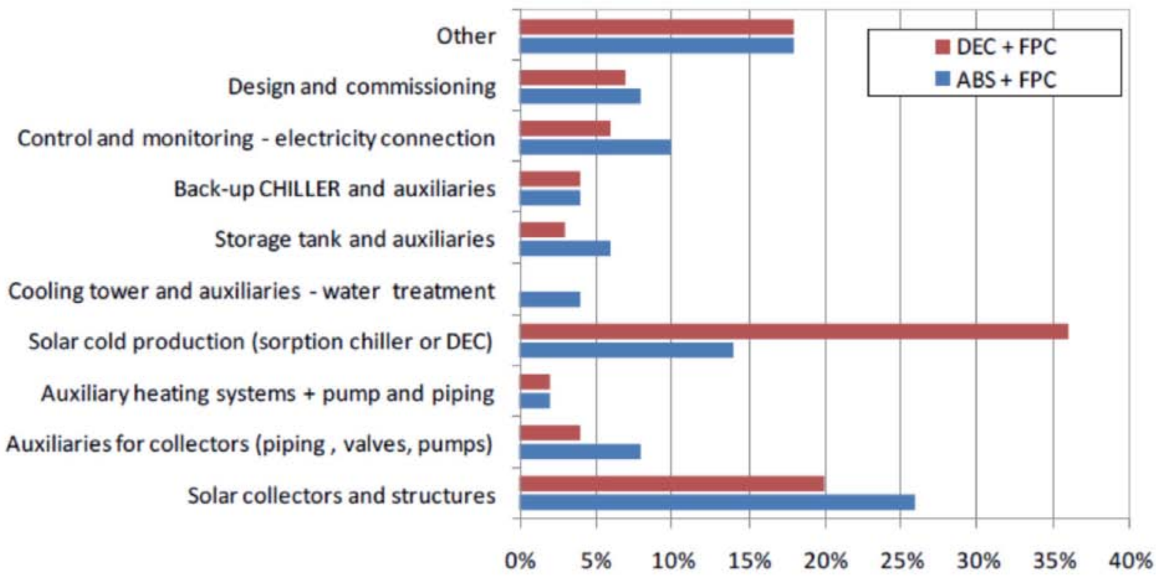


Figure 6.8 Example of structure of initial cost for desiccant and absorption cooling system equipped with flat plate solar collectors [13].

Specific cost curves for different technologies are given in Fig. 6.9 as a function of installed collector area, cooling power, air flow, etc. These curves are derived from the data collected on pilot installations during the work of IEA-SHC Task 38. As can be seen, these curves vary significantly from project to project, depending e.g. on manufacturer, installer and/or other site related conditions.

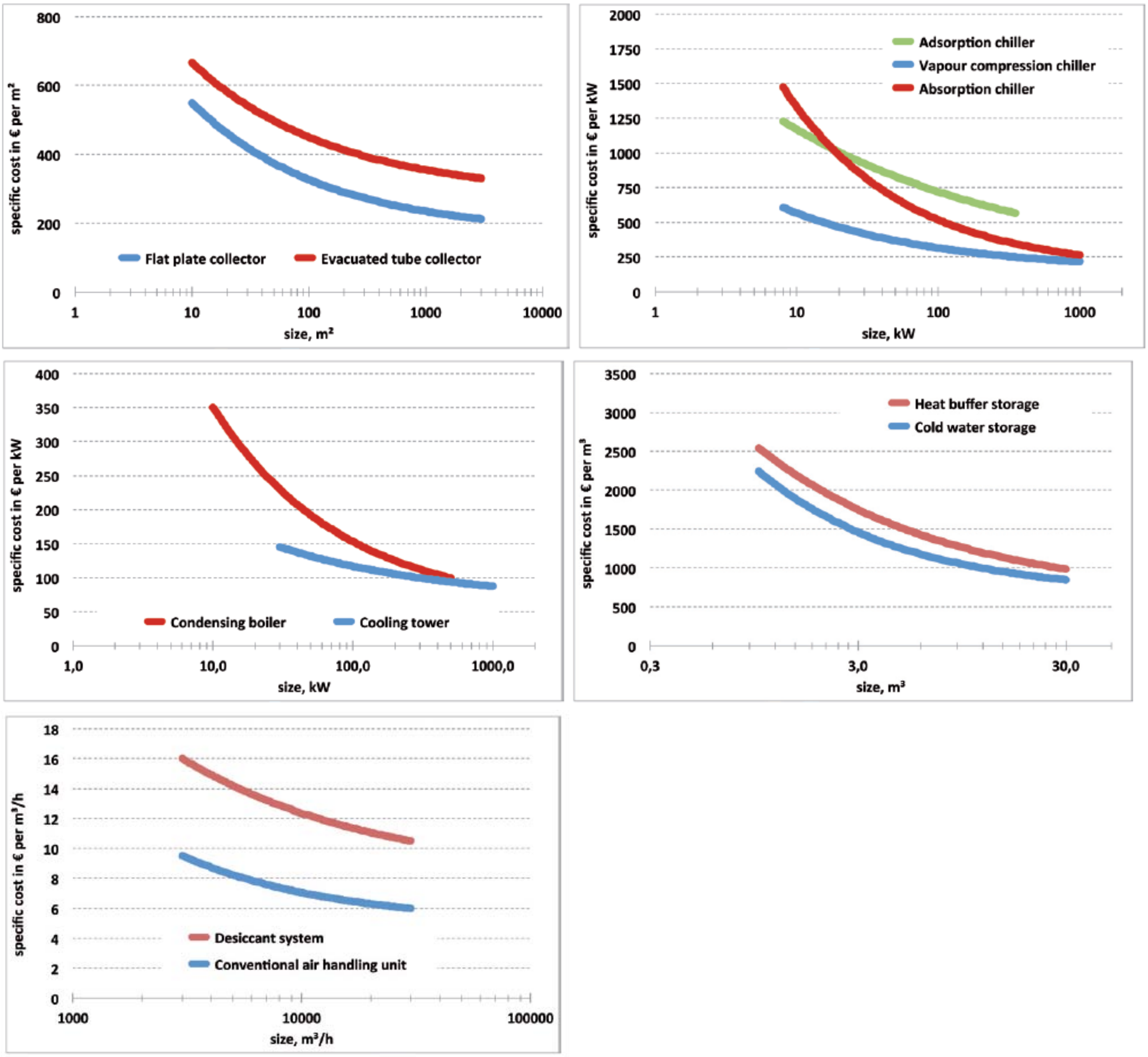


Figure 6.9 Typical cost curves (logarithmic x-axis) for different technologies and size/capacity of main components [13].

Annual operating costs comprise fuel, water and electricity costs. Maintenance costs refer to the services. These costs are used for assessment of the simple payback period, which can be useful in preliminary basic analysis, Fig. 6.10. So called 'total annual cost', where the annuity is added to the annual cost, should be used at decision stage for comparison of different solutions, Fig. 6.11.

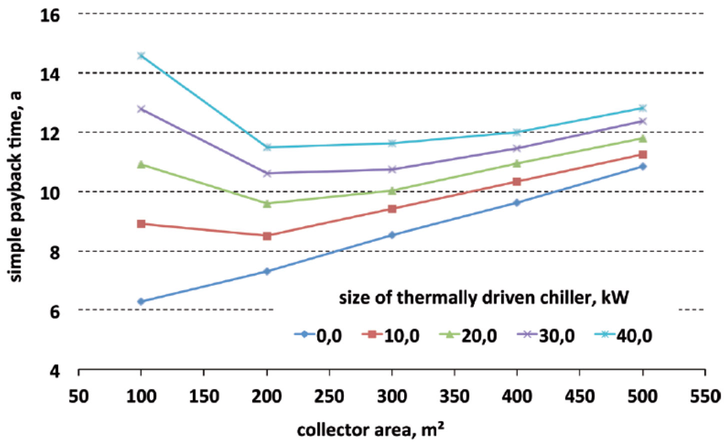


Figure 6.10 Simple payback time of solar cooling systems [13].

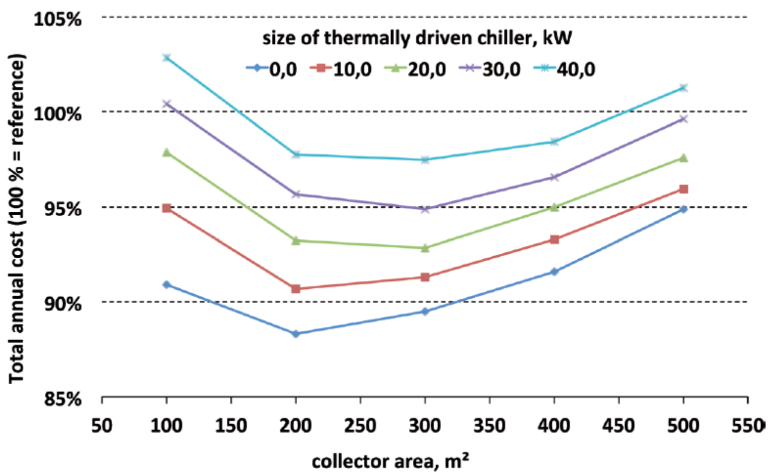


Figure 6.11 Total annual cost of solar cooling system (hotel example in Madrid), comparison with reference conventional system [13].

The solar cooling systems should also be evaluated in terms of fractional primary energy savings relative to the conventional reference system (saving/primary energy of ref. system), Fig. 6.12. Usually fractional primary saving is lower than solar fraction (collected solar energy/energy demand) due to the influence of electrical energy used by the auxiliary equipment.

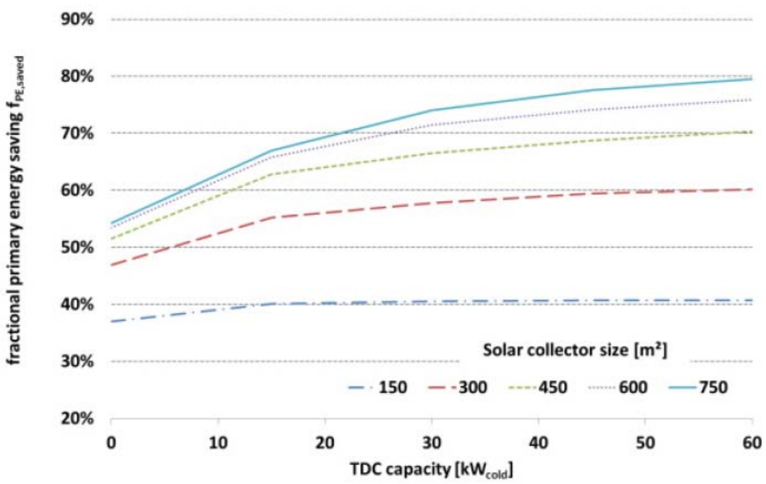


Figure 6.12 Fractional primary energy saving relative of the solar heating and cooling system to the conventional reference system (example of the hotel in Malta) [13].

The other advantages of using solar cooling systems, that cannot be directly expressed in economic figures, but should be taken into account at planning/design stage, are:

- Solar cooling systems decreases primary energy consumption, that e.g. improves energy class of building energy performance certificate
- There is related reduction in CO_2 emissions - an important target for each country energy policy
- Solar cooling cycles normally operate with refrigerants with zero ozone depletion potential and no or a very small global warming potential.
- Solar cooling systems can be combined with space heating systems, whereby considerably decrease conventional energy consumption also during heating season
- As they reduce electricity consumption during cooling season, these systems can decrease peak load occurring during daytime due to work of increasing number of conventional air-conditioning systems (good example is the Mediterranean region). This also improves grid stability in such areas.

6.9 Overview of design recommendations and guidelines

Overall, the most beneficial conditions for installing solar cooling systems are:

- climates with mild winters and summers with a high solar irradiation (insolation)
- operating conditions with a good match between loads and solar irradiation since this reduces the storage need
- high cost for application of conventional systems and energy
- good technical support and qualified services in heating and cooling systems

The potential motivation for an investor is reduced primary energy consumption of solar cooling systems, i.e. its environmental soundness, even if the system is not considered economically viable. The potential economically viable applications and potential users of solar cooling systems are:

- office buildings and hotels, solar system with non-tracking solar collectors should be used for heating, cooling and domestic hot water
- production buildings where cooling is needed for industrial processes (e.g., food industry).Collectors, depending on the driving temperature, can be non-tracking or single-axis tracking, and can be accommodated on factory roofs or installed on a ground
- family houses and all other low energy and passive buildings with low heating/cooling need and those with installed low temperature delivery systems like floor and/or the ceiling heating/cooling systems

Summarizing the design considerations from previous sections of this chapter, the following design and operation guidelines with high energy saving potential are listed:

- Before a solar cooling system is considered and sized, the building cooling load should be carefully assessed and reduced by passive cooling strategies, like reduction of solar gains by sunblind, implementation of thermal mass, night ventilation, orientation and size of windows etc.
- -In new buildings, enough space for solar collectors should be ensured at early design stages. At existing buildings, an availability of roof surface area should be checked before considering the solar system.
- In applications/climates with heating and cooling demand, combined systems (solar heating and cooling) should be considered.

- Keep systems as simple as possible in order to minimize risk of failures during installation, operation and maintenance.
- Thorough design is needed (especially for cooling capacities >20 kW) in order to optimize size of the key components and find design that suits well actual load.
- The system should be optimized in the way it runs at highest efficiency under full and part load conditions
- Comparisons/assessment of the environmental impact of solar heating and cooling systems should be done on the primary energy level and on CO_2 emissions.
- For economic reasons, the system and its components should not be sized to cover peak loads, as they account for only a few hours of the year.
- In applications where cooling load significantly fluctuates and power failure often occur, solar collector overheating should be prevented by e.g. drain back strategy
- Min. 70% cooling fraction with solar cooling systems is recommended. 100% cooling fraction is technically and economically rarely feasible. In these cases a back-up source provides cooling when the solar irradiation cannot cover the current load.
- When fossil-fuelled heat source is used as a back-up, a high solar fraction is necessary in order to assure significant primary energy savings. In order to reliably assess the solar fraction and correctly size the solar collector area and energy storage, a dynamic system simulation needs to be performed
- Use of a conventional vapour compression chiller as a back-up is particularly appropriate for large installations. It is also more preferable from a primary energy saving point of view. In this case solar cooling fraction can be kept low, whereby the solar system is used to reduce electric energy consumption as well as peak electricity loads.
- Autonomous solar systems without back-up for cooling can be implemented in building of a large thermal inertia (thermally activated concrete slabs or storage rooms in food and agro industry)
- Solar collector area should be sized in the way it can be exploited within a whole year and during a whole day, i.e. for cooling and domestic hot water production (or swimming pool heating) in summer period and eventually for space heating/domestic hot water purposes in winter period, especially in regions with mild winters and high solar irradiation throughout the year, as it is case in the Mediterranean.

- In heating season thermally driven chiller can be used as a heat pump for provision of low-grade heat for space heating using heat from condenser and absorber
- In cases when absorber and condenser heat is rejected at elevated temperatures, (e.g. ammonia/water chillers up to 65°C), it can be considered for domestic water preparation or heating of swimming pools.
- For solar cooling only highly efficient collectors with a selective absorber coating should be employed.
- When choosing a specific collector type (hot water, air), the electricity consumption of the auxiliary devices as fans and pumps needs to be assessed, as they have huge impacts on operation costs and primary energy consumption. Therefore, all auxiliary devices should be highly energy-efficient.
- Solar heating and cooling systems with a cooling capacity <20 kW should be pre-defined by the manufacturer (if available) in terms of the heating/cooling power, system layout and the size of all components. No detailed planning process is necessary to install such a system for a given application and load.
- For the Mediterranean, solar cooling systems can yield to 40 - 50% decrease of primary energy. Related cost of saved primary energy is about 0.07 €/kWh.
- Small cooling capacities and tracking solar collectors for two stage systems increase the cost significantly.
- The cost of different system designs can be evaluated by a tool "Easy Solar Cooling" (http://www.solairproject.eu/218.0.html). It enables user to identify the most appropriate solutions for a given operating conditions.
- A careful commissioning of the system is important to ensure system future operation according to the project conditions
- Further continuous monitoring is recommended in order to assure operation at highest performance.

7. Installed systems - examples

This chapter shows some examples of existing small-scale (pre-engineered kits <30 kW) and large scale (custom made) solar cooling and heating systems. Few examples of thermal driven chillers are provided on Figs. 7.1 - 7.3.

a) b)

Figure 7.1 Examples of absorption chillers [13]: a)10 kW single-stage water/ammonia chiller, Source: Pink, b) 100 kW single-stage LiBr/water chiller, Source: Thermax

a) b) c)

Figure 7.2 Examples of adsorption chillers [13]: a) 8 kW single stage water/zeolite chiller, Source: InvenSor, b) 15 kW single-stage water/silica gel chiller, Source: SorTech, c) 350 kW single stage water/silica gel chiller, Source: Mayekawa

The project is co-funded by the European Union Instrument for Pre-Accession Assistance

Figure 7.3 Desiccant air-handling unit equipped with heat pump (cools supply and heat up return regenerating air) and two additional heat exchanger coils (nominal air flow 1500 m3/h), Dep. Energy, University of Palermo, Italy [13] and [25].

7.1 Pre-engineered kits

Recently, pre-engineered systems are introduced on the market, primarily in the EU as compact standardized kits, which are simple, easy to install and have low initial cost. They consist of solar kit (solar collectors with all necessary connectors, hot water storage tank, pump, manometer) thermally driven chiller (absorption or adsorption), a heat rejection and a control unit. The cold water storage can also be part of the kits.

Table 7.1 Examples of commercially available solar cooling kits up to 100 kW

Company	Product name	Chiller technology	Offered cooling capacities [kW]
SolarNext (Germany)	chillii® Cooling Kit	Absorption and Adsorption	8, 9,, 10 ,11, 15, 17.5, 19, 30, 35, 50, 70 and 105
Schüco (Germany)	LB Cooling System	Absorption	15 and 30
Pink (Austria)	Cooling Kit	Absorption and Adsorption	8, 14, 15, 19, 30 and 54
EDF Optimal Solutions (France)	Package Clim'solaire	Absorption	35, 70 and 105
INNOPTIM (France)	Climatisation solaire	Adsorption	8 and 15
Kloben (Italy)	SOLARTIK	Absorption	17.5, 35, 70 and 105

Example 1:
10 kW absorption chiller in office building in Garching, Germany

The system is designed to cover cooling and heating load of 400 m² of office area as well as to provide domestic hot water. Solar system consists of 57.4 m² of flat plate collectors connected to water/LiBr absorption chiller of 10 kW nominal capacity. Back-up heater is a wood pellet boiler used in the heating season, while a water well as back-up in cooling season. Latent heat storage PCM is used to store heat from the heat rejection unit allowing for lower rejection temperatures (higher COP_h) during day time. A heat discharge via dry cooler is done during night time when the electricity cost is lower.

Table 7.2 Example 1 summary

In operation since	2007
Air-conditioned area	400 m^2
System used for space heating?	Yes
System used for DHW preparation?	Yes
Central air-conditioning unit	
Nominal capacity	10 kW_{cold} (base load)
Type of closed system	Absorption (Water-LiBr)
Chilled water application	Ceiling panel
Dehumidification	No
Heat rejection system	Dry cooler supported by a latent heat storage
Solar thermal	
Collector type	Flat plate
Collector area	57.4 m^2
Tilt angle, orientation	40°, south + 10° west
Collector fluid	water-glycol
Typical operation temperature	92°C driving temperature for chiller operation
Configuration	
Heat storage	2x1 m^3 water tank (in series) and 1.6 m^3 latent heat store
Cold storage	None
Auxiliary heater	Pellet boiler
Auxiliary chiller	Water well

The project is co-funded by the European Union Instrument for Pre-Accession Assistance

Figure 7.4 Absorption chiller and phase change materials (PCM) latent storage tank for solar cooling and heating, ZAE Bayern, Germany [13] and [26].

Table 7.3 Recorded data during monitoring of absorption chiller operation in office building

2009:	June	July	August	Average
Total electrical COP_{el}*	5.4	7.0	7.2	6.6
Primary energy savings	49.4%	60.0%	61.2%	56.9%

*only for the solar cooling part, not water well backup

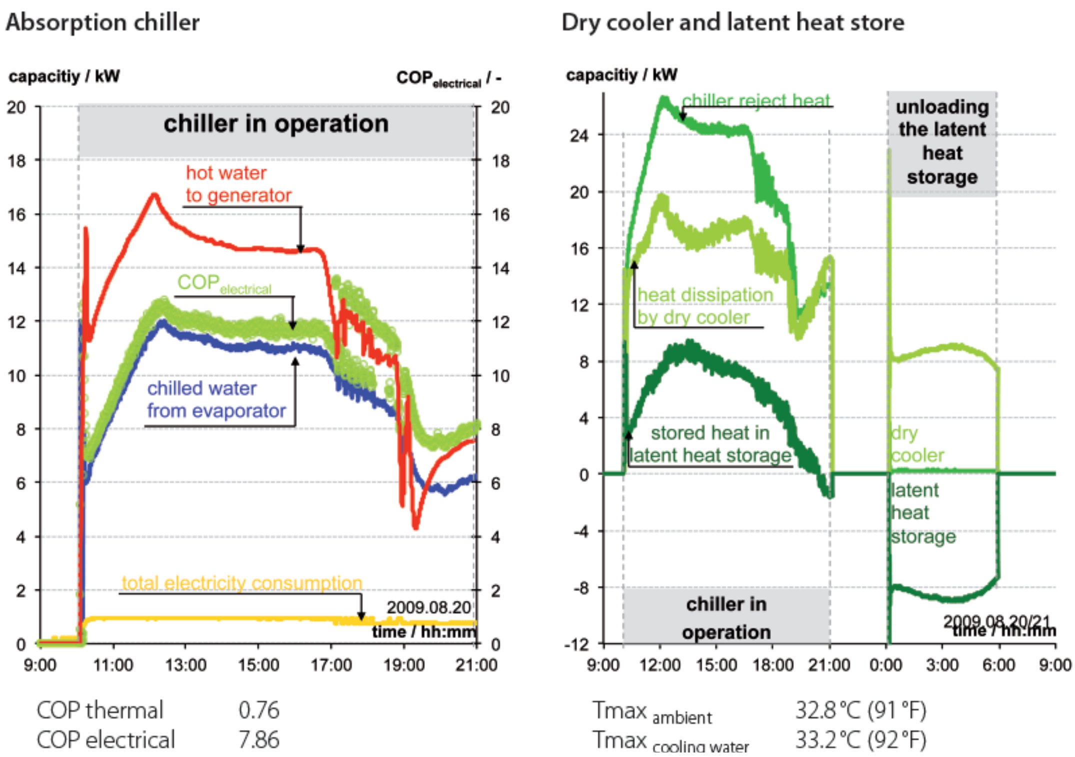


Figure 7.5 Daily monitored system performances (summer) [13].

Example 2:
7.5 kW– adsorption chiller in laboratory building in southern France (Perpignan)

The system consist of 24 m² double and glazed flat plate collectors and adsorption chiller of 7.5 kW nominal cooling capacity. The system produces cold energy independently; in parallel to a general multi split compression system (i.e. solar cooling system covers the base load and the backup covers rest of the load). The distribution system for the solar cooling system works at 14/18°C supply/return temperature level, in cooling mode. It is connected to 3 fan-coils


Figure 7.6 Double-glazed flat plate collectors field and adsorption chiller, laboratory/office building in Perpignan, France [13] and [27].

Table 7.4 Example 2 summary

In operation since	2008
Air-conditioned area	180 m^2 (on 2 levels)
System used for space heating?	Yes
System used for DHW preparation?	No
Central air-conditioning unit	
Nominal capacity	7.5 kW_{cold}
Type of closed system	Adsorption (Silica gel-water)
Chilled water application	Fan coils
Dehumidification	No
Heat rejection system	Dry cooling tower with optional spring water spraying
Solar thermal	
Collector type	Double-glazed flat plate collectors
Collector area	25 m^2 absorber area
Tilt angle, orientation	30° tilt, 45° west
Collector fluid	Water (drain-back system)
Typical operation temperature	75°C driving temperature for chiller operation
Configuration	
Heat storage	0.3 m^3 water
Cold storage	0.3 m^3 water
Auxiliary heater	el. heat pump (separated system, multisplit)
Auxiliary chiller	el. heat pump (separated system, multisplit)

Table 7.5 Recorded data during monitoring of adsorption chiller in laboratory building

2009	May	June	July	August	September	Average
Total electrical COP*	5.5	4.7	4.6	4.0	4.4	4.6
Primary energy savings	49.3%	40.3%	38.8%	30.2%	35.8%	38.9%

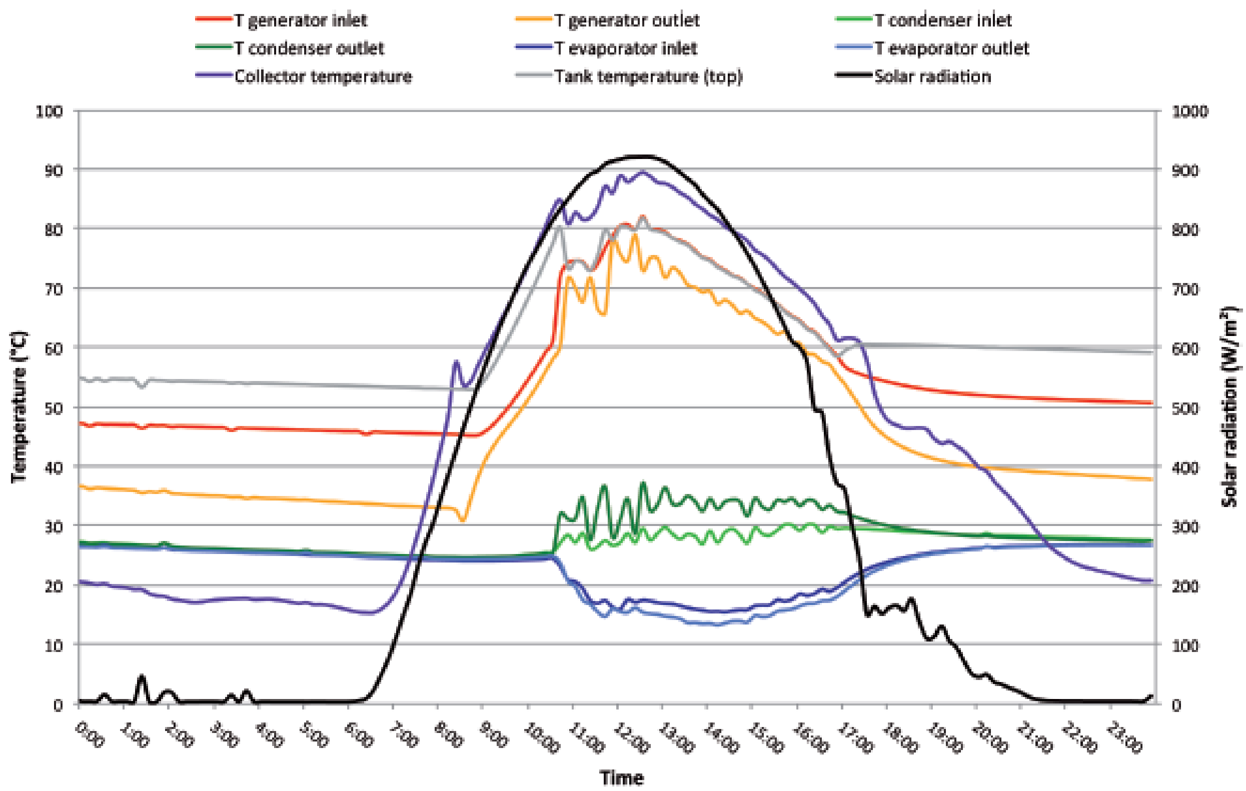


Figure 7.7 Daily monitored temperatures and solar radiation (July) [13].

The total electrical COP_{el} averaged over the summer months is shown in Fig. 7.8 for various monitored systems [13]. As shown, the COP_{el} differs significantly from one system to another, mostly in a good range 4 - 7. Some systems have low COP_{el}<1.5-1.8, which means that they consume more electricity than would a conventional vapour compression chiller. The reason for such low COP_{el} can be too high electricity consumption of auxiliary equipment, which remains constant during part load operation of the chiller. Fig. 7.9 shows example of electricity consumption for small scale heating and cooling system. This all outline an importance of adapting other system components to the reduced load and their careful selection.

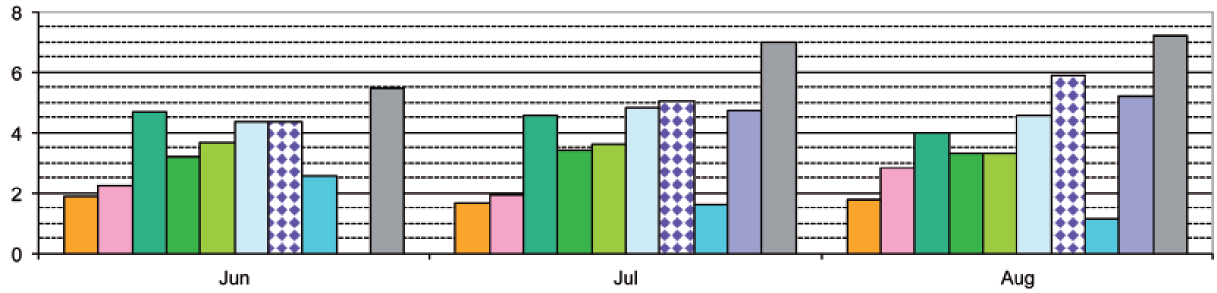


Figure 7.8 Total electrical COPel of monitored small-scale solar cooling systems during the summer months [13], [28] and [29].

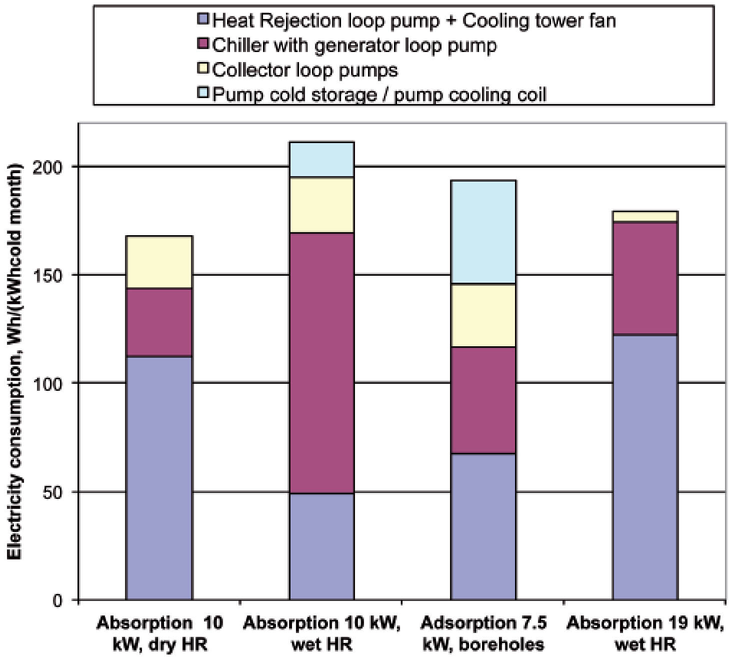


Figure 7.9 Measured electricity consumption of monitored small-scale solar cooling systems during the summer months [13].

7.2 Custom-made systems

Custom-made cooling systems consist of components manufactured by different companies (e.g. solar collectors, thermally driven chiller, control, hydraulic components). Unlike small scale pre-engineered units, they require careful planning and design procedure. Custom-made systems can be distinguished as: installations with closed cycle machines and open (desiccant) cooling systems.

Example 1:
Autonomous absorption system, 30 kW, education centre in La Reunion island – France

In this system solar energy is the only heat source driving the absorption chiller with no heat or cold back-up system employed.

The solar collectors are connected to a single-stage water/LiBr absorption chiller. Heat and cold water storage tanks are installed providing 45 minutes autonomy. Fan coils are installed in classrooms to cool and heat the indoor air.

Figure 7.10 Solar collectors installation on the flat roof of the University of Reunion Island main building [13], [30] and [31].

Table 7.6 Example 1 summary

Thermally driven cooling systems	Technology	Absorption chiller
	Nominal capacity	30 kW_{cold}
	Heat rejection system	70 kW open wet cooling tower
Solar thermal collectors field	Technology	Flat plate collector
	Gross area	90 m^2 gross area
	Tilt angle, orientation	0°C North
	Typical operation temperature	80°C
Storage system	Cold storage	1 m^3
	Heat storage	1.5 m^3

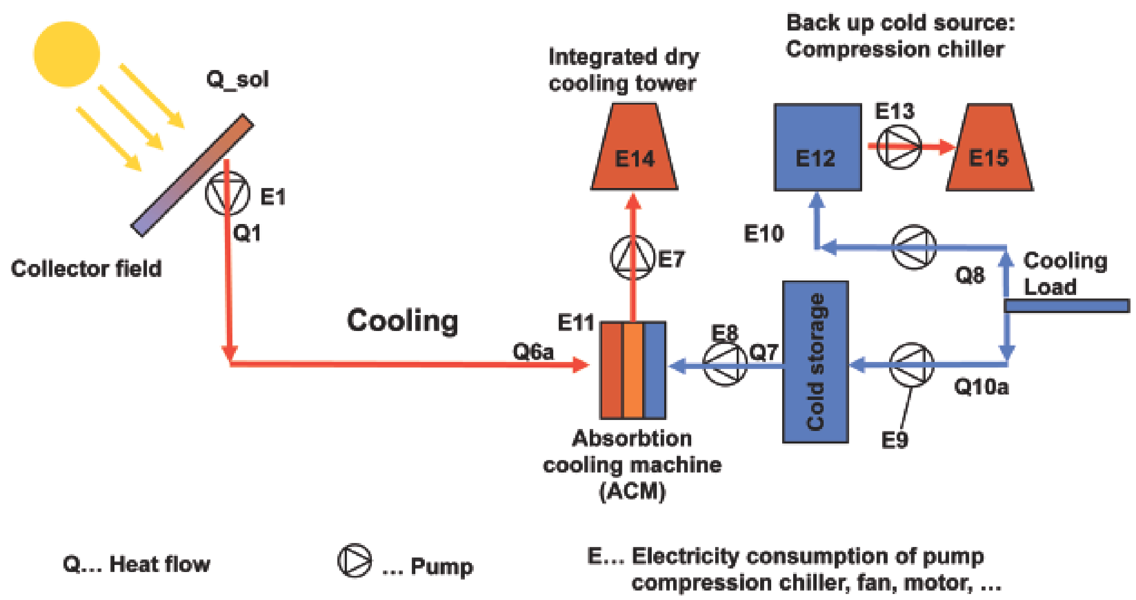


Figure 7.11 Schematic of the solar cooling system installed at the University of Reunion Island (Ile de La Reunion, France) [13], [28] and [29].

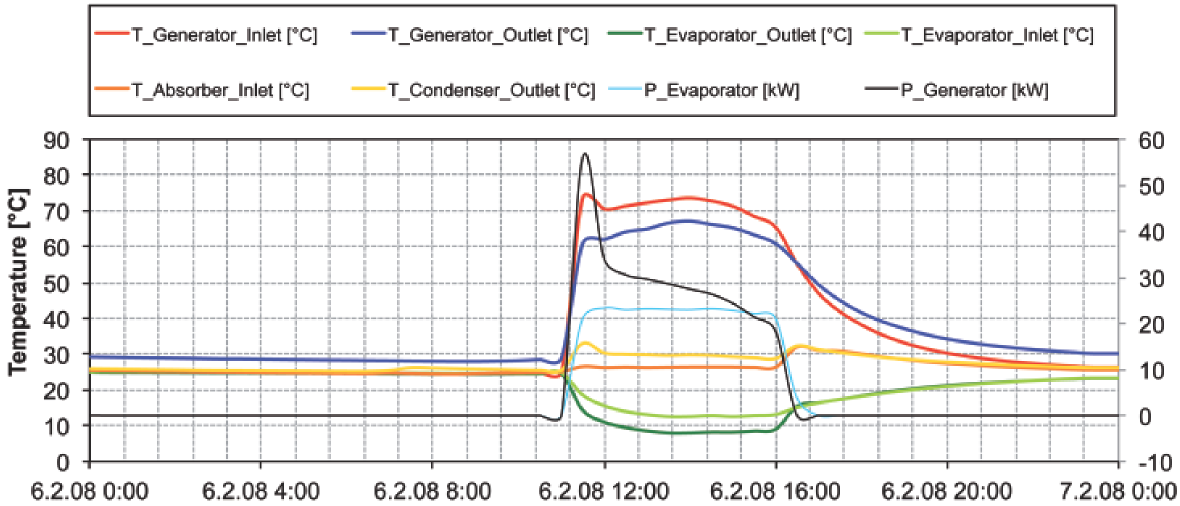


Figure 7.12 Recorded temperatures and the generator / evaporator power for solar cooling system of education centre [13].

Example 2:
Desiccant cooling system, 24 kW, Department of Energy and Environmental Research – Università degli Studi di Palermo (Unipa), Italy

This system is intended for air conditioning in hot humid climates. It consists of the solar system, a desiccant evaporative cooling (DEC) system air handling unit, a vapour compression chiller, all connected to a radiant ceiling system. Back-up heater is a gas boiler.

Figure 7.13 Photo of DEC air handling unit [13] and [32].

Table 7.7 Example 2 summary

Location: Palermo, Italy	
Latitude	38°11' N
Longitude	13°36' E
Elevation	14 m
Application	Office
Solar collector field	22.5 m^2
Heat storage capacity	0.6 m^3
Conditioned volume	450 m^3
Radiant ceiling surface	78 m^2
DEC system volume flow rate	1500 m^3/h
Chiller cooling power	24 kW
Chiller rated EER	3.47

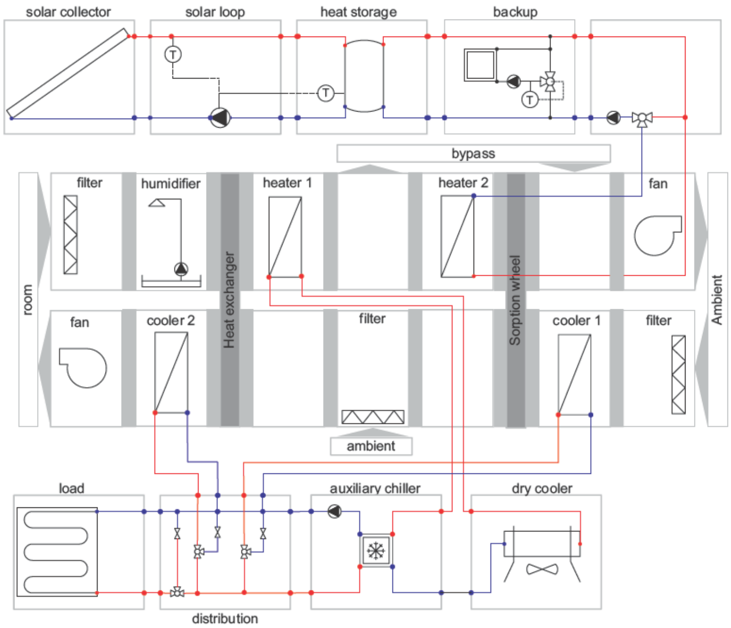


Figure 7.14 Scheme of the DEC system in cooling mode; in heating mode, there is a connection between solar loop and cooler 2 used to preheat supply air [13].

Table 7.8 Recorded data during monitoring of DEC system at Unipa, Palermo [13].

Month		**May**	**Jun**	**Jul**	**Aug**	**Sep**
A) Total cooling energy to supply air	kWh	599	2000	3374	2846	1532
B) Cooling energy to supply air from chiller	kWh	146	807	1941	1138	1007
C) Regeneration heat from solar	kWh	525	1232	1316	1390	829
D) Regeneration heat from chiller condenser	kWh	214	647	722	451	348
E) Solar collectors output	kWh	808	1765	1876	1962	1396
F) Available solar energy input	kWh	2087	4556	4741	4698	3481
G) Electricity DEC	kWh	239	514	589	565	446
H) Electricity solar	kWh	4	9	9	10	7
I) Electricity chiller	kWh	50	330	691	508	382
Thermal COP, (A-B)/(C+D)	-	0.61	0.63	0.70	0.93	0.45
Solar energy utilization factor, E/F	%	39	39	40	42	40
Chiller average EER, B/I	-	2.95	2.44	2.81	2.24	2.64

7.3 Future developments

Absorption and adsorption cooling are main technologies applied in small scale solar cooling systems (< 30kW). Mostly they are installed as central air conditioning systems connected to fan coils or cooled ceilings.

The new small scale and medium scale sorption chillers, have been tested as prototypes and turned into small serial production.

List of small scale absorption chillers with 7.5 kW to 15 kW cooling capacities is provided in Table 7.9. The adsorption chiller *chillii STCR* is intended for use in residential buildings, the ammonia/water absorption chiller *chilli PSC12* for office buildings or process cooling e.g. milk cooling. Water/silica gel absorber *chilli STC 15* and the water/lithium bromide absorber *chilli WFC 18* (17.5 kW cooling capacity) are intended for air conditioning, e.g. office buildings, hotels, banks, bakeries, public and administration buildings.

Table 7.9 Small scale absorption chillers [22].

Company	**SorTech, a)**	**SolarNext, b)**	**SorTech, c)**	**Yazaki, d)**
Product name	chillii® STC8 (ACS 08)	chillii® PSC12	chillii® STC15 (ACS 15)	chillii® WFC 18 (WFC-SC5)
Technology	adsorption	absorption	adsorption	absorption
Working pair	water/silica gel	ammonia/water	water/silica gel	water/lithium bromide
Cooling capacity [kW]	7.5	12	15	17.5
Heating temperature [°C]	75/68	85/78	75/69	88/83
Cold water temperature [°C]	18/15	12/6	18/15	12.5/7
COP	0.56	0.62	0.56	0.70
Dimensions (LxDxH) [m]	0.79 x 1.06 x 0.94	0.80 x 0.60 x 2.20	0.79 x 1.35 x 1.45	0.60 x 0.80 x 1.77
Weight [kg]	260	350	510	420
Electrical power	20	300	30	72

a) b) c) d)

Figure 7.15 Absorption chillers a) SorTech, b) SolarNext, c) SorTech, d) Yazaki

Recently, research and development is directed towards thermally driven water chillers in the medium and small cooling capacities range (5 - 50 kW). Technologies employed are based on liquid sorption materials, water/LiBr or ammonia/water, as well as solids such as silica gel/water, zeolite water or solid absorption using salt/water systems. Table 7.10 provides an overview of main developments.

Table 7.10 Main developments in solar cooling – an overview [21].

Cycle type	Working fluid	Sorption material	Developer/s	Driving temperature	Key features characteristics, description
Closed cycles	Water	Lithium-bromide	Company Rotartica, Research center Ikerlan (both Spain)	70-95°C	Rotating absorber, very low temperatures on HXs
			Company EAW, Research center ILK Dresden (both Germany)	80-90°C	Market available system (cooling capacity >15 kW)
			Company Phönix Sonnenwärme, Research center ZAE Bayern, Technical University Berlin (all Germany)	70-95°C	Good part load behaviour, compact design, prototypes in operation
			Polytechnic Univ. Catalunya (Spain)	75-95°C	Directly air cooled, still in research status
		Silica gel	Company SorTech, Research center Fraunhofer Institute ISE (both Germany)	65-95°C	Compact design, no mechanical moving parts, prototypes in operation
		Lithium-chloride	Company climatewell, Solar Energy Research Center (both Sweden)	70-100°C	High efficient storage included
		Sodium-sulphide	Company Sweat, Research center ECN (both Netherlands)	80-90°C	High efficient (long term) storage, modular system, modular operation
	Ammonia	Water	Company AoSol, Research center INETI (both Portugal)	100-120°C	Standard components, dry air cooling
			Research institute Joanneum Research (Austria)	80-110°C	Prototype in operation, adjustable to different application, low temperatures possible
			University of Applied Science Stuttgart (Germany)	70-120°C	No solution pump, still in research status
Open cycle	Water	Lithium-chloride	Company Menergy (Germany)	60-90°C	Liquid sorption integrated in indirect evaporative cooling systems, pilot plant in operation
			Technion Haifa (Israel)	60-90°C	Liquid desiccant systems, pilot plant in operation
			Research center ZAE Bayern (Germany)	60-90°C	liquid desiccant systems with high efficient energy storage, pilot plant in construction
		Silica gel	Research center Fraunhofer Institute ISE (Germany)	60-100°C	High efficient indirectly cooled system for air cooling and dehumidification

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