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Review

Solar refrigeration options – a state-of-the-art review

D.S. Kim[a,*], C.A. Infante Ferreira[b,1]

[a]Arsenal Research, Sustainable Energy Systems, Giefinggasse 2, 1210 Vienna, Austria
[b]Delft University of Technology, Engineering Thermodynamics, Leeghwaterstraat 44, 2628 CA Delft, The Netherlands

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ABSTRACT

A state-of-the-art review is presented of the different technologies that are available to deliver refrigeration from solar energy. The review covers solar electric, solar thermal and some new emerging technologies. The solar thermal systems include thermo-mechanical, absorption, adsorption and desiccant solutions. A comparison is made between the different solutions both from the point of view of energy efficiency and economic feasibility. Solar electric and thermo-mechanical systems appear to be more expensive than thermal sorption systems. Absorption and adsorption are comparable in terms of performance but adsorption chillers are more expensive and bulkier than absorption chillers. The total cost of a single-effect LiBr–water absorption system is estimated to be the lowest.

Options en froid solaire : l'état de l'art passé en revue

Mots clés : Réfrigération ; Énergie solaire ; Enquête ; Technologie ; Capteur solaire ; Système à compression ; Système à sorption

1. Introduction – solar refrigeration in a warming globe

Since the beginning of the last century, average global temperature has risen by about 0.6 K according to UN Intergovernmental Panel on Climate Change (IPCC). It is also warned that the temperature may further increase by 1.4–4.5 K until 2100 (Climate Change, 2001). Having realized the seriousness of the situation, the world community decided to take initiatives to stop the process. One of such efforts is the Kyoto Protocol, a legally binding agreement under which industrialized countries will reduce their collective emissions of greenhouse gases by 5.2% compared to the year 1990. Especially regarding the reduction of carbon dioxide, being an inevitable byproduct

* *Corresponding author*. Tel.: +43 505506668; fax: +43 505506613.
E-mail addresses: dong-seon.kim@arsenal.ac.at (D.S. Kim), c.a.infanteferreira@tudelft.nl (C.A. Infante Ferreira).
[1] Member of IIR Commission B1. Tel.: +31 152784894.

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Nomenclature

A	area (m^2)
COP	coefficient of performance
I_p	solar radiation perpendicular to collector surface (kW/m^2)
PV	photovoltaic panel
Q	heat transfer rate (kW)
T_H	temperature of high-temperature heat source (K)
T_L	temperature of heat sink (K)
T_M	temperature of low-temperature heat source (K)
W	work (kW)

Greek symbols

η	efficiency

Superscript

id	ideal

Subscripts

a	absorber
c	condenser
cool	cooling
e	evaporator
el	electrical
g	generator
heat	driving heat
pow	mechanical power
s, sol	solar radiation, solar collector

of industrial activities, industries should improve facilities and processes to achieve the goals.

Refrigeration industry is one of those hardest hit by the effect of the protocol. In Europe, use of HFC-134a will be banned for the air conditioning units in new cars starting from 1 Jan 2009. Inspection and/or monitoring are required for all stationary HFC-based refrigeration, air conditioning and heat pump units for the safe containment of HFCs.

Reduction of energy consumption for refrigeration, however, cannot be relied solely on the improvement of efficiency. Reduction in the use of synthetic refrigerants and production of CO_2 provide a new opportunity for solar refrigeration. Considering that cooling demand increases with the intensity of solar radiation, solar refrigeration has been considered as a logical solution. In the 1970s solar refrigeration received great interests when the world suffered from the oil crisis that had been initiated by Arab members of OPEC. There were many projects for development or demonstration of solar refrigeration technologies and solar refrigeration continued to be an important issue in the 1980s (Lamp and Ziegler, 1998). A variety of solar refrigeration technologies have been developed and many of them are available in the market at much cheaper prices than ever.

The first aim of this paper is to give an overview of the state-of-the-art of the different technologies that are available to deliver refrigeration from solar energy. Unlike most review articles that were limited to solar thermal, especially sorption cooling technologies (Lamp and Ziegler, 1998; Li and Sumathy, 2000; Grossman, 2002), this paper is intended to give a broader overview including solar electric, thermo-mechanical, sorption and also some newly emerging technologies. The second aim is to compare the potential of these different technologies in delivering competitive sustainable solutions. The current commercial status of different solar refrigeration technologies may be quickly viewed in a comparison of the initial costs of various cooling systems.

2. Solar electric refrigeration

A solar electric refrigeration system consists mainly of photovoltaic panels and an electrical refrigeration device. Solar cells are basically semiconductors whose efficiency and cost vary widely depending on the material and the manufacturing methods they are made from. Most of the solar cells commercially available in the market are made from silicon as the ones shown in Fig. 1.

In Eq. (1), efficiency of a solar panel is defined by the ratio of power W (kW) to the product of solar panel surface area A_s (m^2) and the direct irradiation of solar beams I_p (kW/m^2). $I_p = 1\ kW/m^2$ is commonly used for the calculation of nominal efficiency.

$$\eta_{\text{sol-pow}} = \frac{W}{I_p \times A_s} = \frac{W}{Q_s} \tag{1}$$

Although higher efficiencies are reported from laboratories, a high-performance solar panel sold in the market yields about 15% efficiency under the midday sun in a clear day. A study on building-integrated solar panels reported an overall efficiency of 10.3% (Fanney et al., 2001). Price of a solar panel varies widely in the market. For example, retail price of a solar panel in Germany varies between €3 and €7 (Solar Rechner) per W_p (peak Watt), i.e. production of 1 W under 1 kW/m^2 of solar radiation.

The biggest advantage of using solar panels for refrigeration is the simple construction and high overall efficiency when combined with a conventional vapour compression system. A schematic diagram of such a system is given in Fig. 2. In Fig. 2, the work W is consumed by the mechanical compressor to produce the cooling power Q_e. Refrigeration machine efficiency is defined as the cooling power Q_e divided by the work input W:

$$\eta_{\text{pow-cool}} = \frac{Q_e}{W} \tag{2}$$

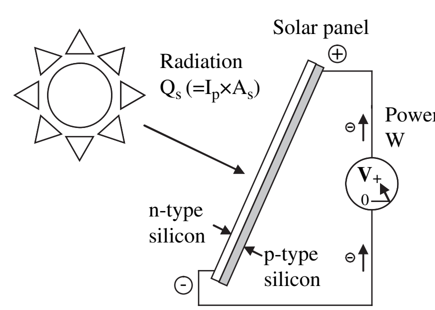

Fig. 1 – Schematic diagram of a solar photovoltaic panel.

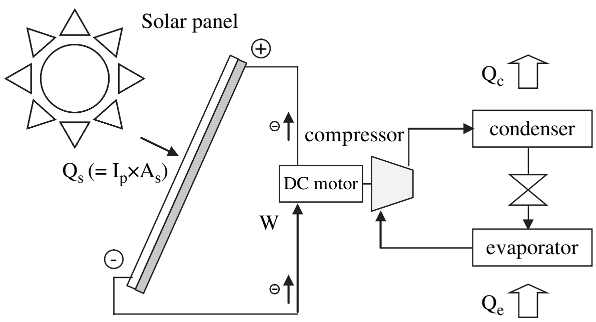

Fig. 2 – Schematic diagram of a solar electric compression air-conditioner.

Combination of the two efficiencies in Eq. (1) and Eq. (2) gives the solar-to-cooling or the overall efficiency of a solar electric cooling system:

$$\eta_{\text{sol-cool}} = \eta_{\text{sol-pow}} \times \eta_{\text{pow-cool}} = \frac{Q_e}{Q_s} \quad (3)$$

COP (Coefficient of Performance) is an alternative term to efficiency commonly used in thermodynamics.

Solar electric vapour compression refrigeration systems are limited and only a few systems are found in literature. Several solar electric refrigeration systems were designed for autonomous operation and packaged in standard containers (Rudischer et al., 2005). Cooling COPs of the vapour compression machines in those systems ranged from 1.1 to 3.3 for different evaporator temperatures between −5 and 15 °C and condenser temperatures between 45 and 61 °C. Monocrystalline PV modules and variable-speed compressors were used with batteries or generators as a backup.

There are several challenges in the broader commercialization of this type of systems. Firstly, the systems should be equipped with some means to cope with the varying electricity production rate with time, e.g. electric battery, mixed use of solar- and grid-electricity or a variable-capacity compressor. Secondly, the price of a solar photovoltaic panel should be further decreased to compete with other solar cooling technologies. If a 10% efficiency solar photovoltaic panel is combined with a vapour compression air conditioner with 3.0 COP, the overall efficiency will be 30%. Assuming the unit price of the solar panel is €5/W_p, the solar panel alone would cost ca. €1700 to produce 333 W electricity for 1 kW cooling.

Besides vapour compression cooling, some other types of electric refrigeration technologies can be used in combination with solar panels. Thermoelectric elements are made of semiconducting materials such as bismuth telluride and antimony telluride alloys (Bi_2Te_3 and Sb_2Te_3). Since they have neither moving parts nor refrigerant and can be made very small, they have been used in electronic chip cooling, portable refrigerators and in space applications like satellite and space ships where physical size of a cooling system is extremely limited. COP of this system is currently very low, ranging from 0.3 to 0.6. Small thermoelectric air-conditioners with a few hundred Watt capacity are available in the market.

A Stirling refrigerator can be connected to solar panels to provide cooling. Although an ideal Stirling cycle should work as efficiently as Carnot cycle, COPs of the Stirling refrigerators are lower than those of vapour compression counterparts. Ewert et al. (1998) reported the test results of a small (maximum cooling capacity 100 W) free-piston Stirling refrigerator. The COP decreased from 1.6 to 0.8 while temperature lift (air-to-air) varied from 13 to 33 K with ambient temperature from 23 to 28 °C. Berchovitz et al. (1999) reported the COP of a similar machine (nominal capacity 40 W), which decreased from 1.65 to 1.17 with decreasing cold-side temperature from −1.4 to −19.1 while hot-side temperature was maintained between 28.4 and 30.3 °C. There are many practical difficulties in developing an efficient Stirling refrigerator or air-conditioner. Major problems are low COP and limited power density due to the poor heat transfer between working fluids (mostly helium) and the ambient (Kribus, 2002). For this reason, only a small Stirling refrigerator, where surface-to-volume ratio is relatively large, is competitive against small domestic vapour compression refrigerators.

Electrically driven thermo-acoustic refrigeration machines are another option for solar refrigeration. These machines use pressure changes in acoustic waves to transfer heat between two reservoirs at different temperature levels. The working principle is discussed in American Institute of Physics (2004). Efficiencies of thermo-acoustic cooling systems are lower than those of vapour compression systems. Poese et al. (2004) reported the performance of a refrigeration system with a cooling capacity of 119 W designed for 200-l ice cream cabinet. The system yielded COP of 0.81 with heat transfer fluid temperatures in heat exchangers of 33.9 °C and −24.6 °C. These performance figures are comparable to those of the small Stirling refrigerators described above. Fischer and Labinov (2000) mentioned the development of a 10 kW air conditioning system expecting COP of 2.0 with ambient temperature at 35 °C. Although a thermo-acoustic system has a very simple construction with no moving part, cooling power density is low and no machine has been reported with a reasonably large capacity for air conditioning.

Magnetic cooling, which has long been used in cryogenics, is also a possibility. Recently, a few permanent-magnet room-temperature magnetic refrigeration systems have been developed (Gschneider, 2001; Shir et al., 2005). Gschneider (2001) demonstrated an overall COP of 3.0 with a rotary magnetic refrigerator/freezer. Although this technology has a potential of outperforming conventional vapour compression technology, the cost of magnetic material is prohibitively expensive ($1830/kW cooling, gadolinium without processing cost – Fischer and Labinov, 2000) for practical application.

3. Solar thermal refrigeration

Solar thermal systems use solar heat rather than solar electricity to produce refrigeration effect.

Flat-plate solar collectors are the most common type, which consists of a metallic absorber and an insulated casing topped with glass plate(s). Evacuated collectors have less heat loss and perform better at high temperatures. Evacuated collectors are typically made in a glass tube design, i.e. a metallic absorber inserted in an evacuated glass tube, to withstand the pressure difference between the vacuum and the atmosphere. Fig. 3 shows schematic diagrams of these two collectors.

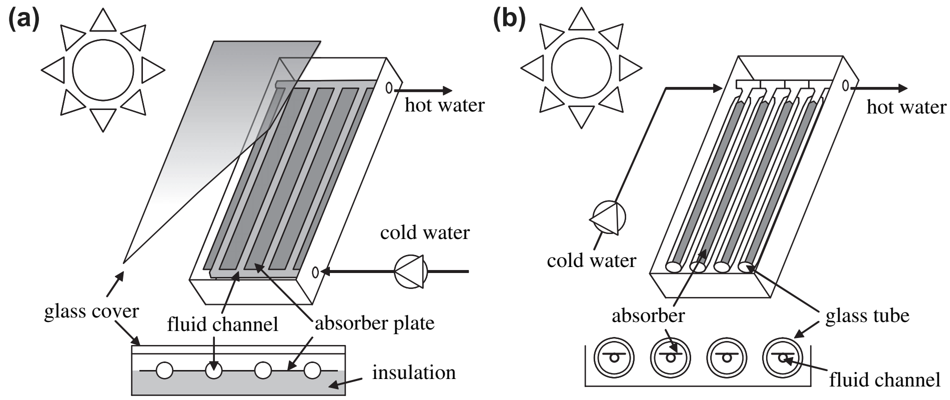

Fig. 3 – Schematic diagrams of non-concentrating solar collectors. (a) Flat-plate type and (b) evacuated tube type.

A solar collector provides heat to the "heat engine" or "thermal compressor" in a heat-driven refrigeration machine. The efficiency of a solar collector is primarily determined by its working temperature. At a higher working temperature, the collector looses more heat to ambient and delivers less heat. On the other hand, the heat engine or thermal compressor generally works more efficiently with a higher temperature. A solar thermal system is designed in consideration of these two opposing trends.

3.1. *Thermo-mechanical refrigeration*

In a solar thermo-mechanical refrigeration system, a heat engine converts solar heat to mechanical work, which in turn drives a mechanical compressor of a vapour compression refrigeration machine. A schematic diagram of such a cooling system is shown in Fig. 4. In the figure, a solar collector receives solar radiation Q_s [the surface area A_s (m^2) multiplied by the solar radiation perpendicular to the surface I_p (kW/m^2), see Eq. (4)] from the sun and supplies Q_g to a heat engine at the temperature T_H. The ratio of supply heat Q_g to the radiation Q_s is defined as the thermal efficiency of a solar thermal collector, $\eta_{sol\text{-}heat}$.

$$\eta_{sol\text{-}heat} = \frac{Q_g}{I_p \times A_s} = \frac{Q_g}{Q_s} \tag{4}$$

$\eta_{sol\text{-}heat}$ is less than 1 due to optical and thermal losses.

A heat engine produces mechanical work W and rejects heat Q_a to ambient at temperature T_M. The efficiency of engine, $\eta_{heat\text{-}pow}$ is defined as the work produced per heat input Q_g in Eq. (5).

$$\eta_{heat\text{-}pow} = \frac{W}{Q_g} \tag{5}$$

The mechanical work W in turn drives the compressor of the refrigeration machine to remove heat Q_e from the cooling load at temperature T_L. Waste heat Q_c, which is equal to the sum of Q_e and W, is rejected to ambient at the temperature T_M. Efficiency of the refrigeration machine is the same as in Eq. (2).

Then the overall efficiency of a solar thermo-mechanical refrigeration system is given by the three efficiencies in Eqs. (4), (5) and (2) as follows:

$$\eta_{sol\text{-}cool} = \eta_{sol\text{-}heat} \times \eta_{heat\text{-}pow} \times \eta_{pow\text{-}cool} = \frac{Q_e}{Q_s} \tag{6}$$

The maximum efficiencies of the real engine and refrigeration machine are limited by those of Carnot cycles working at the

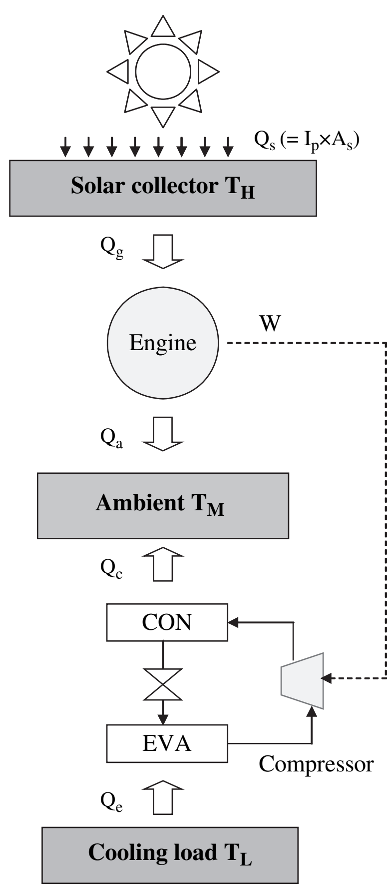

Fig. 4 – Solar thermo-mechanical refrigeration system.

same temperatures. The efficiency of a Carnot power cycle working between T_H and T_M is given by

$$\eta^{id}_{heat\text{-}pow} = \frac{T_H - T_M}{T_H} \tag{7}$$

and the efficiency of a Carnot refrigeration cycle working between T_M and T_L is given by

$$\eta^{id}_{pow\text{-}cool} = \frac{T_L}{T_M - T_L} \tag{8}$$

The product of the two Carnot efficiencies in Eqs. (7) and (8) gives the efficiency of an ideal heat-driven refrigeration machine working between the three temperatures as

$$\eta^{id}_{heat\text{-}cool} = \eta^{id}_{heat\text{-}pow}\eta^{id}_{pow\text{-}cool} = \frac{T_L}{T_H}\left(\frac{T_H - T_M}{T_M - T_L}\right) \tag{9}$$

which limits the maximum efficiency achievable with any real heat-driven refrigeration machine working between the same temperatures. In a solar thermo-mechanical system, the efficiency of a heat engine is of particular interest. Because the heat source temperature T_H varies in different projects, the performance of a real engine is often compared to that of a Carnot cycle working at the same temperatures. The ratio of real efficiency to Carnot efficiency is called ''second law efficiency''. This is a measure of how closely a real machine operates to an ideal machine.

For solar power generation, Rankine and Stirling power engines have been frequently considered.

Solar Rankine systems were actively investigated in the 1970s and 1980s. Prigmore and Barber (1975) designed a water-cooled organic Rankine cycle based on R-113 to produce turbine shaft work with 11.5% efficiency (58% second law efficiency) from 101.7 °C water from solar collectors. When 50% solar collector efficiency is assumed, the solar-to-power efficiency would have been 5.8%.

With higher heat source temperature, higher engine efficiency can be achieved. In early 1980s, a trough ORC (Organic Rankine Cycle) solar power plant has been reported. The system used a trough type concentrating collector and a toluene Rankine power cycle. A peak heat-to-power efficiency of 24% (57% second law efficiency) was attained with a heat transfer fluid temperature of 268 °C (Larson, 1983). Higher power generation efficiency was reported from a large-scale solar power generation system. The Solar One demonstration plant was equipped with a 35%-efficient (58% second law efficiency) Rankine power generation system driven by 516 °C superheated steam from the tower-mounted receiver on which solar radiation was focused by thousands of sun-tracking mirrors on the ground (Stein and Geyer, 2001).

If a 24%-efficient Rankine cycle working at 268 °C heat is connected to a state-of-the-art trough collector of today, e.g. EuroTrough from Geyer et al. (2002) has an efficiency of 67% at this temperature, the system would yield about the same efficiency as a high-performance solar electric panel (ca. 16%) in the market.

Stirling engines have also been actively studied for power generation from the sun. Stirling engines can operate at a very high temperature at which a Rankine engine cannot. Although Stirling cycle efficiency approaches that of a Carnot engine in theory, the efficiencies of Stirling engines are in the range of 55–88% of second law efficiency (Reader and Hooper, 1983). A heat-to-electricity efficiency of 41% (≈57% of second law efficiency) has been reported (Stein and Diver, 1994) for Stirling engines. Its success in this particular solar application is attributed to its high-temperature operability (gas temperature above 700 °C) and relatively simple design. The maximum capacity of a Stirling engine is practically limited by the fact that its efficiency decreases with increasing capacity, i.e. decreasing surface-to-volume ratio.

In order for a solar thermo-mechanical refrigeration system to be competitive, the combination of a solar collector and a heat engine should be at least comparable to a solar electric panel in terms of price. Assuming that a 60%-Carnot-efficient engine works with 150 °C heat source and 28 °C heat sink, the heat-to-power efficiency of this engine will be 17%. Among non-concentrating type solar collectors, only some evacuated tube type collectors can operate efficiently at 150 °C. A high performance evacuated tube collector working with 60% efficiency at 150 °C is available at the price of €771/m^2 (Sydney SK-6, Henning, 2004). If this collector is combined with the heat engine, its solar-to-power efficiency would be 10%. Per 1 m^2 of the solar collector, 100 W of work will be produced under 1 kW/m^2 solar radiation. Therefore the collector price per produced work is €7.71/W exclusive heat engine costs. This is rather high compared to the price of a solar electric panel in the current market (€3–7/W$_p$, Solar Rechner). A solar thermo-mechanical refrigeration system is likely more expensive than a solar electric refrigeration system.

3.2. *Sorption refrigeration*

Sorption refrigeration uses physical or chemical attraction between a pair of substances to produce refrigeration effect. A sorption system has a unique capability of transforming thermal energy directly into cooling power. Among the pair of substances, the substance with lower boiling temperature is called sorbate and the other is called sorbent. The sorbate plays the role of refrigerant.

Fig. 5 shows a schematic diagram of a closed sorption system. The component where sorption takes place is denoted as absorber and the one where desorption takes place is denoted as generator. The generator receives heat Q_g from the solar collector to regenerate the sorbent that has absorbed the refrigerant in the absorber. The refrigerant vapour generated in this process condenses in the condenser rejecting the condensation heat Q_c to ambient. The regenerated sorbent from the generator is sent back to the absorber, where the sorbent absorbs the refrigerant vapour from the evaporator rejecting the sorption heat Q_a to ambient. In the evaporator, the liquefied refrigerant from the condenser evaporates removing the heat Q_e from the cooling load.

In an adsorption system, each of the adsorbent beds alternates generator and absorber function due to the difficulty of transporting solid sorbent from one to another.

In sorption refrigeration machines, a single heat-to-cooling efficiency is often defined by

$$\eta_{heat\text{-}cool} = \frac{Q_e}{Q_g + W_{el}} \tag{10}$$

where W_{el} in the denominator denotes electrical work. This efficiency, also called COP, is often compared with the ideal

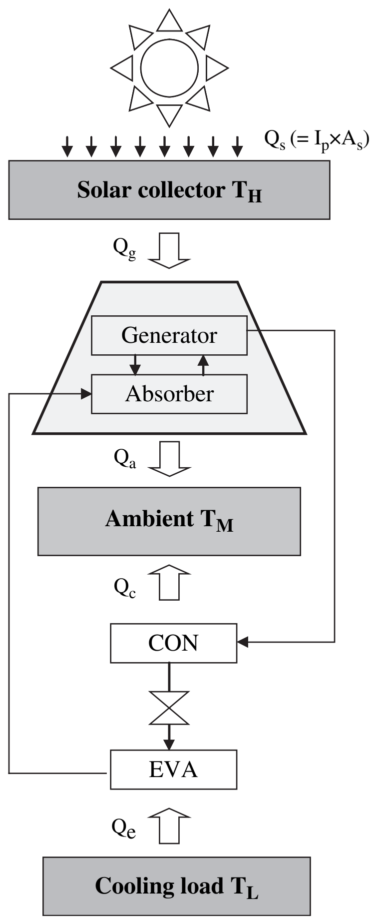

Fig. 5 – Solar sorption refrigeration system.

efficiency in Eq. (9) to measure how the system efficiency deviates from ideal efficiency.

Absorption refers to a sorption process where a liquid or solid sorbent absorbs refrigerant molecules into its inside and changes physically and/or chemically in the process. Adsorption, on the other hand, involves a solid sorbent that attracts refrigerant molecules onto its surface by physical or chemical force and does not change its form in the process. Desiccation refers to a sorption process where a sorbent, i.e. a desiccant, absorbs moisture from humid air. This process is employed in open sorption cycles, which are classified into either liquid or solid desiccant cycles depending on the phase of the desiccant used.

3.2.1. *Absorption*

Absorption refrigeration has been most frequently adopted for solar refrigeration. It requires very low or no electric input and, for the same capacity, the physical dimensions of an absorption machine are smaller than those for adsorption machines due to the high heat transfer coefficient of the absorbent. Besides, the fluidity of the absorbent gives greater flexibility in realizing a more compact and/or efficient machine. Table 1 summarizes the details of a number of studies related to solar absorption refrigeration.

Other than listed in Table 1, numerous studies have been reported including various absorption cycles (Chinnappa and Martin, 1976; Sofrata et al., 1981; Ziegler et al., 1993; Alizadeh, 2000; Göktun and Er, 2001) and different working pairs (Sawada et al., 1994; Romero et al., 2001; Arivazhagan et al., 2005).

Current absorption technology can provide various absorption machines with COPs ranging from 0.3 to 1.2. Choice of an absorption cooling machine is primarily dependent on the performance of the solar collector to be used. For solar collectors capable of efficiently working at around 150 °C, double-effect LiBr–water chillers with COPs around 1.2 are available for air conditioning. For refrigeration, ammonia–water GAX chillers with COPs around 0.8 can be considered. Heat transfer medium can be either a liquid with a high boiling temperature or steam. A high performance evacuated tube or a concentrating type collector can be considered. According to Collector Catalogue (2004), a 40%-efficient evacuated tube collector at this temperature level costs €600–700/m^2 (gross area). For less expensive collectors working at around 90 °C, a single-effect LiBr–water or an ammonia–water absorption machine with a COP between 0.6 and 0.8 can be considered. Price of a solar collector varies widely in this temperature range. The price of a 50%-efficient collector at 90 °C ranges between €300 and €600/m^2. It must be noted that the solar collector efficiencies listed above are only indicative and actual efficiencies will depend on ambient air temperature and solar radiation.

3.2.2. *Adsorption*

3.2.2.1. Physical adsorption. Adsorbents like zeolite, silica gel, activated carbon and alumina are physical adsorbents having highly porous structures with surface–volume ratios in the order of several hundreds that can selectively catch and hold refrigerants. When saturated, they can be regenerated simply by being heated. If an adsorbent and a refrigerant are contained in the same vessel, the adsorbent would maintain the pressure by adsorbing the evaporating refrigerant. The process is intermittent because the adsorbent must be regenerated when it is saturated. For this reason, multiple adsorbent beds are required for continuous operation.

Employed working pairs include activated carbon and methanol or ammonia (Pons and Guilleminot, 1986; Wang et al., 1997, 2000; Critoph, 2002) and silica gel–water (Grenier et al., 1988; Hildbrand et al., 2004). Current solar adsorption technology can provide a daily ice production of 4–7 kg per unit square meters of solar collector with a solar-to-cooling COP between 0.1 and 0.15 (Wang and Oliveira, 2005). Recently, several small-capacity silica gel–water adsorption chillers have been developed for solar air conditioning (Saha et al., 2001; Nuñez et al., 2004; Liu et al., 2005). Cooling capacities were reported between 3.2 and 3.6 kW. COPs ranged from 0.2 to 0.6 with heating temperatures from 55 to 95 °C. Unlike the more common single-staged double-bed systems, Saha et al. (2001) developed a double-staged four-bed cycle machine to use very low driving temperatures. The machine produced 3.2 kW cooling with COP of 0.36 from 55 °C hot water.

Presently, there are two major manufacturers of adsorption chillers (Saman et al., 2004). Their machines are all

Table 1 – Overview of solar absorption refrigeration studies

References	Application	Q_e [kW]	A_s [m^2]	$\eta_{heat\text{-}cool}$ [–][d]
Single-effect LiBr–water chillers				
Löf and Tybout (1974), Ward and Löf (1975) and Ward et al. (1979)	Space cooling/heating			
Hattem and Data (1981)	Space cooling	4	36[a]	0.11
Al-Karaghouli et al. (1991)	Space cooling	210	1577[b]	0.31
Best and Ortega (1999)	Space cooling	90	316[a]	0.26–0.36
Izquierdo et al. (2005)	Space cooling	35	49.9[a]	0.34
Storkenmaier et al. (2003) and Kühn et al. (2005)	Prototype chiller	10		0.37
Safarik et al. (2005)	Prototype chiller	16		0.40
Double-effect LiBr–water chillers				
Ishibashi (1979) and Lamp and Ziegler (1998)	Fuel-fired solar-assisted prototype			
Lokurlu and Müller (2005)	Cooling/steam (144 °C) generation	140	180[c]	0.5–0.6
Ammonia–water chillers				
Gutiérrez (1988), Kunze (2000) and Jakob et al. (2003)	Diffusion–absorption prototype	<2.5		0.1–0.25
Shiran et al. (1982), McLinden and Klein (1983), Alvares and Trepp (1987), Best (1991) and ARTISC (2003)	Refrigeration/heat pump			
SACE (2003)	Wine cooling	10	100[a]	
Richter and Safarik (2005)	Space cooling	15		0.27

[a] Flat-plate collectors.
[b] Evacuated tube collectors (no. of tubes).
[c] Trough collectors (aperture area).
[d] Where not given, a collector efficiency of 0.50 has been assumed.

based on silica gel–water with cooling capacities between 70 and 350 kW (Wang and Oliveira, 2005). According to the manufacturer's specification (HIJC USA Inc.), one of their models produces 72 kW cooling from 90 °C hot water with COP of 0.66 when 29 °C cooling water is supplied. The operation weight is 5.5 ton and its dimensions are $2.4 \times 3.6 \times 1.8$ m^3. One of the single-effect LiBr–water absorption chiller models available in the market produces 70 kW cooling from 88 °C hot water with COP of 0.7 when cooling water temperature is 31 °C (Yazaki Energy Systems Inc.). Its operation weight is 1.2 ton and its dimensions are $2 \times 1.1 \times 1.3$ m^3. The adsorption chiller is 4.6 times heavier and 5.4 times bulkier than the absorption chiller. The major problem associated with adsorption technology is its low cooling power density.

For a high specific cooling power (SCP), various ideas have been tried including the use of extended surfaces such as plate-fin heat exchangers (Liu et al., 2005; de Boer et al., 2005), adsorbent-coated heat exchangers (Talter and Erdem-Şenatalar, 2000; Wojcik et al., 2001), consolidated composite adsorbents (Tamainot-Telto and Critoph, 1997; Poyelle et al., 1999; Wang et al., 2004).

Adsorption chillers seem to be comparable with absorption chillers in terms of maximum achievable COP. But their cooling power densities are much lower. Adsorption technology may be competitive in large solar cooling systems where its low power density is not a problem. For small- or medium-size solar cooling systems, it tends to be too bulky and expensive (Saman et al., 2004).

3.2.2.2. *Chemical adsorption*. Chemical adsorption is characterized by the strong chemical bond between the adsorbate and the adsorbent. Therefore it is more difficult to reverse and thus requires more energy to remove the adsorbed molecules than in physical adsorption.

The most commonly used chemical adsorbent in solar cooling applications has been calcium chloride ($CaCl_2$). Calcium chloride adsorbs ammonia to produce $CaCl_2 \cdot 8NH_3$ and water to produce $CaCl_2 \cdot 6H_2O$ as a product (Wang et al., 2004). It has also been used together with other physical adsorbents including some silicates (Tokarev et al., 2002; Restuccia et al., 2004). Tokarev et al. (2002) developed a composite material by impregnating calcium chloride in MCM-41 (a silicate) matrix. A COP of 0.7 was achievable with condenser and generation temperatures at 40 °C and 110 °C, respectively. Restuccia et al. (2004) developed a chiller based on a similar composite and reported COP of 0.6 at the condenser temperature of 35 °C and the generation temperature between 85 and 95 °C.

Metal hydride refrigeration uses hydrogen as a refrigerant. The interest in metal hydride refrigeration systems is increasing for their integration into hydrogen-fuelled systems. In a basic two-bed refrigeration system, one bed is filled with a high-temperature hydride and the other is filled with a low-temperature hydride. In recharge mode, the high-temperature bed is heated to release hydrogen while the low-temperature bed is cooled to absorb the hydrogen. When the high-temperature bed is cooled in cooling mode, hydrogen is released from the low-temperature bed creating heating effect by absorbing heat. The research issues on metal hydride refrigeration are basically the same as the other adsorption technologies including the enhancement of specific cooling capacity and heat transfer in the beds. Driving temperature of a single-stage system starts from as low as 80 °C depending on the hydride and the heat rejection temperature. COPs of single-stage systems are in the vicinity of 0.5 (Gopal and Murthy, 1995; Hovland, 2002).

3.2.3. *Desiccant cooling*

Open sorption cooling is more commonly called desiccant cooling because sorbent is used to dehumidify air. Various desiccants are available in liquid or solid phases. Basically all water absorbing sorbents can be used as a desiccant. Examples are silica gel, activated alumina, zeolite, LiCl and LiBr.

In a liquid desiccant cooling system, the liquid desiccant circulates between an absorber and a regenerator in the same way as in an absorption system. Main difference is that the equilibrium temperature of a liquid desiccant is determined not by the total pressure but by the partial pressure of water in the humid air to which the solution is exposed to. A typical liquid desiccant system is shown in Fig. 6. In the dehumidifier of Fig. 6, a concentrated solution is sprayed at point *A* over the cooling coil at point *B* while ambient or return air at point 1 is blown across the stream. The solution absorbs moisture from the air and is simultaneously cooled down by the cooling coil. The results of this process are the cool dry air at point 2 and the diluted solution at point *C*. Eventually an aftercooler cools this air stream further down. In the regenerator, the diluted solution from the dehumidifier is sprayed over the heating coil at point *E* that is connected to solar collectors and the ambient air at point 4 is blown across the solution stream. Some water is taken away from the diluted solution by the air while the solution is being heated by the heating coil. The resulting concentrated solution is collected at point *F* and hot humid air is rejected to the ambient at point 5. A recuperative heat exchanger preheats the cool diluted solution from the dehumidifier using the waste heat of the hot concentrated solution from the regenerator, resulting in a higher COP.

A solid desiccant cooling system is quite different in its construction mainly due to its non-fluid desiccant. Fig. 7 shows an example of a solar-driven solid desiccant cooling system. The system has two slowly revolving wheels and several other components between the two air streams from and to a conditioned space. The return air from the conditioned space first goes through a direct evaporative cooler and enters the heat exchange wheel with a reduced temperature ($A \rightarrow B$). It cools down a segment of the heat exchange wheel which it passes through ($B \rightarrow C$). This resulting warm and humid air stream is further heated to an elevated temperature by the solar heat in the heating coil ($C \rightarrow D$). The resulting hot and humid air regenerates the desiccant wheel and is rejected to ambient ($D \rightarrow E$). On the other side, fresh air from ambient enters the regenerated part of desiccant wheel ($1 \rightarrow 2$). Dry and hot air comes out of the wheel as the result of dehumidification. This air is cooled down by the heat exchange wheel to a certain temperature ($2 \rightarrow 3$). Depending on the temperature level, it is directly supplied to the conditioned space or further cooled in an aftercooler ($3 \rightarrow 4$). If no aftercooler is used, cooling effect is created only by the heat exchange wheel, which was previously cooled by the humid return air at point *B* on the other side. Temperature at point 3, T_3, cannot be lower than T_B, which in turn is a function of the return air condition at point *A*.

From a thermodynamic point of view, the dehumidification process is not much different from a closed sorption process. Neglecting the enthalpy changes in the air flow, the same heat will be required to remove 1 kg of water from a sorbent regardless it is in a closed vessel or it is in a humid air stream. Therefore, in principle, the COP of an open desiccant system is similar to its closed counterpart. For example, COP of 0.7 was said achievable with a solid desiccant cooling system under ''normal'' operating conditions (Henning, 2004). Similar COPs were also reported for liquid dehumidifiers (Matsushita et al., 2005). But in practice, COP varies widely depending on operating conditions.

A desiccant cooling system is actually a complete HVAC system which has ventilation, humidity and temperature control devices in a ductwork. Therefore it is inappropriate to

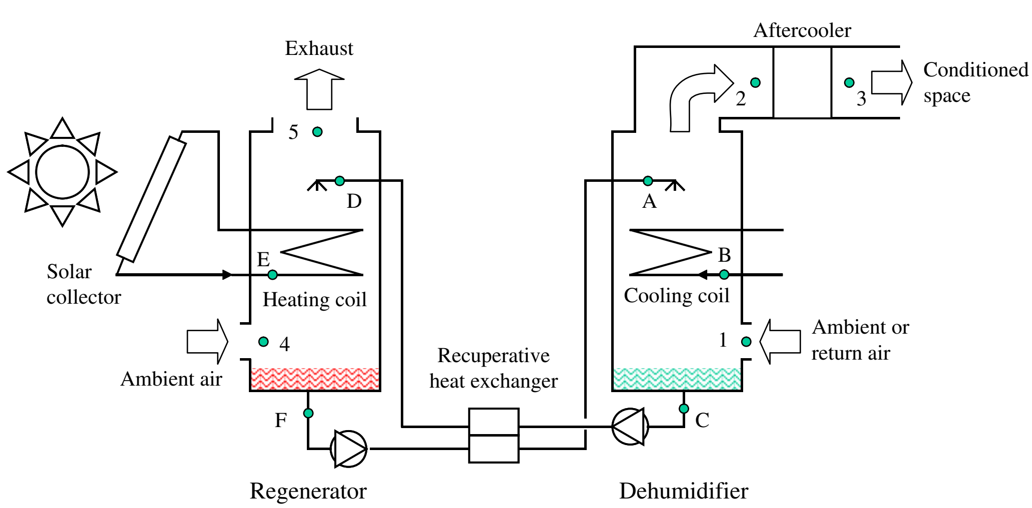

Fig. 6 – A liquid desiccant cooling system with solar collector.

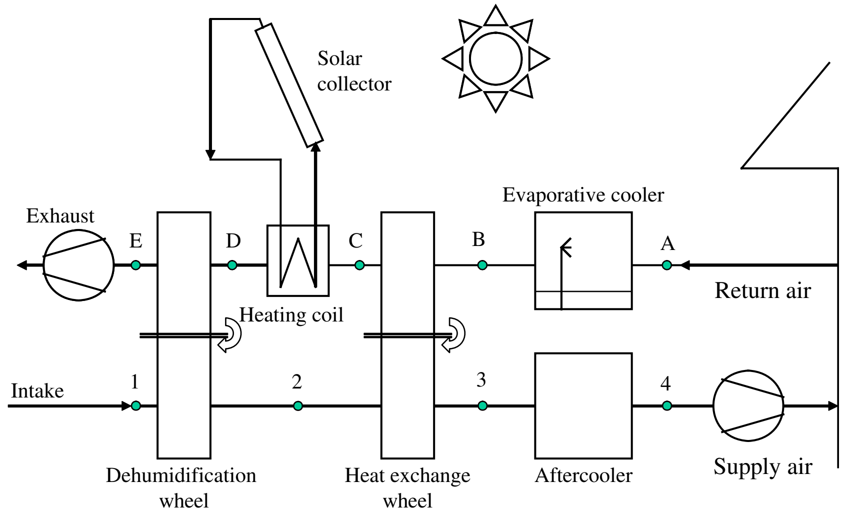

Fig. 7 – A solid desiccant cooling system with solar collector.

compare a desiccant cooling system with such components as chillers. Desiccant dehumidification offers a more efficient humidity control than the other technologies. When there is a large ventilation or dehumidification demand, solar-driven desiccant dehumidification can be a very good option.

4. Other technologies

Electrochemical refrigeration is a new concept, which uses the thermal effects of the reversible electrochemical reactions such as in a reversible electrochemical cell. This new refrigeration concept is based on the idea that a reversible electrochemical cell that releases heat when voltage is applied would absorb heat when the voltage is reversed (Gerlach and Newell, 2003). This technology is very young and currently being investigated for its technical feasibility.

Ejector refrigeration technology was used for air conditioning of trains and large buildings (Garris et al., 1998). With a generator temperature between 85 and 95 °C, COPs reported are in the range of 0.2–0.33 for a condenser temperature between 28 and 32 °C (Murthy et al., 1991; Nguyen et al., 2001; Alexis and Karayiannis, 2005). Although Balaras et al. (2007) reported a much higher COP of 0.85 for a pilot steam ejector plant, this relatively high performance was only possible with a heat source temperature at 200 °C. Noeres (2006) very recently reported on the possibilities for further development of combined heat, cold and power production with steam jet ejector chillers. Although the simple construction of ejector systems is a great advantage, their COP makes it difficult to compete with the other heat-driven technologies. Garris et al. (1998) and Fischer and Labinov (2000) considered it unlikely that COP could be improved to a competitive level due to the inevitable energy dissipation in the working mechanism of conventional ejectors.

A variety of combined or hybrid systems have also been investigated. By selectively combining different technologies, creation of new functions or enhancement of performance was intended. These systems are generally more complex and expensive and will not be discussed here.

5. Discussion – affordable solar refrigeration

Although several solar refrigeration technologies are considered mature, until today, the total cooling capacity of the solar air conditioning systems in Europe is only 6 MW (Nick-Leptin, 2005). Although each technology has its own positive and negative aspects, high initial cost is a common problem.

Although differing in technical maturity and commercial status, the various solar refrigeration technologies discussed in the previous sections are compared in terms of performance and initial cost in Fig. 8. The three last columns indicate the specific cost of photovoltaic solar panels, the specific cost of thermal solar collectors plus specific engine costs and the specific chiller cost, respectively. Since the existing chillers based on these technologies differ widely in cooling capacity ranging from a few tens to several mega Watt, the efficiencies and the unit cost values assumed in Fig. 8 are those of the smallest machines available from the different refrigeration technologies. It is also noted that solar collector efficiencies listed in this article are only indicative and will depend on ambient air temperature and solar radiation.

Solar electric systems are assumed to be equipped with 10%-efficient solar photovoltaic panels with a unit price at €5/W_p (Solar Rechner). These solar panels convert a solar radiation of 1000 W/m^2 into 100 W of electricity and the various electric chillers transform this electric energy into cooling power according to their specified COPs. As shown in the figure, only magnetic chiller is comparable to vapour compression chiller in terms of solar panel cost. No other electric cooling technology is currently competitive with compression refrigeration technology in terms of total cost.

In order to generate the same amount of electricity, a thermo-mechanical system needs a high-temperature solar thermal collector and a heat engine. In Fig. 8, the efficiency of

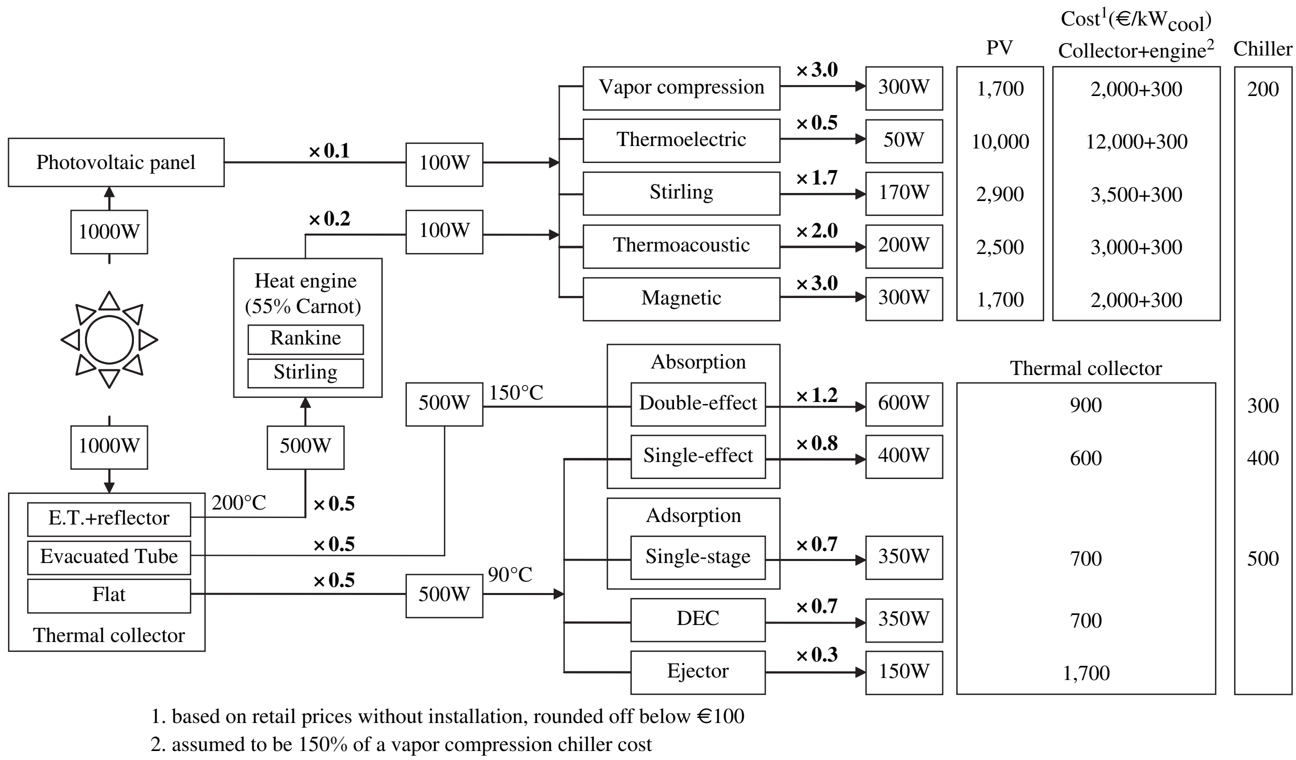

Fig. 8 – Performance and cost of various solar refrigeration systems.

a solar collector is assumed 50% at 200 °C and that of a heat engine is assumed 20% (56% second law efficiency). Among non-tracking solar collectors, a Sydney type collector, which is evacuated tubes with cylindrical absorbers and CPC concentrators (ca. €600/m^2, Collector Catalogue, 2004), may satisfy this application. As shown in Fig. 8, the cost for a thermo-mechanical system is far larger than that of an equivalent solar electric system even without the engine cost. A solar thermo-mechanical system is not likely to be cheaper than a solar electric system in terms of operation cost either.

Among the solar thermal systems shown in Fig. 8, a double-effect LiBr–water absorption chiller requires the highest driving temperature at 150 °C. A 50%-efficient evacuated tube collector at this temperature would cost approximately €550/m^2 (Collector Catalogue, 2004) and a double-effect LiBr–water chiller costs ca. €300/kW$_{cooling}$ (Peritsch, 2006). All the rest of the thermally driven chillers are equipped with a 50%-efficient flat collector at 90 °C, which costs ca. €250/m^2 (Collector Catalogue, 2004). The cost of a single-effect LiBr–water absorption chiller is estimated at ca. €400/kW$_{cooling}$ (Peritsch, 2006) and that of a single-stage adsorption chiller is estimated at about €500/kW$_{cooling}$ (ECN, 2002).

Although an ejector chiller would cost less than the other sorption chillers, its low COP would cost more for solar collectors. A desiccant system would also cost more than the other sorption systems due to the need of handling large quantities of air and water. The double-effect LiBr–water absorption and the single-stage adsorption systems are comparable in terms of total cost at around €1200/kW$_{cooling}$. The total cost of a single-effect LiBr–water absorption system is estimated as the lowest at €1000/kW$_{cooling}$.

Although Fig. 8 is based on ideal assumptions, it is clear that solar electric and thermo-mechanical systems are more expensive than solar thermal systems. Besides, these technologies are not compatible with the biggest solar infrastructure existing today, i.e. solar heating systems. Among the sorption cooling technologies, desiccant cooling can be a good solution for the applications where good indoor air quality is essential. But in general, high initial cost is likely to limit its application to large facilities. Absorption and adsorption cooling technologies are comparable in terms of performance. But presently, an adsorption chiller is more expensive than an absorption chiller. The low power density of an adsorbent tends to increase the price of an adsorption machine by requiring bigger components for the same capacity.

Current solar absorption refrigeration technology is not likely to deliver much financial benefit. This was shown in Henning (2004) and Balaras et al. (2007), where the annual cost of a solar system was always higher than that of a conventional (electric compression) system. The main reason is the high initial cost of a solar system, of which the largest portion is usually taken up by solar collectors. For the reduction of initial cost, an absorption chiller should be made to work with less or cheaper solar collectors. That is, either the chiller's COP should be increased or its driving temperature should be lowered. Considering the numerous efforts carried out in the past, it is unlikely that significant cost reduction can be achieved by merely improving the existing chillers. It would require development of new thermodynamic cycles and/or working fluids.

Regarding the direction of future R&D in solar refrigeration, it would better be focused on low-temperature sorption systems. This is because firstly, the cost of a solar collector system tends to increase with working temperature more rapidly than the COP of a sorption machine does. And secondly, high temperature-driven chillers would not be compatible with the existing solar heating systems which were originally designed to produce domestic hot water. Another important subject in the future R&D is the development of air-cooled machines. Currently, there is only one air-cooled machine for solar cooling in the market. Its performance, however, seems to become unsatisfactory for ambient air temperatures above 35 °C. A wet cooling tower is unfavorable in most of the small applications where regular maintenance work is impossible or in the arid regions where water is scarce.

6. Conclusions

A variety of options are available to convert solar energy into refrigeration effect. This review lists the main options and ranks the options according to their reported performance and the required investments per kW cooling.

Solar thermal with single-effect absorption system appears to be the best option closely followed by the solar thermal with single-effect adsorption system and by the solar thermal with double-effect absorption system options at the same price level.

Solar thermo-mechanical or solar photovoltaic options are significantly more expensive. Here the vapour compression system and magnetic systems are the most attractive options followed by the thermo-acoustic and Stirling systems.

Desiccant systems and ejector systems will be more expensive than the first three systems but since these systems require specific equipment their exact position is difficult to identify.

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