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Review and recent improvements of solar sorption cooling systems

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ABSTRACT

This paper aims to provide the current state of the art of solar sorption systems. Through comprehensive literature review of solar sorption systems, it was concluded that these technologies have several limitations and drawbacks. The low performance and the high cost are the main disadvantages of these technologies. However, solar cooling is considered attractive because solar radiation is in phase with the demand for cooling. Due to its attractiveness, the research is still going and focusing on solving the technical, economical, environmental problems, to achieve high performance and low cost of solar sorption systems. Improvements through investigating geometrical, system configurations, physical parameters, and operational modes on the performance of solar thermal sorption cooling systems are presented. A survey of the new configurations, novel additions, new techniques, new methodologies are also presented in this paper. Several cases studies in different climatic conditions are summarized. Economic feasibility for absorption and adsorption systems is discussed. It can be concluded that cost and energy effective solar sorption systems can be developed if suitable combinations of system components with operating conditions are selected. Finally, there is still the need for more research on solar sorption systems to make them both cost and energy competitive with the conventional cooling technologies.

Contents

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Nomenclature

A_C	Area solar collector (m)
F_R	Heat removal factor
I_t	Solar insulation
Q	Heat flow across heat exchanger(W)
$Qaux$	Auxiliary heater capacity (kW)
Qs	The nominal capacity of the chiller (kW)
Q_u	Useful collected energy (KJ)
T	Temperature/temperature of solution (°C)
T_{fi}	Inlet fluid temperature (K)
T_{fo}	Outlet fluid temperature (K)
T_a	Ambient temperature (°C)
T_{pm}	Plate mean temperature (K)
T_s	Storage temperature
ηc	Collector efficiency

Subscripts

a	Adsorpate bed
abs	Absorber
amb	Ambient
chill	Chilled water
cond	Condenser
des	Desorper
evap	Evaporator
gen	Generator
o	Outlet
CPC	*Parabolic trough collector*
SCP	*Solar cooling power*

1. Introduction

The nonrenewable sources in the world are finite and they eventually will be consumed. Currently, it is estimated that 80% of electricity is generated by fossil fuels. Energy requirements for refrigeration and air conditioning are increasing each year. Hence, utilizing renewable energy sources for meeting this growing energy demand is taken seriously. Solar energy is most appropriate option among other renewable energy sources because solar energy level is in phase with air—condition demand.

Solar cooling can be classified as electric and thermal powered systems. The electric powered systems utilized PV cells for converting solar energy into electricity energy and then using this electric energy for generating cooling power using conventional methods. The main thermal solar cooling systems are; absorption, adsorption, and desiccant. These systems have the ability to operate with low energy source using environmental friendly refrigerants. It is found that electric cooling system using PV is more expensive than solar thermal cooling systems [1]. Furthermore, solar energy conversion by PV systems is significantly less than the energy conversion using thermal cooling system, i.e. PV convert less than one third of solar radiation while thermal systems convert most of solar radiation [2]. Due to these advantages and findings, solar thermal systems receive significant attention by many researchers recently [3].

The objective of this paper is to provide literature review about recent development in the field of solar sorption cooling system technologies. These developments aimed to overcome the low performance and high cost of these systems. Several solar thermal cooling configurations, cooling mechanisms, performance, developments, and designs, including the cost and feasibility, are provided and discussed. The performance of several different available and actually installed solar thermal technologies in the world used for cooling or air-conditioning purposes are provided.

Recent research on solar thermal cooling has focused in developing methods to meet cooling demand with high efficiency and low cost. Solar cooling systems are found in literature in several types and configurations. Fig. 1 present summary of classification for solar cooling methodologies [4,5].

The basic configuration of solar thermal sorption cooling systems is shown in Fig. 2. Adsorber is used in the adsorption system in instead of the absorber. The main components of solar thermal cooling system are; solar collector, storage tank, condenser, evaporator, heat exchanger, expansion valves and the refrigeration chamber. In an absorption system, the refrigeration chamber is an absorber while in the adsorption system it is an adsorber.

The absorbed solar radiation by the solar collector is transferred to the heat transfer fluid. Solar collectors are divided into concentrator and nonconcentrator. Flat Plate Collector FP and Evacuated Tube Collector VT are examples of nonconcentrator. The solar collector efficiency is defined as the ratio between the absorbed energy and the solar irradiation [6]. Convection and conduction are the main heat transfer modes in solar collector. The efficiency of flat plat collectors is relatively low due to their limited outlet temperature. Evacuated tube collectors have much higher efficiencies where temperature can reach easily *up to 120* °C. Moreover, recent development of innovative evacuated flat plate collectors which can capture both beam and diffuse radiation can deliver high temperatures (up to 200 °C) without concentration [7].

Fig. 3 shows the flowchart for the main equations and steps for designing and sizing solar thermal sorption cooling system. Detailed presentation about the thermal performance calculations of flat plate collector can be found in [8]. Furthermore, the most widely used tracking modes for PTC are presented in [8].

2. Solar absorption cooling system

Among solar thermal cooling technologies, absorption system is the dominant solar thermal system due to

- its high COP,
- its ability to operate with low temperature;
- available for large-scale applications;
- their cost is lower than the rest of thermally-driven air-conditioning systems;
- and low noise.

The COP of an absorption system typically ranges between 0.5 and 0.8 [9,10]. The main factors affecting the annual performance of an absorption system are external temperature, solar irradiance, solar collector performance, cooling load scheduling, and absorption chiller dimensions, the system configurations, control logic of the whole system, and operational conditions. The main absorption chiller operating parameters are: the desorber temperature and the cooling temperature achievable by the heat rejection system. Usually, the system relies on an auxiliary heater to achieve the desired desorber temperature, in case insufficient solar radiation. Information about tested configurations, main technical system features and findings, in terms of yearly and/or seasonal system performance, of the relevant studies found in literature are listed in Table 1. One of the major drawbacks of this system is the narrow range of absorption chiller temperature to work properly [11]. The most used configuration is the series connection where the absorption chiller is supplied only by means of the hot storage tank. The parallel connection, where heat can be supplied to the absorption chiller directly by means of the solar thermal collectors or from the storage tank, is still developing technology [12]. In this configuration, solar collectors store surplus heat meanwhile supplying the chiller when high solar irradiance.

Table 1
Solar absorption summarized results.

Ref. No	Solar collector		Absorption chiller (KW)	Storage capacity (m^3)	Solar fraction%	Collectors efficiency%	COP	Prototype Location
	Type	Area m^2						
[11]	VT	96	8	3	43.5%	–	0.31	Shanghai
[52]	VT	72	35	–	81%	–	–	Thailand
[53]	FP	50	6–10	0.040/m^2 of FP	56%	–	–	Spain
[29]	FP	500	100 (2stage)	–	–	55%	0.38–0.43	South china
[54]	VT	35	3.5kw	–	0.8	–	–	Malaysia's climate
[55]	VT	108	35.17	1.5	31.1–100%	35.2–49.2%	0.37–0.81	Oberhausen, Germany
[56]	FP	30	16	0.8	–	–	0.8–0.91	Tunisia
[61]	EVFP					61	0.69	
[62]	Fresnel	352	174 kW gas/solar (double stages) powered double effect LiBr–water		75	35–40	1.1–1.25	Spain
[63]	FP	90	30 kW					Reunion Island
[57]	Fresnel collector	352	134	75%	75%	(35–40)%	1.1–1.25	Seville
[58]	FP	49 .9	35	2	–	50%	0.42	Madrid, Spain
[59]	FP	37 .5	4.5	0.700	–	–	0.46–0.6	Zaragoza (Spain)
[60]	VT	12	4.5	1			0.58 (average)	Cardiff University, UK
[69]	FP	23 .3					0.78 (max)	Beirut, Lebanon

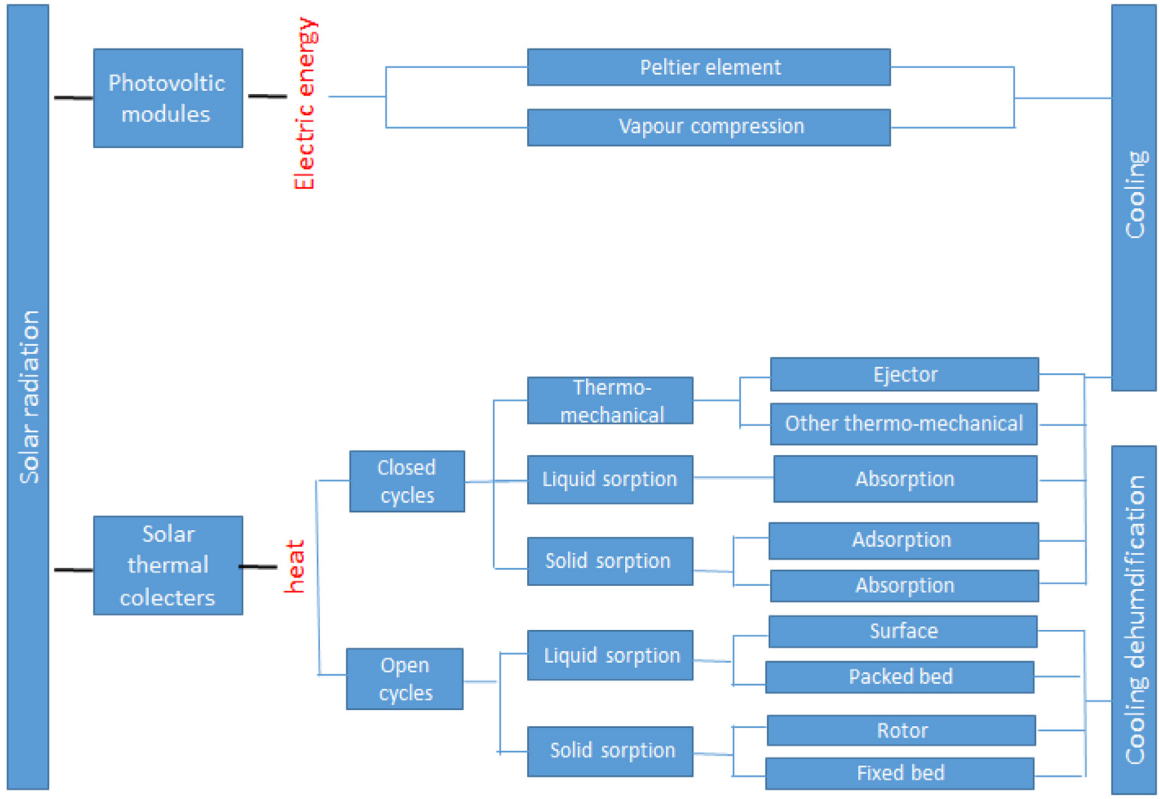

Fig. 1. Solar cooling Technologies.

2.1. The effect of collector types

Concentrating collectors can only utilize the direct normal irradiance (DNI) as opposed to FPCs and ETCs which can also harvest solar diffuse radiation. Type of collectors should be carefully selected to make solar absorption cooling system more attractive from energy and cost view point. It has been concluded through studies found in the literature that the best choice for solar collector type depends on several factors such as; number of absorption effect, solar radiation level and percentage of DNI. Kundu et al. studied the effect of solar collector types on the performance of solar absorption systems [13]. Their results showed that parabolic collector is the ideal choice for the absorption system. However, the authors did not investigates the fraction of DNI to the defuse irradiation. The evacuated flat plate collector powered double-effect absorption chiller maintains an acceptable range of thermal efficiencies [14]. The combination of double-effect chiller and evacuated flat plate collectors achieves a good energetic and economic performance in a wide variety of climatic conditions [15]. The combination of triple-effect chiller and parabolic trough collectors results in the most energy-efficient and cost-effective plant for climatic regions with very high DNI faction [15]. The energy performance of solar double-effect absorption cooling system using EFPC units tested in Saudi Arabia is higher than that of a reference system which was based on concentrating solar thermal collectors [16].

2.2. Thermal storage tank

The solar absorption system can be configured with or without thermal storage tank. Not utilizing thermal storage has five main advantages; (a) allows earlier cooling effect, (b) leads to faster time for the absorption chiller to reach its generator temperature, (c) higher cooling power peaks are achieved, (d) less cost, (e) simpler and smaller. On the other hand, without thermal storage tank, the absorption chiller cannot meet thermal load in the late afternoon because of lower generator temperatures. Furthermore, thermal storage systems are necessary to overcome the random nature of solar radiation levels, variation in cooling demand, and overcome the mismatch between the supply and demand of energy. There are two basic configurations of a thermal storage tank in a solar cooling facility, i.e. a well-mixed temperature and stratified thermal storage tank. Experimental results show that for solar cooling applications, having a well-mixed temperature in the thermal storage tank produces more daily cooling energy than in a stratified one [17].

Design of storage tank, its control strategy, and thermal stratification play an important role in the system efficiency. The size of thermal storage plays key role in determines the energy performance of the solar absorption cooling system. Because $LiBr/H_2O$ absorption chiller can operate at high temperature reaching boiling temperature, the volume of the storage tank must be designed properly to prevent energy loses by maintaining the temperature close to the chiller operating temperature. Hang et al. found that the solar fraction varies between 51% and 57% when the hot storage tank volume increases from 0.01 m^3 to 0.11 m^3 per unit area of solar collector [18]. Mateus and Oliveira reported that the optimal hot storage tank volume ranges between 0.05 to 0.11 m^3 per collector area for the cooling office building [19]. It has been reported that stratified storage tank achieves higher heat output and higher COP than the conventional fully mixed storage tank [20,21]. Li reported that stratification within storage tank affect significantly the overall solar heating or cooling performance [22]. It is found that heat removal factor has more effect on the system performance compared to the thermal stratification for small collector area. Furthermore, there is an optimum value for (tank volume/collector area) for a solar powered absorption air conditioning system. Li et al. compared between the performance of solar absorption air conditioning system with a partitioned hot water storage tank and

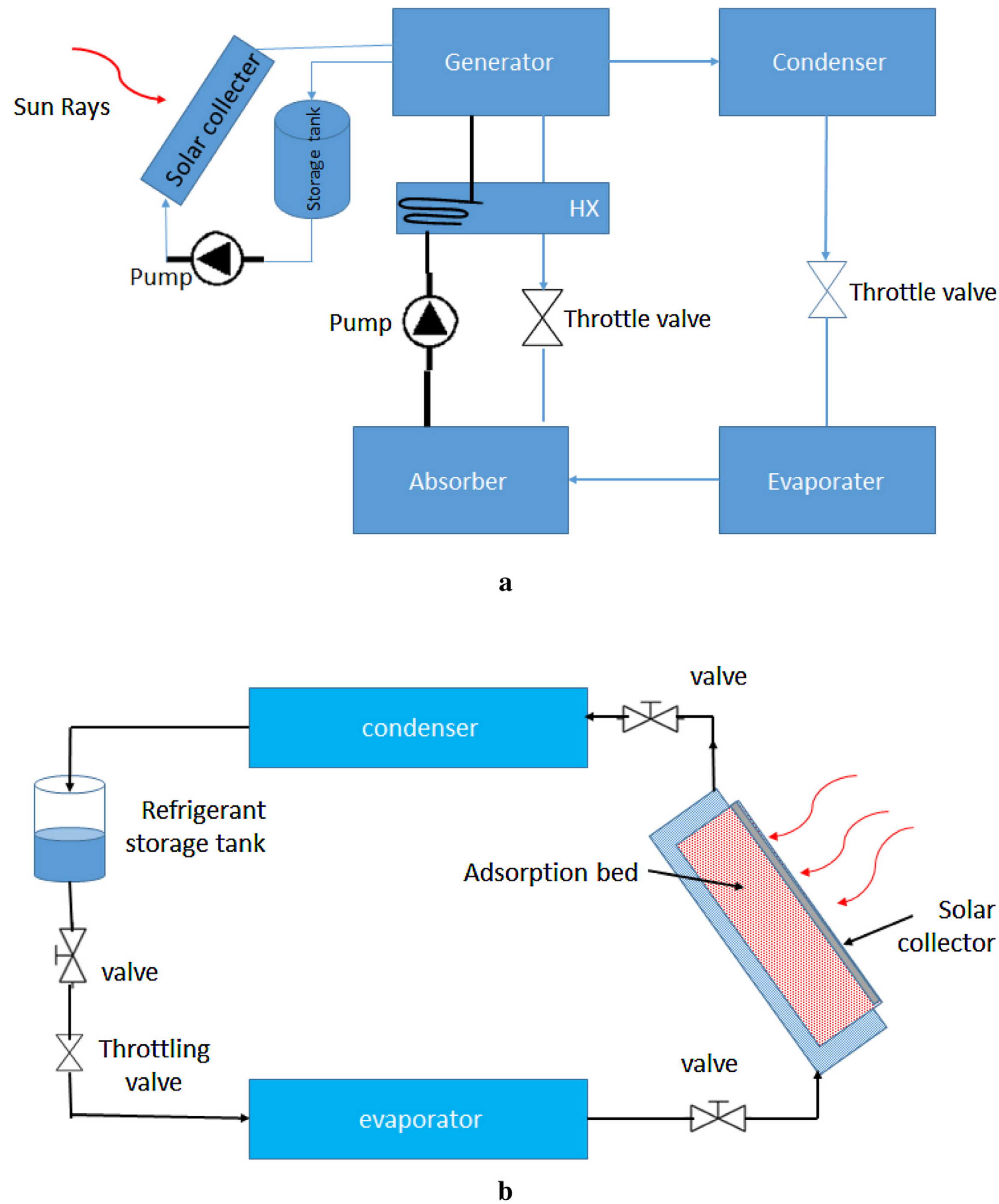

Fig. 2. Basic configuration of solar thermal sorption cooling system technologies, a) absorption [64], b) adsorption system.

traditional whole-tank mode [23]. It is found that a 15% increase in solar COP can be achieved with partitioned mode compared to traditional storage system.

Recent study comparing between the solar thermally driven cooling system with or without storage tank concluded that the system with storage tank produces higher cooling capacity at the beginning and at the end of the day [24]. On the other hand, system without heat storage has the advantage of higher electrical efficiency, but it requires larger solar collector area in order to be able to cover the building cooling demand [24]. For continuous operation of the cooling system, auxiliary heater can be used to power the absorption chiller when solar energy is insufficient. The cost of adding an auxiliary heater is less than the cost associated with increasing the thermal storage tank capacity.

2.3. Absorption chiller type for solar cooling

The main configurations of absorption chiller are; the single effect absorption system, the double effect absorption system, triple effect absorption system, and the generator–absorber heat exchange GAX absorption chiller. The single effect absorption system shown in Fig. 2a has two circuits: the refrigerant circuit from generator to absorber and LiBr–water solution circuit from the absorber to generator through the heat exchanger. Among all useful collector energy: $Q_u = A_c \cdot I \cdot \eta_c$

performance of a solar collector: $\eta_c = F_R \cdot \left[(I_s \alpha) - U_L \cdot \frac{T_{in} - T_{out}}{I} \right]$

Flat plate collector

$$Q_u = A_c F_R \left[\alpha \tau I_s(t) - U_L (T_{pm} - T_a) \right]$$

$$F_R = \frac{\dot{m}}{A_c} \frac{C_p}{U_L} \left[\frac{T_{fo} - T_{fi}}{\frac{S}{U_L} - (T_{fi} - T_a)} \right]$$

$$U_L = U_t + U_b + U_s$$

Evacuated tube collector

$$Q_u = \dot{m} C_w (T_f - T_i)$$

$$Q_u = A_c F_R \left[\alpha \tau I - U_L (T_i - T_a) \right]$$

$$U_L = \frac{\dot{m} C_p (T_i - T_f)_{av}}{A_a (T_m - T_a)_{av}}$$

For good quality tubes, the heat loss coefficient varies from 0.5 W/m²K (at $T_m - T_a = 25$ °C) to 0.65 W/m²K (at $T_m - T_a = 65$ °C). For lower quality tubes, the coefficient varies from 0.7 W/m²K to 0.9 W/m²K for the same temperature range.

Parabolic trough collector

$$Q_{u.th} = A_{ap} F_r \left[I_t - \frac{A_{r.ext}}{A_{ap}} U_l (T_{f,i} - T_a) \right]$$

$$A_{r.ext} = \pi D_{r,ext} L, \quad A_{ap} = (W - D_{r,ext}) L$$

$$\frac{1}{U_l} = \frac{1}{C_3 (T_{pm} - T_c)^{0.25} + \left[\frac{\sigma (T_{pm}^2 + T_c)(T_{pm} + T_c)}{\left\{ \frac{1}{\varepsilon_p} + \frac{D_{r,ext}}{D_{ci}} \left(\frac{1}{\varepsilon_C} - 1 \right) \right\}} \right]}$$

$$+ \left(\frac{D_{r,ext}}{D_{co}} \right) \left(\frac{1}{h_w + \sigma \varepsilon_c (T_c^2 + T_a^2 (T_c + T_a))} \right)$$

$$C_3 = \frac{17{,}74}{(T_{pm} + T_c)^{0.4} D_{r,ext} \left(D_{r,ext}^{-.75} + D_{ci}^{-0.75} \right)}$$

$$\left(\frac{T_c - T_{amp}}{T_{pm} - T_{amp}} \right) = 0.163 \left(\frac{D_{r,ext}}{D_{co}} \right)^{0.4} h_w^{-0.67} \left[2 - 3\varepsilon_p + \frac{(1 + 3\varepsilon_p) T_{pm}}{100} \right]$$

$$\text{for } 513 < \text{Tpm} < 623$$

$$T_{s,new} = T_{s,old} + \frac{\Delta t}{(m \cdot C_{vw})_s} \cdot \left[Q_u - Q_L - (U \cdot A)_s \cdot (T_s - T_o) \right]$$

$$Q_s = \sum (L_i - Q_o) \cdot \Delta t_i$$

$$Q_e = m_{chill} \cdot c_{pwater} \cdot (T_{ie} - T_{oe})$$

$$Q_G = m \cdot c_p \cdot (T_{iG} - T_{oG})$$

$$Capacity_{chiller} = \frac{\sum_{h=1}^{h=24} cooling_{load}}{Hours_{available}}$$

$$COP = \frac{Q_{ref}}{Q_G}$$

Fig. 3. Flowchart of main equations used in thermal analysis of solar sorption system.

absorption chiller types, single effect absorption chiller has a simple configuration and fewer components. It can be used for both air conditioning and freezing application. The dynamic analysis of a single-effect LiBr–H2O absorption chiller was studied by [25–27]. The thermal mass of the condenser has the highest effect on the COP and the exergetic efficiency and both can be improved when the effects of all thermal masses are studied simultaneously [25]. The evaporator temperatures ranges between −30 °C and −20 °C with COPs around 0.60 can be obtained at generation temperatures ranges between 80 °C and 110 °C [28]. The single-effect cannot operate at high condensation temperature and require relatively high generation temperatures.

Double-stage thermo-cycle of LiBr/H_2O system shown in Fig. 4 consists of high pressure and the low pressure stage. As shown in Fig. 4, after the vapor is condensed in high pressure stage, it transported to the evaporator in the low pressure stage. Double-effect can operate with low driving temperature as opposed to single effect which requires moderately higher temperature [137]. This feature allows the system to work under unpredictable low levels of solar radiation. The double-effect system has a higher COP due to reducing the effect of irreversibility. Double-effect system is able to operate at high condensation temperature T_c which can reach up to 53 °C and low generation temperature T_g of 80 °C to achieve COP of 0.38 [29]. Tierney reported that small size double-effect absorption chillers combined with PTCs can save up to 86% as compared to typical solar single-effect chillers [30]. Calise concluded, through dynamic modeling of double-effect LiBr–H2O absorption chiller with parabolic trough collectors, that these systems are the most profitable renewable energy technology in Mediterranean countries [31]. However, the author did not compare the cost of the double-effect LiBr–H2O high temperature absorption chiller against solar single-effect chillers.

Due to low cost of air-cooled chiller, several simulation studies are conducted to investigate their feasibility [32–39]. It has been reported that two-stages air-cooled ammonia–water absorption chiller is practically solar cooling applications. Llamas-Guillén et al. built an air cooled ammonia–lithium nitrate absorption cooling system driven by evacuated tube solar collector [32]. Using generator temperature around 110 °C, the system produced chilled water at evaporator temperatures below 10 °C for ambient temperatures 25–35 °C. The COP values of solar-powered air-cooled single effect LiBr–water absorption chiller is comparable to the values featured by solar-powered water-cooled absorption chillers [33,34]. This finding eliminates the drawbacks of using water-

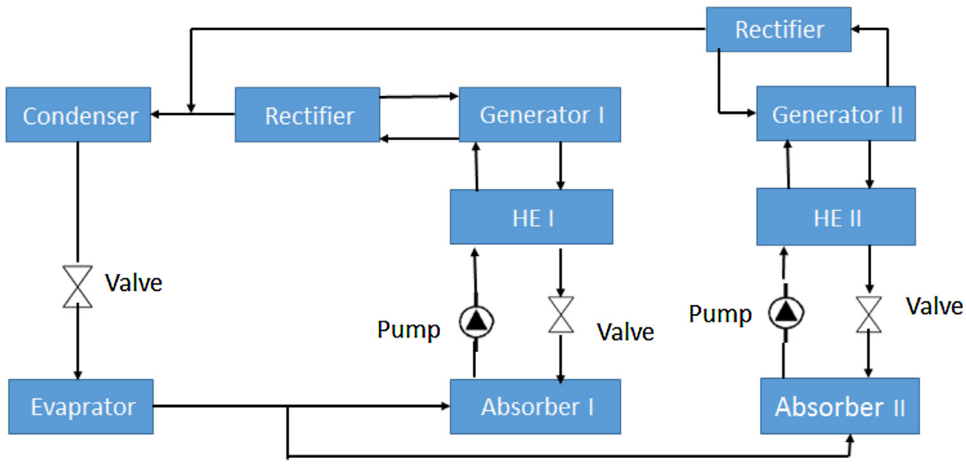

Fig. 4. Double-effect absorption cooling system.

cooled systems which require cooling towers, water consumption and require maintenance costs, not to mention the likelihood of *Legionella* infection. The COP of an air-cooled two-stage LiBr-H_2O absorption system studied by Kim and Ferreira was 0.37 when the environmental temperature was 35 °C with deriving temperature of 90 °C [36]. Furthermore, the average COP of two-stage absorption system was higher than single effect absorption system when using flat plate collectors. In order to verify the experimental results, Du et al. developed experimental study of air-cooled two-stage NH3-H2O system [39]. The thermal COP and electric efficacy reach 0.21 and 5.1 respectively when the prototype is driven by temperature equals to 85 °C of hot water, ambient air temperature of 29 °C, with evaporating temperature of 8 °C. To increase the COP, the low pressure absorber should be placed a head of double-stage system, while the middle or high pressure absorber should be placed at the end [39,138].

The triple-effect absorption chillers have the highest COP when coupled with high temperature solar thermal collectors. In order to achieve the desired high COP, the generator temperatures have to be above 150 °C [40,139]. However, the high COP is accompanied by high initial cost [34]. Plant consists of heliostat and central receiver used to power triple-effect absorption chiller integrated with an ejector and cascaded cycles is found suitable for refrigeration application (80 °C–50 °C) from thermodynamic viewpoint [41].

Fig. 5 summarizes the COP and the required deriving temperatures for single-effect, double effect, and triple effect chiller. As can been seen in Fig. 5 that using high temperature solar collectors with multi-effect absorption chillers leads to significant increase in COP [40]. The driving heat source temperature for single-effect chillers is about 80–100 °C, while their COP is limited to around 0.7. Double effect is about 100–150 °C, while their COP can reach up to 1.4. Moreover, triple-effect chillers require driving temperatures of around 180–240 °C, and can reach COPs of up to 1.8. It is clear that higher effect absorber has higher COP, if a high temperature heat source is available. It is not clear from the literature whether solar-powered multi-effect absorption chillers can be competitive because they require expensive collectors, pipework, tracking, maintenance, and lower solar gain per unit area (work with only direct normal irradiance).

The generator–absorber heat exchange GAX absorption cycle shown in Fig. 6 has attracted many researchers recently since the internal heat recovery plays a positive role. The solid line represents the GAX cycle while the dotted line represent the conventional single effect cycle. The performance of GAX cycle is much better than that of the single effect cycle [42]. The improvement in COP using absorber heat recovery cycle, simple GAX and branched GAX cycle are about 10–20%, 20–30% and 30–40% respectively compared to conventional single effect system for the same set of operating conditions. Velázquez and Best performed thermodynamic analysis for air-cooled GAX system. The system was driven by solar energy and hybrid natural gas [43]. For ambient air up to 40 °C with a relative humidity of 24% as cooling source, the COP of cooling and COP of heating and the heat recovery were found to be 0.86, 1.86 and 16.9 kW, respectively. Zheng et al. [44] simulated GAX cycle and single stage ammonia absorption system. They used the concept of exergy coupling to study the cycles. The absorption cycle was divided into the heat engine and heat pump sub-cycles for thermodynamic analysis. The results of the study showed that the exergy demand of the heat pump subcycle in the GAX cycle and the exergy demand of the single stage cycle was the same. When the generation temperatures is too high for a single effect chiller and not enough to drive a double effect LiBr-water absorption chiller, absorber generator heat exchanger AGX can be added [45]. Adding GAX to the cycle expand the workable range of generation temperatures. For example study conducted by Xu et al. showed that the COP for the GAX cycle with generation temperatures of (93–140) °C ranges between 0.75 and 1.08 [45]. Moreover, experimentally, the COP for GAX cycle using LiBr-water the ranges between 0.69 and 1.08 under generation temperature between 95 and 120 °C [45].

2.4. Coefficient of performance

The coefficient of performance of a refrigeration system is defined as the ratio between its ability to remove heat from a cold source to the supplied energy for its operation. This ratio is a measure of the efficiency of the cooling system. The solar COP is defined as:

$$COP = \frac{Q_e}{Q_s}, \tag{1}$$

where Q_e is the cooling effect produced at the evaporator, and Q_s is all the solar energy received by the solar collector surface. Although solar absorption systems are more feasible than the adsorption system, their high initial cost and low efficiency prohibit their commercial spread. These facts provoke significant number of studies to overcome these disadvantages. The COP of the absorption system has increased by using concentrator solar collector, double effect chiller, advanced control and improved chiller efficiency [46]. COP increases with increasing generator inlet temperature and increasing surface area of both absorber and heat exchanger

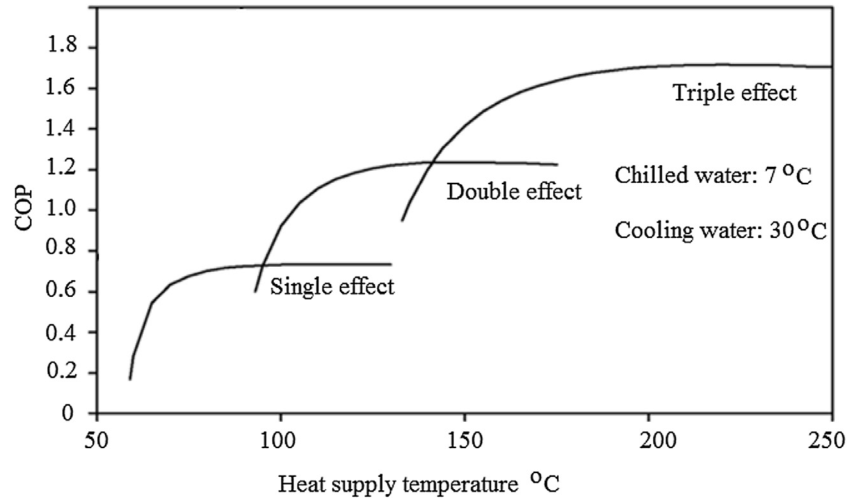

Fig. 5. Performance of different type of absorption chillers with solar collector types [40].

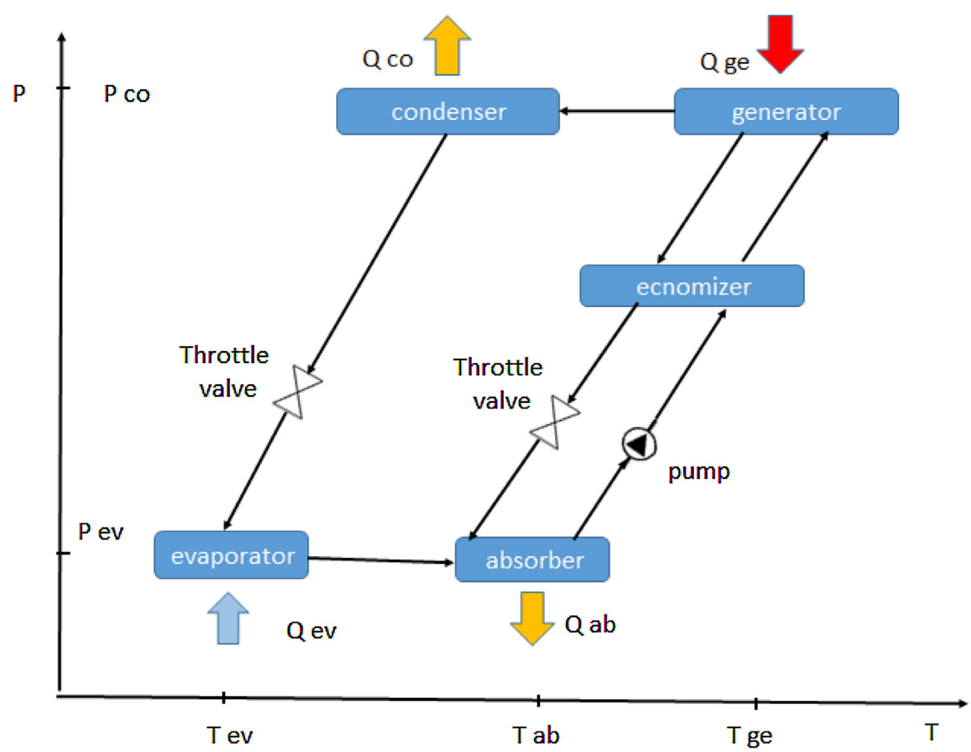

Fig. 6. Schematic diagram of GAX absorption cycle.

[47]. Moreover, COP increases with increasing water storage tank temperature.

High values of outdoor temperatures have negative effect on COP, especially when the dry cooling tower is used to evacuate the absorption heat. Palacin et al. used open geothermal cycle as a heat rejection tank to remove this negative effect on chiller COP [48]. Eicker et al. made comparison study between geothermal, wet, and dry geothermal heat rejection tank and found that the electrical COP is between 4 and 8 when using dry heat rejection [49].

Both external and internal irreversibilities have high negative impact on the performance. External irreversibility found in heat conduction, mass transfer, friction, eddy and others. Moreover, internal irreversibility such as (heat engine cycle/generator-absorber and refrigeration cycle/evaporator-condenser) has higher impact on reducing the performance [50]. Fathi et al. studied the impact of irreversibility in solar absorption refrigerator by using four reservoir models each of them has internal and external irreversibilities operated at the maximum cooling load [51]. They compared their modeling results with single-stage system using semi-empirical model. Their results show that both models predict similar values of COP.

2.5. Control strategy and operation modes

Arrangement of the system elements as well as control approach can be used to improve the energy performance of the solar absorption chillers. Throughout the literature survey, it is found that auxiliary heater and storage tank are arranged in both series and parallel configuration. Albers developed a new control strategy to govern the cooling capacity of absorption chillers by changing the hot and cooling water temperatures simultaneously, leading to a 5% reduction in the operating cost of the system [65]. Liao and Radermacher presented a new control strategy based on the variation of the evaporation temperature set point as a function of the condenser to avoid crystallization in lithium bromide chillers condensed by air [66]. Bujedo et al. studied the effect of different control strategies used in a solar cooling installation [67]. The three controls are: the first with fixed flow masses, the second adapting the temperature on the condenser as a function of the generator temperature, and the third adapting the condenser temperature and the flow mass on the generator as a function of the system loads. The results showed that both second and third control have better results in comparison with the base case (first control strategy). Furthermore, the best result obtained when implementing the third strategy (dual control). Shirazi et al. evaluated the effect of control strategy on overall thermal performance of solar heating cooling absorption system [20]. The solar loop in the first scenario uses a constant speed pump, while in the second strategy; the solar loop pump was equipped with a variable speed drive to achieve a constant outlet temperature just above the manufacturer's rated temperature for the chiller. The collector outlet set-point temperature is varied according to the hot water temperature required by the absorption chiller in the third strategy. Their simulation results showed that highest solar fraction of the plant was achieved by third control scheme. Furthermore, the simulation results also indicated that the parallel auxiliary-heater arrangement resulted in 13% and 9% increase in the total solar fraction and the average collector efficiency compared to the series configuration. Finally, it has been concluded that around 20% higher solar fraction can be obtained through the control strategies and operational modes without any considerable additional capital cost to the plant.

2.6. Economic feasibility

Economic feasibility depends on energy policy, price of oil, technology development, and price of solar collector. The economic feasibility of the solar cooling system is measured by payback period *Pb*, net present value *NPV*, present worth value, cost of unit energy, and cost of primary energy saved ($C_{p,saved}$), annual cost comparison method. Detailed description of these indicators are given in [68].

Up to now, solar cooling systems are more expensive than conventional cooling systems. Several studies discussing the economic feasibility of absorption solar technology have been carried out at different locations around the world. The aims of these studies are to improve the system performance and select the best configurations. All of these studies showed that the performance of the solar cooling systems is highly nonlinear functions of collector size, types of collector, locations, cooling and heating loads, operational conditions, and size of backup etc. It has been reported that the solar absorption cooling systems can be economically feasible under specific operational conditions and configurations. Increasing COP reduces Pb and increases the NPV of the solar absorption air-conditioning system. COP can be increased by decreasing the operating temperature of the condenser or increasing the operating temperature of the evaporator, or both. Under Greece climate weather conditions, the payback period of absorption system exceeds the lifetime of system but NPV is positive [69]. It is found that absorption system with LiBr/H_2O is more economical attractive when using an evacuated tube flat plate solar collectors, minimizing the heat losses, and increasing vacuum pressure [70]. Shirazi et al. analyzed the economic of four configurations of solar heating and cooling systems based on power single-effect LiBr eH2O absorption chiller with evacuated tube collectors for Australian buildings [71]. The study concluded that the economic performance of all configurations is still unsatisfactory (without subsidies) due to their high capital costs. However, if a government subsidy of 50% is considered, the results suggest that for system (with a mechanical compression chiller employed as the auxiliary cooling system and reduced chiller size) can be economically feasible, achieving a payback period of 4.1 years and solar fraction of 43%, contributing to 27.16% decrease in the plant primary energy consumption. It has been reported that single-effect chiller system with an evacuated tube collector array has better energetic and economic performance compared to multi-effect absorption chillers coupled with concentrating collectors in low DNI fraction [71]. On the other hand, in climatic regions with very high DNI faction, the combination of triple-effect chiller and parabolic trough collectors results in the most energy-efficient and cost-effective plant configuration, achieving the smallest solar field and the lowest simple payback period [71]. In regions with very low solar irradiation, solar multi-effect chillers are not an efficient option. The combination of double-effect chiller and evacuated flat plate collectors achieves a good energetic and economic performance in a wide variety of climatic conditions. A solar LiBr–H_2O absorption system is economically feasible for use in large commercial buildings that consume electrical energy at higher rates for location with high DNI. Furthermore, the solar absorption system is more feasible and more economically viable than the solar PV-vapor-compression system. A study for eastern province of Saudi Arabia showed that when electricity rate is $0.16/kW h, the payback period for solar absorption system is 9 and the NPV is positive at the electricity rate of $0.0533/kW [72]. Eicker et al. through several recent studies [73–75], concluded that payback periods of about 10 years with today's energy prices can be achieved if the investment costs are reduced by 30–70%, depending on the location and dimensioning, system design and cooling load data. Buonomano recently built solar double-effect absorption cooling system using EFPC prototype [76]. It is found that the proposed prototype could achieve an acceptable profitability, provided that incentives were appropriately designed.

Combining solar cooling system with heating either domestic water heating or space heat can significantly enhance the feasibility of the system. Ghaddar et al. concluded through economic assessment of solar absorption cooling system is marginally competitive only when combined with domestic water heating under Lebanon climate conditions [77]. Tiago et al. evaluated the potential of combining solar absorption cooling and heating systems for building applications for three European cities: Berlin (Germany), Lisbon (Portugal), and Rome (Italy) [78]. Their results showed that combining solar absorption cooling and heating system can reduce total cost and reduce CO2 emission. Moreover, it was found that combined heating and cooling system is more attractive when natural gas is used as system backup energy.

Tawatchai et al. concluded that under Thailand climate condition, solar powered absorption chiller can save up to 98.56% of total electricity consumption of vapor compression system [79]. Shekarchian et al. compared between the vapor compression and absorption systems in different climate in Iran [80]. They showed that increasing the COP of absorption chiller by 0.1, a significant cost saving can be achieved. Louise et al. showed that the production cost of cooling in Swedish municipality can be reduced by 170% by using solar heat driven absorption chiller instead of conventional vapor compression chiller [81].

Recently, four different solar collectors are tested in a single stage absorption chiller operating with LiBr–H2O for Athens (Greece) in summer [82]. Considering the capital cost of the solar system (collectors and storage tank), the evacuated tube collectors (ETC) seem to be the most suitable solution for the examined study-case. It is found that with a capital cost of 60 k€, ETC system is the most feasible solution for covering the cooling load of 100 kW. FPC system is the second choice with 66 k€, while PTC system follows with 78 k€ and CPC system with 84 k€.

Otanicar et al. presented current and projections about future costs of solar thermal conditioning systems [83]. They used ceiling-floor approach to determine current and future costs of each system component. The analyses are conducted assuming fixed amount of cooling 5 tons (17.58 kW) and the solar irradiance is assumed to be a peak value of 1000 W/m^2. The storage system is capable of providing up to 8 h of energy storage. Furthermore, the following values for thermal collectors are taken as floor (and ceiling) prices for the given year: 2010 – \$0.83/Wth (\$0.84/Wth), 2020–\$0.68/Wth (\$0.89/Wth), and 2030 – \$0.50/Wth (\$0.95/Wth) (EIA, 2010). Thermal storage prices are estimated to currently be \$1585/m^3 and estimated this price will fall at 1% per year to 2030. Moreover, since the cost of thermal air condition system hard to predict with little historical data to draw from, they assumed the unit will have a constant price over time life (20 years). Fig. 7 shows the costs for the solar thermal cooling technologies. The International Energy Agency projects a drop of between 35% and 45% in total system cost for solar thermal cooling by 2030 [84]. The cost reduction predicted by Otanicar et al. [83] agrees with those expected by IEA [84]. It should be noted that NH3 absorption systems are because the A/C unit price is much lower \$5000 compared to \$20,000 for a LiBr system.

Thermo-economic performance comparison study between GAX absorption cycle and a hybrid GAX absorption cycle HGAX, showed that HGAX has better performance and higher unit energy cost [85]. The HGAX cycle uses compressor to raise the absorber pressure. Based on their results, the cost the HGAX cycle was 180.5 \$/GJ while the cost of standard GAX cycle was 159.1 \$/GJ at the same operation conditions.

2.7. Optimization analysis

Several studies aimed to develop tools for optimizing the performance of absorption cooling cycles to overcome the major drawbacks of these systems. Part of these studies found in the literature optimized the solar cooling systems either by component by component or variable by variable [86–88]. The main drawback of these approaches is that it is likely to fail in finding the global optimal solutions. to overcome these problems. Systematic methods based on mathematical techniques have been utilized for optimization of absorption cycle [54,89,90]. However, the literature has gap since the approaches have focused on either the absorption cycle or the solar system. A novel a bi-criteria mixed-integer nonlinear programming model that optimizes the design and operating conditions of integrated solar assisted thermal systems considering economic and environmental concerns is developed to fill this gap (Gebreslassie) [91]. The approach has been successfully applied for designing solar absorption cooling system under Barcelona and Tarragona climate conditions. The approach concluded that investing on the solar collectors has significant reductions in the environmental impact. Furthermore, the selection the best collector type depends on particular operating conditions and weather data. Regression analysis method is used to optimize the cooling system performance [46,92]. Regression analysis method considers relations between several important factors such as; solar fraction, area and slope of solar collector, and volume of cold and hot storage tank. The regression method allows mathematically studying the influence of different parameters on the objective functions and easily determining their sensitivity, which is difficult to perform with other techniques.

3. Adsorption solar cooling system

Adsorption solar cooling system is considered the second main technology of thermal solar cooling. This technology is found on ice making, making chilled water, and air conditioning. It is found very appropriate for grain storage applications [134]. The research activities in adsorption solar cooling sector are still increasing to allow them to compete with the well-known vapor compression system. The research has focused on solving the technical, economical, environmental problems, and achieve high performance and low cost. The main advantages of an adsorption system are; ability to work with wide range of driving temperature levels, noiseless, non-corrosive, and environmentally friendly [93,94,135]. On the other hand, this technology suffers from long adsorption/desorption time. Up to date, the high cost of adsorption chillers, the low performance of the adsorption chiller, and the size of the system, are the main factors that preventing significant market extent of this technology. Adsorption system requires two or more absorbers to provide continuous operation. Furthermore, acceptable efficiency can be achieved only by well-designed of both solar collector and disrober with good heat release characteristics [94]. The range of temperature source and adsorption working pairs are the main parameters affecting the system efficiency.

Adsorption cooling systems are classified by Best and Ortega into; intermittent adsorption; continuous adsorption, and diffusion. The high cost associated with intermittent cycle has limited their growth. Continuous cycle system is widely used and investigated due to its higher performance and its ability to produc**e** refrigeration and cooling. There are several technologies implementing adsorption continuous cycles; heat recovery adsorption system, mass recovery adsorption system, thermal wave adsorption technology [95], convective thermal wave adsorption technology [93], multi-stage and cascading technology [96–98], and hybrid systems. The solar adsorption cooling consists mainly of solar collector array, heat storage tank, chillers, pumps, fan coil unit, cooling tower and several valves as shown in Fig. 2b. The working principle of an adsorption process is presented in [99]. Thermodynamics modeling of system component of an adsorption cycle is given in [100].

The semi-continuous heat recovery adsorption cooling cycle is usually operated with two adsorption beds. The heat recovery process leads to a higher system COP. Multi-beds can be utilized to increase the COP, but this increases the complexity of the system. The detailed description of chiller mechanism is explained by [98,101,102]. In the two beds chiller system, evaporation and condensation of water in the hot and cold champers lead to heat and pressure changes between two beds. This process has three phases consequently; (i) the adsorption/desorption phase, (ii) the mass recovery phase, (iii) the heat recovery phase. Detailed descriptions of these phases are presented by [98]. Fig. 8 shows the schematic diagram of the two bed adsorption chiller consisted of evaporator, condenser, valves, and two adsorption beds. The cycle of the system begins with refrigerant at adsorber 2 which is heated by solar energy and transferred to the condenser in order to reach the heating-desorption condensation state. Then, the adsorber 1 which is connected to the evaporator is cooled to gather the adsorption refrigeration in it. The condensed liquid flows through valve toward the evaporator to realize adsorption refrigeration in it. Heater and cooler used in this model as a pump in the circuit to drive thermal fluid.

Heat and mass transfer in the bed have direct impact on the performance of adsorption cooling. The increase in the performance of the adsorper heat will increase the total adsorber heat transfer coefficient and enhance the heat transfer rate between the heat media and the adsorbent. Increase the performance of mass transfer will reduce the refrigerant diffusion time in adsorbent and shorten

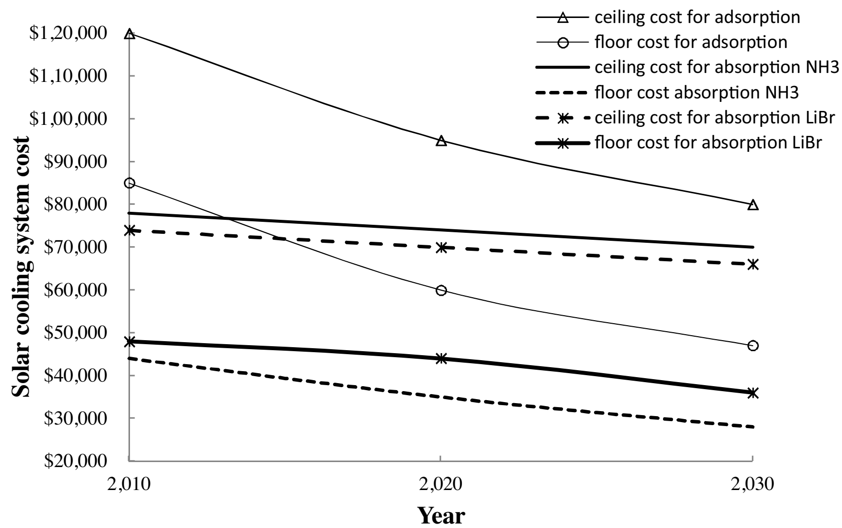

Fig. 7. Current and projected cooling system costs for solar absorption and adsorption cooling systems [83].

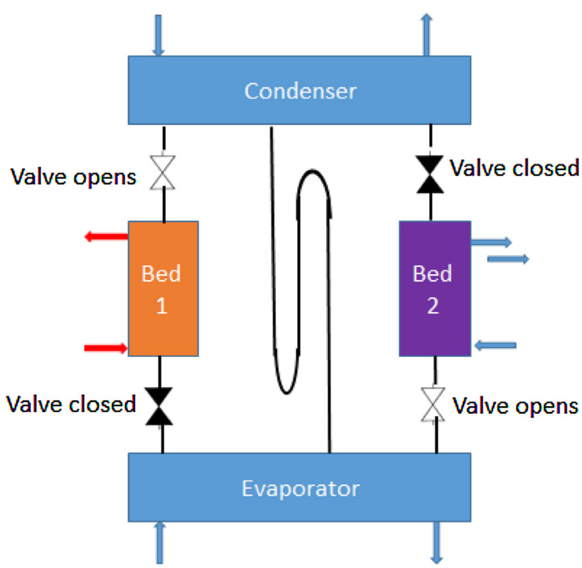

Fig. 8. Schematic diagram of heat recovery silica gel-water two-bed adsorption chiller.

the adsorption-desorption time. Several methods are developed to increase heat transfer area, such as; plate-finned bed, the spiral plate bed, and the pin-fin bed. Furthermore, coating method has been developed in which the slurry adsorption material is applied onto the solid wall of the tube to reduce the contact resistance between the adsorbent particulate and the bed wall. A few advanced cycling modes have been proposed to improve the performance of the adsorption cooling system. The heat recovery cycle has been studied in depth till today and put into practical application. Analysis of the mass recovery cycle showed that the COP can be increased as much as 100% in comparison to the simple cycle. As a novel cycling mode, the thermal wave cycle has presented the attractive result in the numerical study, while in the experiment it has not yet been verified practically. The surface cascade cycle consists of two stages of basic cycle. As the second cycle is operated at a lower driving temperature, the heat required is supplied by the desorbed vapor from the first stage and then the heat necessary from the outside heat source is decreased. The other advanced adsorption cycles include the multi-bed cycle, the multistage cycle, and the dual mode cycle etc.

3.1. Coefficient of performance

The performance of an adsorption cooling system can be improved by increasing heat and mass recovery and by developing multi stage and multi bed technologies. Three mathematical models are used to analyze heat and mass transfer in the adsorption bed namely; lumped parameter model, heat and mass transfer model, and thermodynamic model [100]. The lumped model focuses on the adsorbent bed, ignored bed geometry, and is described by ordinary differential equations. The heat and mass transfer model described by partial differential allows investigating the effect of different operating and geometric parameters on the COP. Finally, the thermodynamic model given by algebraic equation allows investigating the heat transfer and temperature on the COP of the system.

The general equation used to calculate the COP of adsorption cooling system is [4]:

$$COP = Te\,(Tg - Ta)\,/Tg\,(Ta - Tc) \tag{2}$$

where T_e is evaporating temperature, T_g is generating temperature, T_a is the ambient temperature. Liu and Leong [101,102] studied the effect of operating condition on the performance of an adsorption cooling system based on zeolite 13X/water working pair. They found that the adsorption temperature, generation temperature, condensing temperature, evaporating temperature have significant effect on system performance. Moreover, adsorption temperature and generation temperature have an optimal value. Furthermore, the fluid velocity has positive effect on optimal cooling power, while the driven temperature of heat exchange fluid has no effect on system performance. COP can be improved by; decreasing the cold water inlet temperature to the bed, increasing fluid velocity to the bed, enhance both heat recovery and mass recover processes, and increasing adsorbent mass [103]. Furthermore, it was found that heat recovery can improve the COP of system, while mass recovery can improve both the COP and cooling capacity [103].

The main reasons behind lower values of COP and solar cooling power SCP; heat losses from adsorber during cooling cycle and non- adsorbing mass [103]. These problems can be minimized by; magnifies the contact area between adsorbent material and heat exchanger and reduce the thickness of effective adsorption to the parts of millimeter to overcome the low conductivity [104]. Enhancing the thermal conductivity of the adsorption bed increases heat transfer rate. Furthermore, using ripped plate has a higher heat transfer performance compared to the plate adsorption bed

Table 2
Summary of solar adsorption systems.

Paper ref.	Solar collector		adsorption chiller output	Adsorption pairs	No. beds	Solar Radiation (Locations)	Country	Solar cooling COP
	Type	Area m^2						
[10]	FT	10.5	500 W	Theoretical study	1-bed	900–1200 W/m^2	Poland	0.23
[97]	VT	150	8.5 KW	Silica gel	2-chillers	18.5 MJ/day m^2 (Shanghai)	China	0.35
[94]	VT	–	12 kg/day	Activated carbon-methanol	2-bed, dual purpose	61.2 MJ/day m^2	Malaysia	0.44
[109]	VT	49.4	3.2–4.4 KW	Silica gel	1-bed	16–21 MJ/day m^2	China	0.1–0.13
[120]	FT	3.5–4.5	10 kg/day	activated carbon-ammonia	4-beds	650 W/m^2	Saudi Arabia	0.2
[91]	CPC	32.175	15 KW	Silica gel	1-bed	963.89 W/m^2	Spain	0.3
[119]	FP	2	–	Silica gel	1-bed	13–22 MJ/day m^2	Switzerland	0.1–0.25
[107]	Glass tube	–	6.9–9.4 kg/m^2	Charcoal methanol	1-bed	20 MJ/day m^2	13.6–15.9	0.46–0.50
[121]	FP	1.5	2–5 kg/day	activated carbon–methanol			Italy	0.045–0.11
[118]	FP W/tube			activated carbon–ammonia			Morocco	0.111
[122]	CPC	1.029		Zeolite–Water	1-bed	14.7 MJ/day	*Nigeria*	0.838–1.48
[127]	CPC	3	47–78 W (day 57.6–104.4 W (night	Activated-carbon-methanol	2-beds	560 W/m^2 (average)	India	0.196 (day 0.335(night)
[128]	FP W/tube	1.2	5 kg/day	Activated carbon–Methanol	1-bed	28.7 MJ/day m^2	Italy	0.08
[129]	FP W/tube	0.63		Silica gel–water	1-bed	19 MJ/day m^2	Algeria	0.083–0.09
[130]	FP	1		activated carbon AC-35/methanol		26.12 MJ/m^2	Algeria	0.14[a] 0.19[b] 0.21[c]
[123]	CPC	0.55	2.2 MJ/m^2 per day	Activated-carbon-methanol	1-bed		Spain	0.117–0.087
[124]	FP W/tube	1		Activated-carbon-methanol	1-bed	900 W/m^2 (max)	Canada	0.211
[115]	FP w/tube	1.2	5 kg/day (Ice)	Silica–gel þLiCl–Methanol			Italy	0.33
[126]	PTC W/tube	3.7		Olive waste–Methanol	1-bed	56.2 MJ/day m^2	Saudi Arabia	0.75
[131]	advanceded flat plate[d]	70	9.8 kW 9.5 kW 9.6 kW	silica gel/water pair 205	2-beds	23.2 MJ/day m^2 24.3 MJ/day m^2 25.2 MJ/day m^2	Greece Cyprus Egypt	0.408 0.418 0.415
[125]	FP	2	27.82 kg/day (Ice)	Activated-carbon-methanol	1 bed	24.18 MJ/day m^2	Egypt	0.618
[132]	FP	1		silica-gel and water		24418.2 [kJ][e]		0.061 (experimental)0.069 (simulation)
[133]	FP	0.25		activated alumina 75AA[f] 25AA[g] activated carbon			Indonesia	0.054 0.056 0.06 0.074

[a] Single glazed cover.
[b] Double glazed cover.
[c] Transparent insulation material (TIM) cover.
[d] Advanced flat plate collector is coated with chromium selective coating.
[e] Total solar energy received by the collector during a complete cycle [J].
[f] A mixed of 75% activated alumina and activated alumina.
[g] A mixed of 25% activated alumina and 75% activated carbon.

[103]. Moreover, it was found that using thin-tube has a higher performance than the tubular adsorption bed [105]. Furthermore, adsorption beds with fins have low heat resistance between the metal and adsorbent materials and the heat transfer rate [106]. Khattab et al. successfully increased COP by adding small pieces of blackened steel to the charcoal-methanol solid adsorption pair and covered the bed of glass shell [107,108].

Mass recovery phase plays an important role in increasing the cooling power of the system without affecting its COP. Khattab et al. investigated mass recover phase using three-bed silica gel adsorption chiller [107]. Their results showed that an increase in cooling production of system with mass recovery chiller compared to chiller without mass recovery when the temperature of heat source ranges between 60 and 90 °C. Luo et al. [109] used heat and mass recovery phase in silica gel adsorption chiller operating out-of-phase to improve efficiency and allows continuous cooling production. They incorporated methanol evaporator into two water evaporators by using the gravity heat pipe concept. Increasing the mass flow rate of cooling water through condenser increases the COP and specific cooling power SCP [108,109].

3.2. Effect of adsorbent pair

Many researchers have studied the performance of many adsorbents-refrigerants pairs and found that the adsorbent-refrigerant pair is crucial step in system design. The best pair choice depends on the application, i.e. whether it is for ice making, deep freezing, food conservation, air conditioning. The most widely used pairs are: activated carbon–ammonia, activated carbon fibers-methanol, activated carbon–ethanol, zeolite–water, silica gel-water, calcium chloride–ammonia. . .etc. [94,110]. Low cost, low toxicity, large vaporization enthalpy, chemical stability, low freezing temperature, good thermal conductivity, and small molecular dimensions are the desired features of refrigerant. As for adsorbent, the desired features are; high adsorption and desorption capacity, low specific heat; high thermal conductivity, low toxicity; and low cost. The most widely used pair is activated- carbon due to their properties such as; micro pure volume and high surface area. It has been reported that heat losses can be reduced significantly by using refrigerants having large vaporization enthalpy [111–113].

A comparison study conducted by Anyanwu and Ogueke [114]. concluded that COP from using zeolite-water is about 0.3 which is higher than COP of using activated carbon-methanol and activated carbon-ammonia. The activated carbon–ammonia pair is the best choice for low temperature applications, while, the zeolite–water pair is better suited for air-conditioning applications [114].

Activated carbon–ammonia pair requires high regeneration temperatures (above 150 °C). It requires high operating pressures, which enhances the heat and mass transfer performance and reduces the cycle time, also preventing the infiltration of air into the system. All previous factors and the high cooling capacity of ammonia, make it the best pair. However, the activated carbon has a lower adsorption capacity with ammonia than with methanol; high toxicity due to ammonia. Activated carbon–methanol operates at low regeneration temperatures and suited for refrigeration. It has large cyclic adsorption capacity, low adsorption heat, low freezing point and high evaporation latent heat of methanol. Nevertheless, activated carbon has poor thermal conductivity high toxicity and flammability. The silica gel-water is able to work at temperature below 100 °C, i.e. 55 °C–95 °C with COP equals to 0.403 [114,136]. This workable low temperatures make them ideal for solar energy applications. Moreover, water has the advantage of having a greater latent heat than other conventional refrigerants. This feature makes the pair very suitable for air-conditioning applications with high chilled water flows. This refrigerant has no crystallization problem and no corrosion problems. Low adsorption capacity and low vapor pressure are the main problems of this pair. It has been reported that non-condensable gas will cause a significant reduction in the system's performance due to the fact that this pair requires vacuum conditions in the system [114].

The regeneration temperatures for the zeolite–water pair can exceed 200 °C. The pair has high stability even at elevated temperatures, the latent heat of water is much higher than that of other traditional refrigerants. Due to the solidification temperature of water, this pair is only suitable for air-conditioning applications. Moreover, it is found that the specific cooling capacity of the systems using this pair is low. Water evaporation pressure is very low which slow adsorption rate and delay the mass transfer, and increase the sensible heat of the adsorber.

3.3. Applications of solar adsorption cooling

Adsorption cooling system used for many purpose i.e. ice making and refrigeration and air conditioning. Maggio et al. used the composite sorbent "lithium chloride in silica gel pores" and methanol as refrigerant [115]. The maximum theoretical COP was 0.33 and the maximum daily ice production was 20 kg per m^2 of collector. These results obtained for an ice-maker used 36 kg of adsorbent material and equipped with a solar collector area of 1.5 m^2. Li et al. used two different working pairs; the activated carbon–ethanol and the activated carbon–methanol in solar ice making [116]. Their experimental results showed that with a solar radiation of 15.2 MJ, the activated carbon–ethanol was not able to produce ice, while the activated carbon–methanol produced 2.55 kg of ice, with a solar COP of 0.113. At solar radiation of 8.2 MJ, once again, activated carbon–ethanol was not able to produce ice, while the activated carbon–methanol produced 2.6 kg of ice, with a solar COP of 0.105.

Solar adsorption refrigeration systems are used to meet the needs for refrigeration requirements. Al Mers et al. described the mass and heat transfer in the cylindrical finned reactor used in solar adsorption refrigerator [117]. Their results showed that using finned reactor significantly increases the solar COP of the system about 45% of non-finned bed, i.e. from 0.07 to 0.105 under Morocco meteorological data. Louajari et al. studied the effect of a cylindrical adsorber on the solar adsorption refrigerating system performance using an activated carbon–ammonia pair [118]. It was found that the mass cycled adsorber equipped with external fins is more significant than the adsorber without fins. Moreover, maximal temperature in the adsorber with fins around 97 °C while in the adsorber without fins reaches only 77 °C (Table 2).

3.4. Economic feasibility

Economic studies showed that the solar systems have a high initial cost and low operational cost. It has been reported that the initial cost of the adsorption system is high compared to conventional system. On the other hand, solar adsorption systems are still promising alternative to the conventional systems, because they do not have running cost of fusel fuel, electricity transmission cost, or energy conversion cost, and system maintenance cost. The high cost of adsorption pairs' like zeolite- water and activated carbon-methanol prevented their market growth despite their technical success [103]. Increasing the geometrical construction, such as width pipe, increases the cost in other hand simplified or reduced construction will reduce the cost [140,141]. Chang et al. successfully reduced manufacturing cost while maintain high efficiency by using vacuum tank that contains evaporator, condenser, and adsorption bed [110].

Although system using silica gel–water has been commercialized, but the gel–water adsorption chiller has not been widely used due to mainly large size, low efficiency, and high cost of chiller.

Adsorber is the most significant and expensive component in an adsorption chiller. To overcome the these problems, Wang et al. optimized chiller with a modular design, where the sub adsorber is a compact fin-tube heat exchanger which expands heat transfer area to effectively enhance heat transfer performance [142]. The modular design show an increase in the COP. Furthermore, the cost and size of the optimized adsorption chiller are reduced which allow wider commercialization. Fig. 7 shows that the adsorption system has the highest projected and current costs mainly due to the low COP.

4. Conclusion

A comprehensive review about solar sorption system is presented. It has been concluded that the high initial cost of the system and low performance are the main reasons that prohibit significant commercial growth of this technology. Due to the awareness of an increase in energy demand for refrigeration and air conditioning, high oil price, environmental serious issues awareness, the research is still active to reduce the technical and economical drawbacks of solar sorption technologies. Solar energy is most appropriate option because solar energy level is in phase with cooling demand. Recent developments in absorption and adsorption solar cooling systems are summarized in this review paper to enhance their market growth.

Absorption solar cooling system has the higher COP compared to adsorption technology. It was found that the performance of the absorption cooling system depends mainly on: heat and mass transfer process, system configurations (i.e. number of stages, number of beds), type of solar collector, operational conditions, control strategy, and capacity of the thermal storage. Through the massive number of studies conducted to reduce high initial cost and low efficiency, it was found that the system efficiency and cost can be significantly improved if careful combinations between system configurations, collector type, and climate data are made. For example, single-effect chiller system with an evacuated tube collector outperforms multi-effect absorption chillers coupled with concentrating collectors when the percentage of direct normal irradiant is low. It is found that if high temperature heat source is available, higher effect absorber leads to high COP, however, their economic feasibility is questionable due to high cost of concentrator collectors.

It is found that the long adsorption/desorption time, the size of the system, high cost of adsorption chillers, and low performance of the adsorption chiller are the main factors that limits commercializing of solar adsorption systems. The reported COP values from several studies are very low (less than 0.5). New developments on the solar adsorption system from large number of studies are presented to eliminate these drawbacks. Enhancing heat and mass recovery and by developing multi stage, multi bed technologies, and optimization operating conditions are the main developments which successfully enhance system economic and energy performance.

Up to date, the cost of both solar absorption and adsorption system are not competitive. It is found that among solar cooling technologies, the absorption system has the highest market penetration among the solar cooling technology. The cost of solar collectors has major contribution towards the total cost of a solar-powered adsorption cooling system. It is found that utilizing thermal energy storage with double purpose systems enhance the feasibly of the system for space cooling.

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