A review of exergectic studies on solar collectors and solar cooling technologies for building applications

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Abstract: The development of air-conditioning systems that run on renewable energy sources will save electrical energy, which is mainly produced by the burning of fossil fuels. These systems will significantly reduce carbon emissions, and as a result environmental pollution and global warming effects of indoor climate control will be reduced. Among the various renewable energy sources, solar energy has proven to be the best option for air-conditioning because the maximum load on an air-conditioning system for cooling coincides with the period of greatest solar radiation input from the sun. Utilization of energy resources is related to sustainable development. To achieve sustainable development, increasing the energy efficiencies of processes utilizing sustainable energy resources plays an important role. Exergy, which is based on the second law of thermodynamics, is a useful tool to analyze energy systems, must be considered during their evaluation and modeling. Exergy analysis has been widely used in the modeling, simulation and performance evaluation of energy systems.This paper provides a comprehensive review of different solar refrigeration technologies with a focus on its exergecticstudies.Solar powered refrigeration basicallyconsists of two types, namely solar PV system, and solar thermal powered refrigeration system. The PVs work in combination with a conventional compressioncycle. Thermally powered refrigeration technologies are further classified into subcategories as follows: sorption cooling system (open systems or closed systems) and thermo-mechanical cooling system (ejector system). Solid and liquid desiccant cycles represent the open sorption system. Absorption and adsorption technologies represent the closed sorption system. These technologies can be designed and selected for its use according to the climatic conditions of the region, Coefficient of Performance(COP), its advantages and disadvantages, which have been discussed and considered in this study. The climatic condition of Saudi Arabia is also briefly presented, which will help in selection and proper utilization of these cooling technologies.

Keywords: Exergy, Solar Cooling Technologies, Solar Radiation, Climatic condition.

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1. Energy and exergy analyses

As stated by the first law of thermodynamics, the different forms of energy may individually undergo quantitative changes, but the overall amount of energy is conserved. The general energy balance can be expressed below as the total energy input rates equal to total energy output rates.

$$\sum \dot{E}_{in} = \sum \dot{E}_{out} \quad (1)$$

The energy balance equation, which quantifies the energy conservation law for a stationary process observed through a control volume, may be written in the rate form as follows:

$$\dot{Q} - P = \sum_{outlet} \dot{m}_k (h + gz + \frac{w^2}{2g})_k - \sum_{inlet} \dot{m}_k (h + gz + \frac{w^2}{2})_k \quad (2)$$

Exergy is defined as the work potential of a system relative to its reference state. The general exergy balance can be expressed in the rate form as follows:

$$\sum \dot{E}x_{in} - \sum \dot{E}x_{out} = \sum \dot{E}x_{destruction} \quad (3a)$$

or

$$\dot{E}x_{heat} - \dot{E}x_{work} + \dot{E}x_{mass,in} - \dot{E}x_{mass,out} = \dot{E}x_{destruction} \quad (3b)$$

with

$$\dot{E}x_{heat} = \sum \left(1 - \frac{T_0}{T_k}\right) \dot{Q}_k , \quad (3c)$$

$$\dot{E}x_{work} = \dot{W} \quad (3d)$$

$$\dot{E}x_{mass,in} = \sum \dot{m}_{in} \psi_{in} \quad (3e)$$

$$\dot{E}x_{mass,out} = \sum \dot{m}_{out} \psi_{out} \quad (3f)$$

where $\dot{Q}_k$ is the heat transfer rate through the boundary at temperature T_k at location k and $\dot{W}$ is the work rate. According to Dincer el al. [8], the total exergy rate of a system $\dot{E}x$ can be divided into four basic components, i.e. physical, kinetic, potential and chemical exergies.

$$\dot{E}x = \dot{E}x^{PH} + \dot{E}x^{CH} + \dot{E}x^{KN} + \dot{E}x^{PT} \quad (4)$$

Although exergy is extensive property, it is often convenient to work with it on a unit of mass or molar basis. The total specific exergy on a mass basis may be written as follows:

$$ex = ex^{PH} + ex^{KN} + ex^{PT} + ex^{CH} \quad (5a)$$

The flow (specific) exergy is calculated as follows:

$$\psi = (h - h_0) - T_0 (s - s_0) \quad (5b)$$

whereas h is the enthalpy, s is entropy, and the subscript zero indicates properties at the restricted dead state of P_0 and T_0.

The rate form of the entropy balance can be expressed as

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = 0 \quad (6)$$

where the rates of entropy transfer by heat transferred at a rate of $\dot{Q}_k$ and mass flowing at a rate of $\dot{m}$ $\dot{S}_{heat} = \frac{\dot{Q}_k}{T_k}$ are $\dot{S}_{mass} = \dot{m}s$ and, respectively.

Taking the positive direction of heat transfer to be to the system, the rate form of the general entropy relation can be rearranged to give the following equation:

$$\dot{S}_{gen} = \sum \dot{m}_{out} s_{out} - \sum \dot{m}_{in} s_{in} - \sum \frac{\dot{Q}_k}{T_k} \quad (7)$$

Also, it is usually more convenient to find $\dot{S}_{gen}$ first and then to evaluate the exergy destroyed or the irreversibility rate $\dot{I}$ directly from the following equation, which is called Gouy–Stodola relation [9]:

$$\dot{I} = \dot{E}x_{destruction} = T_0 \dot{S}_{gen.} \quad (8)$$

The specific exergy of an incompressible substance (i.e., water) is given by the following equation as reported by Szargut [9].

$$\psi_w = C(T - T_0 - T_0 \ln \frac{T}{T_0}) \quad (9)$$

The total flow exergy of air may be calculated from the following equation according to the Wepfer et al. [10]

$$\psi_{a,t} = (C_{p,a} + \omega C_{p,v}) T_0 [\left(\frac{T}{T_0}\right) - 1 - \ln \frac{T}{T_0}] + (1 + 1.608\omega) R_a T_0 \ln(\frac{P}{P_0}) + R_a T_0 \left\{ \frac{(1+1.608\omega)\ln(1+1.608\omega_0)}{(1+1.608\omega) + 1.608\omega \ln\left(\frac{\omega}{\omega_0}\right)} \right\} \quad (10)$$

where the specific humidity ratio is given by $\omega = \frac{\dot{m}_v}{\dot{m}_a}$.

Assuming air to be a perfect gas, the specific physical exergy of air is calculated by the following relation according to Kotas [11]

$$\psi_{a,per} = C_{p,a}(T - T_0 - T_0 \ln\left(\frac{T}{T_0}\right)) \quad + R_a T_0 \ln\frac{P}{P_0} \tag{11}$$

2.1.Energy and exergy efficiencies

The energy efficiency compares the amount of energy required by the final user (either electric, or mechanical, or thermal) to the total amount of energy exploited by the system, and can be expressed by the following relation,

$$Energy \quad efficiency(\eta) = \frac{Energy \quad Released \quad to \quad user}{Energy \quad provided \quad to \quad system} \tag{12}$$

The above definition is a common one, which may be applied to any form of energy, for example, in the case of a power plant for the production of electric energy, the energy efficiency will become as:

$$\eta = \frac{P}{Q_{thermal}} \tag{13}$$

where P is the output in the form of electrical power and $Q_{thermal}$ is the thermal energy as an input to the system.

Exergy efficiency is defined as the ratio of total exergy output to total exergy input, i.e.\

$$\varepsilon = \frac{\dot{E}x_{output}}{\dot{E}x_{input}} = 1 - \frac{\dot{E}x_{destruction}}{\dot{E}x_{input}} \tag{14}$$

where ''output'' stands for ''net output'' or ''benefit'', and ''input'' stands for ''used'' or ''fuel''. There is no standard set of definitions in the literature, two different approaches are generally used—one is called ''brute-force'', while the other is called ''functional.''[12].

2.2. Exergy efficiency of solar collectors

The exergy analysis of the solar collector is very important, as it will be used in all the solar cooling methods except PVs. The instantaneous exergy efficiency of the solar collector can be defined as the ratio of the increased water exergy to the exergy of the solar radiation [13]. In other words, it is a ratio of the useful exergy delivered to the exergy absorbed by the solar collector.

$$\varepsilon_{scol} = \frac{\dot{E}x_u}{\dot{E}x_{sol}} \tag{15}$$

where as

$$\dot{E}x_u = \dot{m}_{w,out}[(h_{w,out} - h_{w,in}) - T_0(s_{w,out} - s_{w,in})] \tag{16a}$$

or

$$\dot{E}x_u = \dot{m}_w C_w \left[(T_{w,out} - T_{w,in}) - T_0 (Ln\frac{T_{w,out}}{T_{w,in}}) \right] \tag{16b}$$

$$\dot{E}x_u = \dot{Q}_u \left[1 - \frac{T}{T_{w,out} - T_{w,in}} (Ln\frac{T_{w,out}}{T_{w,in}}) \right] \tag{16c}$$

$\dot{E}x_{sol}$ can be expressed in terms of solar Irradiance(G),

$$\dot{E}x_{sol} = AG_T \psi_{srad} \tag{17}$$

whereas exergy of solar thermal emission ψ_{srad} at temperature T is as follows, reported by [14,15].

$$\psi_{srad} = \left[1 - \frac{4}{3}\left(\frac{T_0}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_0}{T_{sun}}\right)^4 \right] \tag{18}$$

where T_{sun} was taken equal to 6000 K, by Petela [14]. Also the exergy of solar thermal emission ψ_{srad} is reported by Joshi et al [16] by the following equation:

$$\psi_{srad,\max} = (1 - \frac{T_0}{T_{sun}}) \tag{19}$$

The exergy of solar radiation with beam and diffuse components for parabolic collectors is also given by the following equation [17,18]

$$\psi_{srad,be,d} = I_{be}(1 - \frac{4T_0}{3T_s}) + I_d(1 - \frac{4T_0}{3T_s^*}) \tag{20}$$

whereas the temperature ratio in the Eq.(20) is given by the following relation:

$$\frac{T_0}{T_s} = 0.9562 + 0.2777 \ln(1/\kappa) + 0.0511\kappa \tag{21}$$

where κ is the dilution factor of diffuse radiation [19].

2.3. Energy and exergy efficiencies of PVs

Solar cell power conversion efficiency (η_{pc}) can be defined as a function of the area, actual current, actual voltage and solar irradiance as given by Joshi et al[16] as follows:

$$\eta_{pc} = \frac{V_m I_m}{GA} \qquad (22)$$

Where G represents the solar irradiance and A is the area of the solar cell. This efficiency is also called electrical efficiency. Another expression for electrical efficiency proposed by Zondag et al.[20] can be given as follows:

$$\psi_e = \eta_{cell}[1 - 0.0045(T_{cell} - 25\ ^oC)] \qquad (23)$$

The exergy efficiency of a system can in general be defined as given by equation (15) above. The thermal energy gained by the system during the operation is not desirable in case of PV system, this becomes a heat loss to the system.

$$\dot{E}x = V_m I_m - [1 - \frac{T_{amb}}{T_{cell}}]\dot{Q} \qquad (24)$$

$\dot{E}x_{sol}$ can be calculated by using Eq. (17). Using the Petela [14] approach,it becomes as follows:

$$\dot{E}x_{sol} = AG\left[1 - \frac{4}{3}\left(\frac{T_0}{T_{sun}}\right) + \frac{1}{3}\left(\frac{T_0}{T_{sun}}\right)^4\right] \qquad (25)$$

Using the approach of Joshi et al [16] for calculation of exergy of solar thermal emission ψ_{srad} as given by Eq. (19) , the $\dot{E}x_{sol}$ can be written as follows:

$$\dot{E}x_{sol} = (1 - \frac{T_0}{T_{sun}})GA \qquad (26)$$

And as a result expression for exergy efficiency for the PV system becomes as

$$\varepsilon_{PV} = \frac{V_m I_m - [1 - (T_0/T_{cell})]\dot{Q}}{[1 - (T_0/T_{sun})]GA} \qquad (27)$$

with

$$\dot{Q} = UA(T_{cell} - T_0) \qquad (28)$$

and

$$U = 5.7 + 3.8v \qquad (29)$$

The overall heat transfer coefficient, which includes both convection and radiation heat transfer coefficient from the PV cell to ambient [21]. The exergy rate of solar irradiance can also be calculated using the formula proposed by Petela [22] as given by equation (25) by the following equation. In this case the exergy efficiency of PV system will become as:

(22)

$$\varepsilon_{PV} = \frac{V_m I_m - (1 - (T_{amb}/T_{cell}))\dot{Q}}{[1 + (1/3)(T_{amb}/T_{sun})^4 - (4/3)(T_{amb}/T_{sun})]GA} \qquad (30)$$

2.4. Exergy efficiencies of PV/T systems

The exergy efficiency can be defined as;

$$\varepsilon = \frac{\dot{E}x}{\dot{E}x_{sol}} = \frac{\dot{E}x_e + \dot{E}x_{th}}{\dot{E}x_{sol}} \qquad (31)$$

By substituting the values of $\dot{E}x$ Eq. (24)and $\dot{E}x_{sol}$ from Eq. (26) in the above equation, the exergy efficiency of a PV/T system results in

$$\psi_{PV/T} = \frac{V_m I_m + [1 - (T_{amb}/T_{cell})]\dot{Q}}{(1 - \frac{T_{amb}}{T_{solarl}})GA} \qquad (32)$$

By substituting $\dot{E}x_{sol}$ from Eq. (25), the exergy efficiency of PV/T system becomes Joshi and Tiwari [23]

$$\psi_{PV} = \frac{V_m I_m + [1 - (T_{amb}/T_{cell})]\dot{Q}}{[1 + (1/3)(T_{amb}/T_{sun})^4 - (4/3)(T_{amb}/T_{sun})]GA} \qquad (33)$$

Joshi and Tiwari [23] has evaluated the exergy efficiency of PV/T air collector in terms of electrical and thermal exergies as follows:

$$\psi_{PV/T} = \frac{\eta_0[1 - \beta\Delta T]GA + \dot{Q}[1 - ((T_0 + 273)/(293 + \Delta T))]}{GA} \qquad (34)$$

where β = 0.00458, T_0 is the reference ambient temperature, η_0 is the electrical efficiency under standard test condition and (ΔT) is the temperature difference between ambient and the collector outlet temperatures.

2. Solar cooling refrigeration methods

Solar cooling technologies can be classified into two main categories: solar thermal and solar electrical. Then these cooling methods are subdivided, as shown in Fig. 1. The solar thermal have two major types of sorption system and thermo mechanical system. The solar electric cooling is achieved by two major systems, i.e., a compression cycle system and a thermoelectric system.

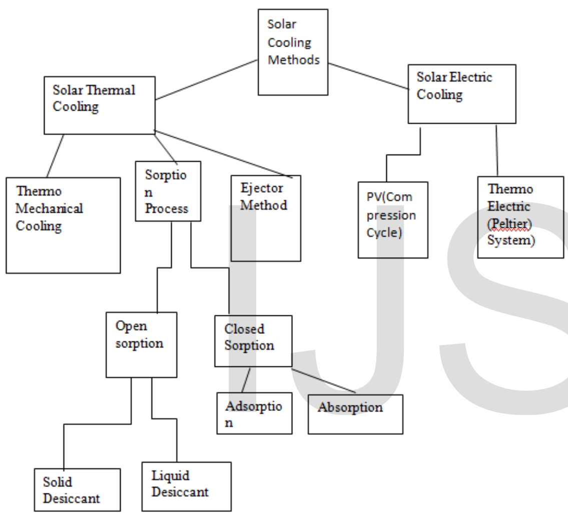

Fig. 1 Solar cooling methods.

3.1 Solar thermal cooling

A solar thermal refrigeration system consists of the following four major components: (i) solar collector, (ii) thermal storage, (iii) thermal air conditioning unit, and (iv) heat exchangers for the transfer of heat between these components[24]. First of all the collectors receives the solar radiation and the temperature of the fluid(water) inside the collector get increases via a heat convection process. The thermal storage tank is used for storing the energy in the form of heat. The thermal Air Conditioning unit is run by hot water coming from the thermal storage tank, and the refrigerant circulates through the thermal AC unit to supply cooling and heating. The heat exchanger is responsible for transferring heat between the hot and cold spaces [25]. The solar collector is the major part of these system. In past many study have been conducted to increase the efficiency of these systems. Currently the efficiency of these system are very low and required further improvement. There are different types of options available for the solar collector array, having a variety of complexity. These options can be categorized in the following way: flat plate solar collectors, evacuated tube collectors, and concentrating collectors. In order to choose between these, one must define the temperature range needed to run the thermal Air Conditioning system. In many cases this value is between 60 and 100 °C. Hence, the flat plate collectors, evacuated tube collectors, and concentrating collectors are all technically good options to consider. Energy and exergy analyses of flat plate collectors have been conducted by many researchers [26,27]. Jafarkazemi and Ahmadifard [26] performed the energy and exergy efficiency analysis of the collector, based on the analysis and comparison of results, the optimal working condition of the system was determined. According to the results, designing the system with inlet water temperature approximately 40° C more than the ambient temperature as well as a lower flow rate will enhance the overall performance of the system. Similarly, Ge et al. [27] presented an exergy analysis model of flat plate collectors, considering non-uniformity in temperature distribution along the absorber plate. The effects of ambient temperature, solar irradiance, fluid inlet temperature, and fluid mass flow rate on useful heat rate, useful exergy rate, and exergy loss rate were examined. It was reported that an optimal fluid inlet temperature existed for obtaining the maximum useful exergy rate. The calculated optimal fluid inlet temperature was 69 °C, and the maximum useful exergy rate obtained was 101.6 W. Exergy rate distribution was analyzed when ambient temperature, solar irradiance, fluid mass flow rate, and fluid inlet temperature were set to 20 °C, 800 W/m^2, 0.05 kg/s, and 50 °C, respectively. The exergy efficiency was 5.96%, and the largest exergy loss caused by the temperature difference between the absorber plate surface and the sun accounted for 72.86% of the total exergy rate.

Suzuki [28] conducted a theoretical comparison between the flat plate and evacuated tube collectors in terms of exergy, assuming a constant overall heat loss coefficient. It was concluded that two different collectors, an evacuated tubular collector and a flat-plate collector, had both nearly equal capabilities in exergy gain. The exergy balance equation on a solar collector was also given. The equation fully explained the exergy loss processes and could be used to derive the approximate optimum operating condition for solar collectors. Dutta Gupta and Saha [29] conducted an exergy analysis assuming a constant overall heat loss coefficient and fluid inlet temperature and calculated the optimal inlet temperature for a number of cases. Wing Han and Chiang Lee[30] compared four different types of solar collectors based on exergy efficiency assuming a constant overall heat loss coefficient. A criterion was proposed to rank the performance of these four types of solar collectors. There were two considerations involved, namely, the quantity and the quality of the annually collected energy. Rabl's method was used to predict this quantity. The average delivery temperature of hot water, assuming a reference insolation of 550 W/m^2, was also predicted. The exergy content of the delivered energy was then evaluated by multiplying the delivered energy by the Carnot efficiency. The annual collectible exergy was used to rank the performance of a collector.. Luminosu and Fara[31] conducted a research in order to demonstrate the dependence of exergy on fluid flowrate as well as collector area and the effect of these two parameters on the overall performance of the collector. Gunerhan and Hepbasli [32] performed an experimental analysis on closed loop solar water heater. The system performance was evaluated based on the experimental data of the Izmir province, Turkey, which was presented as an illustrative example. Exergy destructions as well as exergy efficiency relations were determined for each of the solar water heating system components and the whole system. Exergy efficiency values on a product/fuel basis were found to range between from 2.02 to 3.37%, and 3.27 to 4.39% at a dead (reference) state temperature of 32.77 ^{o}C, which was an average of ambient temperatures at eight test runs from 1.10 to 3.35 p.m., for the solar collector and entire system, respectively. Based on the results, the maximum exergy loss of the system occurred at the collector and the exergy efficiency of the aforementioned solar water heater system was about 4%. Gupta and Kaushik [33] examined the effect of length to width ratio of absorber plate as well as the optimal depth of air duct in order to determine the optimal performance parameters leading to higher output exergy during the collection of solar energy in a flat-plate solar air heater. The procedure to determine optimum aspect ratio (length to width ratio of the absorber plate) and optimum duct depth for maximum exergy delivery was also developed. The effects of the area of solar collector, aspect ratio and duct depth on the energy and exergy output rates of the solar air heater system were investigated. The exergy-based evaluation criterion indicated that the performance was not a monotonically increasing function of the area and aspect ratio, and a decreasing function of duct depth and inlet temperature of air. Based on the exergy output rate, it was found that there should be an optimum inlet temperature of air and a corresponding optimum area for any value of aspect ratio and duct depth. For values of area of solar collector lesser than optimal corresponding to inlet temperature of air equaled to ambient, higher exergy output rate was achieved for the low value of duct depth and high aspect ratio in the range of parameters investigated. Benli [34] conducted an experimental research on five different types of solar air heaters and compared their energy and exergy efficiencies. Based on their results, the heat transfer coefficient and pressure drop increased with shape of absorber surface. The results of the experiments were evaluated at the same time of the days with the same radiation. Alta et al. [35] compared three different types of flat plate solar air heaters(two having fins and the other without fins) while the energy and exergy output rates were evaluated in different mass flow rates, tilt angles and temperature conditions. The results indicated that the heater with fins and double glass cover was more effective than other types. In addition, it was found that the circulation time of air inside the heater played a role more important than of the number of transparent sheet.Akpinar and Kocyigit [36]performed an experimental analysis on a new flat plate solar air heater with different obstacles on absorber plates. As the results indicated, the collector's efficiency depended on solar radiation, surface geometry of the collectors and extension of the air flow line. Farahat et al. [37]conducted a research on optimization of flat plate collectors based on the exergy analysis in order to determine the optimal efficiency and design parameters of such systems. In their analysis, geometric and operating parameters were considered as variables. A simulation program was developed for the thermal and exergetic efficiency calculations. The concluding remarks obtained from their study were as follows: i) By Increasing the incident solar energy per unit area of the absorber plate, the exergy efficiency increased. ii) The exergy efficiency decreased rapidly when the ambient temperature and the wind speed increased, As these parameters changed during the day, for having the maximum exergy efficiency other parameters and the solar collector operating conditions should change during the day and the design of solar collector should be based on the daily average of these parameters. iii) Increasing the fluid inlet temperature increased the exergy efficiency, but there was a maximum point for the fluid inlet temperature where the exergy efficiency decreases quickly. iv) The design parameters such as pipes' diameter had a little effect on the exergy efficiency. Hazemi et al. [38] made a detailed energy and exergy analysis for evaluating the thermal and optical performance, exergy losses as well as exergetic efficiency for integrated solar storage collector under given operating conditions. Based on the results obtained, the integrated solar storage collector had energetic and exergetic efficiencies of 32% and 23.5%,respectively, while it provided acceptable stored thermal heat rate by supplying approximately 80% in domestic hot water requirements for a family composed of 5–6 persons. The exergetic efficiency of finned double-pass solar collectors was evaluated by Fudholi et al. [39]. The optimum energy efficiency was approximately 77% while the exergy efficiency was about 15-28%.Shanmugam et al. [40] constructed and tested a low-cost parabolic dish collector with commercial thermoelectric modules. The energy and exergy efficiencies were experimentally evaluated in the range 0.94-1.68% and 1.01-1.81%, respectively. Khasee et al.[41]performed energy and exergy analyses for a double-pass thermoelectric solar air collector. At a temperature difference of 22.8 ^{o}C, the unit achieved a power output of 2.13 W witha conversion efficiency of 6.17%. The exergy efficiency varied between 7.4% and 8.4%. Bayrak et al. [42] used energy and exergy analysis methods to study the performance of porous baffles inserted inside solar air heaters. The measured parameters were the inlet and outlet temperatures, the absorbing plate temperatures, the ambient temperature and the solar radiation. Five types of solar air heaters were tested and compared with each other in terms of their efficiencies. The obtained results showed that the highest collector efficiency and air temperature rise were achieved by solar air heaters with a thickness of 6 mm and an air mass flow rate of 0.025 kg/s whereas the lowest values were obtained for the solar air heater with non-baffle collectors and an air mass flow rate of 0.016 kg/s. Esen [43] reported an experimental study to evaluate the energetic and exergetic efficiencies of four types of double-flow solar air heaters with several obstacles and without obstacles under a wide range of operating conditions. The measured parameters were the inlet and outlet temperatures, the absorbing plate temperatures, the ambient temperature, and the solar radiation. The study concluded with an exergy relations delivered for different solar air heaters. The results illustrated that the largest irreversibility occurred at the flat plate (without obstacles) collector, in which collector efficiency had the smallest irreversibility. Bouadila et al. [44,45] conducted an experimental study to evaluate the thermal performance of a new solar air heater using a packed bed of spherical capsules with a latent heat storage system. The daily energy efficiency varied between 32% and 45% while the daily exergy efficiency ranged between 13% and 25%. Golneshan and Nemati [46]studied an unglazed transpired solar collector as a suitable device for preheating the outside air, using exergy analysis. This sort of collector was used mostly for preheating ventilation air as well as in heating air for crop drying. First, the exergy efficiency of unglazed transpired collector was derived. Based on this efficiency, optimization was performed on 200 different cases and finally, a useful correlation was proposed to predict the optimum working temperature. Bahrehm and Ameri [47] proposed energy and exergy models for single and two-glass cover solar air collector systems with natural convection flow. Assari et al. [48] performed energy and exergy analyses of an air-water combined solar collector. It was concluded that a dual purpose solar collector had better energy and exergy efficiency than a single collector. Also the triangle passage with a water inlet temperature of 60°C indicated better exergy efficiency than others. Said et al. [49] analyzed a flat-plate solar collector operating with single wall carbon nanotubes based nanofluid as an absorbing medium. The main result presented was that the single wall carbon nanotubes nanofluid reduced the entropy generation by 4.34% and enhanced the heat transfer coefficient by 15.33% compared to water as an absorbing fluid. Sarkar [50] analyzed the energetic and exergetic performance of flat-plate solar collectors using supercritical CO_2 as working fluid. Madadi et al.[51] investigated experimentally and theoretically the energy and exergy performance of a parabolic dish collector. Investigations on the hemisphere cavity receiver of a solar dishsystem were reported by Liu et al. [52], including exergy analysis. A review study of solar air heaters including their energetic and exergetic performance was also performed by Oztop et al. [53]. It was concluded that energy analysis method has been used by many researchers while exergy analysis method has been applied to the relatively low numbers of systems.

3.1.1 Thermo mechanical technology (ejector systems)

In the thermo-mechanical solar cooling, the water is heated to high temperature to produce pressurized steam, the steam passes through a turbine to produce work, as shown in Fig.2. The solar thermal power system in general can be considered as consisting of two subsystems, namely, the collector–receiver subsystem and heat engine subsystem. The collector–receiver circuit consists of a number of collector. The main components of the heat engine circuit are a heater, a turbine, a condenser, pump and a regenerator. Exergy analysis was conducted in two main subsystems along with their components, as given by Singh et al. [54], who calculated the irreversibilities of the solar thermal power plant.

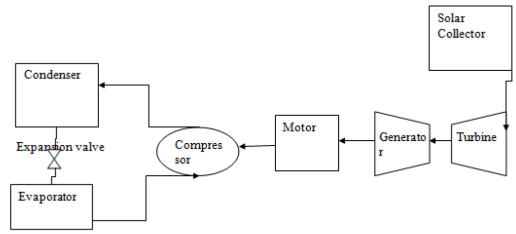

Fig. 2 Schematic drawing of the solar thermal cooling system with turbine and generator.

It was found that the collector and receiver had the highest irreversibilities and the efforts to reduce exergy loss should be made in this assembly. The exergy efficiency values for collector, receiver, collector receiver, boiler heat exchanger, heat engine and the overall system were 29.03%, 68.38%, 19.85%, 96.42%, 66.57% and 12.74% , respectively.

The overall exergy efficiency of the collector–receiver circuit was calculated as follows:

$$\varepsilon_{o,col-rec} = \frac{\dot{E}x_{u,col-rec}}{\dot{E}x_{par,s}} \quad (35)$$

with the useful exergy of the whole parabolic collector, as given by the following relation:

$$\dot{E}x_{u,col-rec} = n_{par,col} E\dot{x}_u \quad (36)$$

and the exergy received by the parabolic collector given by the following relation:

$$\dot{E}x_{par,s} = \dot{Q}_s\left(1 - \frac{T_0}{T_s}\right) \quad (37)$$

where $n_{par,col}$ is the number of the parabolic collectors, $\dot{Q}_s$ is the transferred solar heat and T_s is the solar temperature. The useful exergy of the one parabolic collector $E\dot{x}_u$ was found using Eqs. (16b and c) as follows:

The total efficiency of the heat engine subsystem consisting of the boiler heat exchanger and the heat engine cycle was calculated as follows:

$$\varepsilon_{h,eng} = \frac{\dot{W}_{net}}{\dot{E}x_{h,eng}} \quad (38)$$

with the available exergy of the working fluid of the heat engine cycle given as:

$$E\dot{x}_{h,eng} = \dot{Q}_u\left(1-\frac{T_0}{T_{Ran,0}}\right) \qquad (39)$$

where $\dot{Q}_u$ is the useful transferred heat $\dot{W}_{net}$ is the net work done by the heat engine cycle and $T_{Ran,0}$ is the ambient temperature of the Rankine cycle [54]. The overall exergy efficiency of the solar thermal power system is the product of the two separate efficiencies of the subsystems, as given below [54]:

$$\varepsilon_{h,eng} = \frac{\dot{W}_{net}}{\dot{E}x_{par,s}} \qquad (40)$$

Similarly a solar-driven ejector refrigeration system consists of a solar collector subsystem and an ejector refrigeration subsystem. Fig. 3 illustrates a schematic of this system[55].

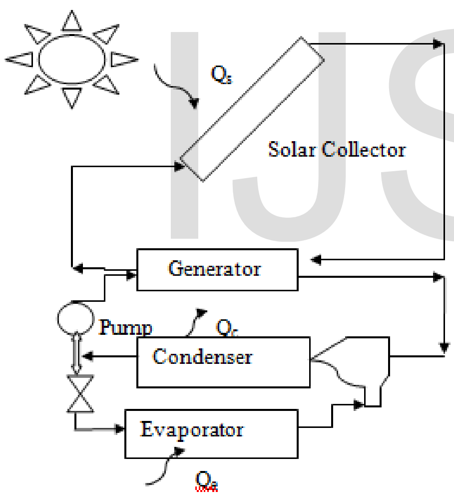

Fig. 3.Ejector Refrigeration Cycle drawn by Pridasawas and Lundqvist [55].

The exergetic efficiency of the ejector refrigeration cycle subsystem is defined as the ratio of the cooling exergy (at the evaporator) and the exergy inputs to the generator and the pump:

$$\varepsilon_{ej} = \frac{E\dot{x}_{evap}}{E\dot{x}_g + E\dot{x}_{p,e}} \qquad (41a)$$

with

$$E\dot{x}_g = \dot{Q}_g\left(1-\frac{T_0}{T_g}\right) \qquad (41b)$$

$\dot{Q}_g$

Whereas energy input to the generator and T_g is the generating temperature. The overall system exergetic efficiency can defined as follows;

$$\varepsilon_{sys} = \frac{E\dot{x}_{evap}}{E\dot{x}_{s,h} + E\dot{x}_{p,e}} \qquad (42a)$$

with

$$E\dot{x}_{s,h} = \dot{Q}_{available}\left(1-\frac{T_0}{T_{scol}}\right) \qquad (42b)$$

whereas $\dot{Q}_{available}$ is available heat for the process in the solar collector and T_{scol} is the average temperature of the solar collector.

De Vos [56] used the following relation for calculating the exergy input to the collector per unit area;

$$E\dot{x}_{scol} = f\sigma T_s^{\,4} + (1-f)\sigma T^4{}_{planet} - \sigma T^4{}_{scol} \qquad (43)$$

where $f$is the sunlight dilution factor, which equal to 2.16*10^{-5}on the earth, σ is Stefan–Boltzmann constant (5.67*10^{-8}W/m^{2}K^4), T_s is the temperature of the sun (K) and T_{planet} is the temperature of the planet (K). The exergy loss during the transformation from solar radiation to heat on the solar collector was reported as follows:

$$\dot{I}_{scol,trans} = E\dot{x}_s - E\dot{x}_{s,h} \qquad (44)$$

The exergy loss from the exergy input to the solar collector to the working fluid was calculated using the following equations:

$$\dot{I}_{scol} = E\dot{x}_{s,h} - E\dot{x}_u \qquad (45a)$$

with

$$E\dot{x}_u = \dot{Q}_u\left(1-\frac{T_s}{T_{scol}}\right) \qquad (45b)$$

Pridasawas and Lundqvist[55] evaluated the performance of an ejector refrigeration cycle driven by solar energy using exergy analysis method.The analysis conditions were as follows: A solar radiation of 700W/m^2, an evaporator temperature of 10 ^{o}C, acooling capacity of 5kW, butane as the refrigerant in the refrigeration cycle and ambient temperature of 30 ^{o}C as the reference temperature. Irreversibilities occurred among components and depended on the operating temperatures. The most significant losses in the system were in the solar collector and the ejector. The latter decreased inversely proportional to the evaporation temperature and dominated the total losses within the system. For the above operating conditions, the optimum generating temperature was about 80 °C. The exergy losses in the ejector refrigeration cycle is shown in Fig. 4 [55]. On the component basis of the ejector refrigeration subsystem, the largest loss (32%) occurred in the ejector followed by the generator (22%), the condenser(21%). The rest of the exergy loss occurred in the pump, the evaporator and the expansion valve. The exergetic efficiency values for the solar collector and refrigeration cycle were14.53% and 4.39%, respectively, while the overall exergetic efficiency was 0.66%.

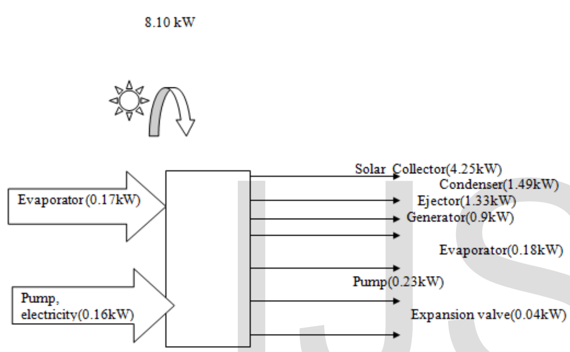

Fig. 4 Exergy balance of a solar-driven ejector refrigeration system for generating temperature of 90 °C and evaporating temperature of 10 °C[55].

Integration of the heat pipe with an ejector will result in a compact and high performance system, which does not require additional pump work. The basic cycle of the heat pipe/Ejector Refrigeration system (ERS) is illustrated in Fig. 5. The system consists of a heat pipe, ejector, evaporator and expansion valve. Unlike other vapor compression refrigeration system, which is powered by mains electricity, the heat pipe/Ejector Refrigeration system (ERS) does not require any electricity input. With the aim of finding the optimum operating conditions for a heat pipe/ERS, Ziapour and Abbasy [57]carried out an energy and exergy analysis. The results showed that COP could reach about 0.30 at generator temperature of 100 °C, condenser temperature of 30 °C and evaporator temperature of 10 °C.

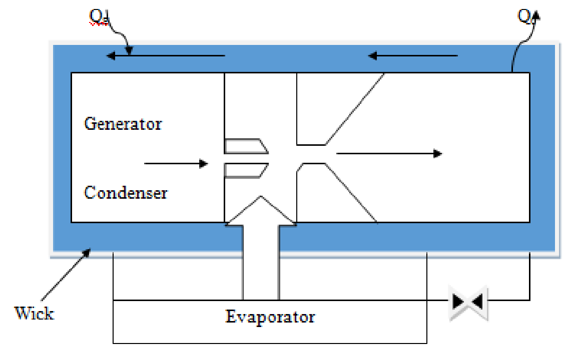

Fig. 5 Schematic diagram of heat pipe/ejector refrigeration system[57].

Wang et al.[58]proposed a combined power and refrigeration cycle as shown in Fig. 6, in which the Rankine cycle is combined with the Ejector Refrigeration system (ERS) by adding an extraction turbine between heat recovery vapor generator (HRVG) and ejector. The HRVG was a device, in which high pressure and temperature vapor was generated by absorbing heat from sources such as solar thermal, geothermal and waste heat. This cycle could produce both power output and refrigeration output at same time. By increasing the area of heat transfer and the coefficient of heat transfer in the HRVG, optimization design parameters in the ejector and turbine would improve the system efficiency.

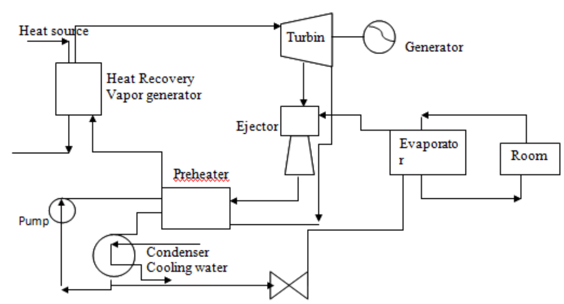

Fig. 6 Schematic diagram of combined power and ERS[58].

Also a combined power and ERS was presented by Zheng and Weng [59]. The cycle combined the Rankine cycle and the ejector refrigeration cycle. The ejector was driven by the exhausts from the turbine to produce power and refrigeration simultaneously. Simulation results illustrated that a thermal efficiency of 34.1%, an effective efficiency of 18.7% and an exergy efficiency of 56.8% could be obtained at a generator temperature of 122 °C, a condensing temperature of 25 °C and an evaporating temperature of 7 °C. It was also noted that while the generator temperature increased the fluid inlet pressure of the ejector increased. The results indicated that the proposed cycle had a great potential for producing refrigeration and most of the exergy losses occur in the ejector. Exergy analysis of combined power and ejector refrigeration cycle presented by Wang and Gao [60] showed that the largest exergy destruction occurred in the HRVG, followed by the ejector and turbine. In order to recover some of the thermal energy from the turbine exhaust, Khaliq et al. [61]combined a Libr-H_2O absorption system with power and ejector refrigeration system using R141b as refrigerant. Results of exergy distribution of waste heat in the cycle illustrated that approximately 53.6% of the total input exergy was destroyed due to irreversibilities in different components, only 22.7% was available as a useful exergy, and the remaining 23.7% was exhaust exergy lost to the environment. The results of the first and second law investigation showed that the proposed cogeneration cycle yielded better thermal and exergy efficiencies than the cycle without absorption system. However, no experimental results were available.

3.1.2 Sorption systems

In a sorption system, refrigeration effect is produced by using the physical or chemical attraction between a pair of substances. These 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 work as a refrigerant.

a) Closed sorption systems

These is the refrigeration system, which is powered by thermal carriers (hot water or steam), directly producing chilled water, which can be used in the air conditioning systems. These systems are already available in the markets long time ago. The common types are the absorption and adsorption systems.

i) Absorption system

An absorption system consists of two different chemical components, with one of them acting as the refrigerant and the second acting as the absorbent. The absorption refrigeration cycle can be considered as a modified version of the vapor refrigeration cycle, where instead of compressing a vapor between the evaporator and condenser, the refrigerant is absorbed into an absorbent and pumped to the generator where heat is added to the processes, causing the refrigerant to desorb from the solution as a vapor [62]. Fig. 7 shows the main components of an absorption refrigeration machine compared to the conventional machine.

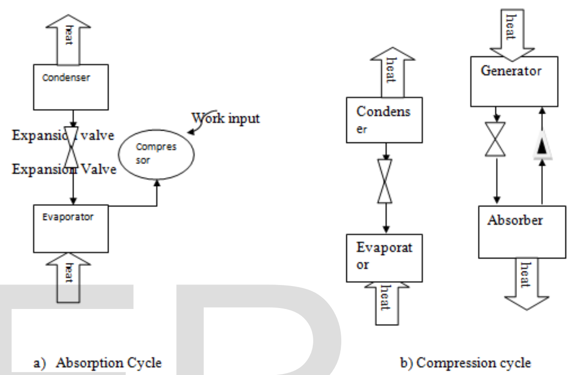

Fig.7. Main components of the absorption Cycles compared to compression chillers.

By combining a liquid solution refrigerant absorbent and a heat source can replace electromechanical compressor. Water/ lithium bromide(H_2O/LiBr) liquid solution is usually used for air conditioning applications (chilled water temperature above 5 °C),and for refrigeration application (chilled water temperature below 5 °C) an NH_3/H_2O mixture is applied.

The pair of working fluids plays a crucial role in absorption cooling system. The basic requirement of a pair of working fluids is that they have a bound of miscibility between them in the range of certain operating temperatures in a liquid phase. The coefficient of performance of an absorption system depends on both the physical and chemical properties of the system [63]. Gomri [64] investigated a hybrid Li/Br absorption cooling system utilizing heat from both an array of solar collectors and a natural gas burner. The temperature of the generator and the condenser varied between 45-83 °C and 28-36 °C, respectively, while the evaporation temperature was fixed at 5 °C. Under optimal operations, the COP was 0.82 while the exergy efficiency was 30 % at the minimization of the required collector surface. The natural gas consumption was significantly low, thus leading to very few CO_2 emissions (always less than 3 kg/h). Lopez-Villada et al. [65] evaluated and compared various solar absorption power-cooling configurations including evacuated tube collectors, parabolic trough collectors and linear Fresnel collectors (LFC) coupled with single-stage combined absorption and the Goswami cycle. Sozen et al. [66] performed the performance tests of the system under the climate condition of Ankara in Turkey. A parabolic collector was installed to obtain required temperatures. In the experiments, high temperature water obtained from the collector was used as heat source needed for the generator. The system design configuration was analyzed by using the experimental data. The effects of irreversibilities in thermal process on the system performance in the system were determined. The exergetic coefficients of performance of the system for cooling and heating were as follows, respectively:

$$COP_{ex,h}=\frac{Q_{cond}\left(1-\frac{T_0}{T_{cond}}\right)+\dot{Q}_{absor}\left(1-\frac{T_0}{T_{absor}}\right)}{\dot{Q}_g\left(1-\frac{T_0}{T_{absor}}\right)+\dot{W}_{pump,e}} \qquad (46)$$

$$COP_{ex,cool}=\frac{Q_{evap}\left(1-\frac{T_0}{T_{evap}}\right)}{\dot{Q}_g\left(1-\frac{T_0}{T_{absor}}\right)+\dot{W}_{pump,e}} \qquad (47)$$

Thermodynamic analysis showed that both losses and irreversibility had an effect on absorption system performance, while potential for the improvement in the components of the system was determined. The components of the evaporator and absorber of the system had a higher exergy loss than the other components. These components needed to be modified in order to improve the performance of the system thermally. Typical $COP_{ex,h}$ values of the system varied from 0.13 to 0.45. The highest and lowest $COP_{ex,h}$ values were obtained at higher evaporator temperature (10 °C) and the generator temperature of 75 °C, and at T_{evap}=10 °C and T_{cond}=28 °C, respectively. It was also concluded that the evaporator temperature could be as low as 3 °C to use the system for preservation of the food and air conditioning.

Some researchers [67-72] tried to give an exergy and a second law analysis of the single effect absorption heat pumps. Sencan et al.[67] evaluated the exergy loss, enthalpy, entropy, temperature, mass flow rate and heat rate in each component of the system considered in their analysis. It was concluded that the condenser and evaporator heat loads and exergy losses were smaller than those of the generator and absorber. This was because the heat of mixing in the solution, which was not present in pure fluids. The simulation program developed for the system in their analysis also indicated that that the cooling and heating coefficient of performance of the system increased slightly by increasing the heat source temperature. In contrast to that, the exergetic efficiency of the system decreased when increasing the heat source temperature for both cooling and heating applications. Aphornratana and Eames [68] performed a second law analysis of a single-effect absorption refrigeration cycle. The results obtained showed that the solution circulation ratio played an important role in determining the performance of the cycle. High solution circulation ratios resulted in high internal irreversibilities at the absorber and the generator. These irreversibilities could be reduced by increasing the solution heat exchanger effectiveness. The results also indicated that the irreversibility associated with heat transfer at the evaporator had a very great effect on cycle performance. It was also reported that a first priority should be given to the evaporator, while the absorber could be considered as the second one to improve the cycle performance. The second law analysis illustrated that it was more thermodynamically efficient if absorption systems would be operated using low-grade waste heat instead of high temperature heat sources.

Talbi and Agnew[69] developed a computer program to evaluate the performance of an absorption refrigeration system. The load in the condenser was higher than that in the evaporator because of primarily superheating of the inlet vapor to the condenser. The evaporator and condenser loads were approximately 27.8% less than the corresponding generator and absorber loads. It was proved that the absorption refrigeration cycle was effective in demonstrating the advantages of the exergy method since there were large irreversible losses in the heat transfer process that were not accounted for in the energy balance.

Lee and Sherif [70] performed an analysis of a lithium bromide/water (LiBr/H_2O) absorption system for cooling and heating on the basis of the first and second laws of thermodynamics. Simulation was used to find the coefficient of performance and the exergetic efficiency of the absorption system for different operating conditions such as the heat source, cooling water, chilled water, and supply hot water temperatures. They concluded that increasing the heat source temperature could increase the cooling coefficient of performance of the system, but as the heat source temperature increased beyond a certain threshold, the coefficient of performance of the system levels off and even decreased in some cases. It was also concluded from the parametric analysis that increasing the heat source temperature would increase both the heating COP and the exergetic efficiency.

Kilic and Kaynakli[71] also performed the second law analysis of a single-stage water-lithium bromide absorption refrigeration system (*ARS*). A mathematical model was introduced, based on the exergy method to assist the performance of the system, exergy loss of each component and total exergy loss of the system as a whole. It was concluded that the performance of the *ARS* could be increased by increasing generator and evaporator temperatures, but decreased with increasing condenser and absorber temperatures. Exergy losses in the expansion valves, pump and heat exchangers were small compared to other components. The highest exergy loss occurred in the generator irrespective of operating conditions, which made the generator the most important component of the cycle. Kaynakli and Yamankaradeniz [72] concluded that better coefficient of performance value was obtained at high generator and evaporator temperatures, and also at low condenser and absorber temperatures. By increasing generator temperature, total entropy generation of the system decreased. Whereas maximum entropy generation occurred in the generator, while entropy generations in the refrigerant heat exchanger, expansion valve and solution pump were very slight.

Gebreslassie et al. [73] performed an exergy analysis for a lithium-bromide/water pair in a multiple- effect absorption system. They conducted a comparative exergy study among single-effect, double-effect and triple-effect absorption for unavoidable destruction. The coefficient of performance, the exergy destruction rates, and exergetic efficiencies were determined and the effect of the heat source temperature was also evaluated in the analysis. It was concluded that the coefficient of performance increased significantly from double lift to triple effect cycles. By changing the configuration the exergy efficiency had small effect. In all configuration cycles, the effect of the heat source temperature on the exergy destruction rates was same for the similar type of components, while the quantitative contributions depended on cycle type and flow configuration. Largest exergy destructions were due to the absorbers and generators, specifically at higher heat source temperatures.

Izquierdoetal. [74] presented an exergetic analysis for a solar plant, consisting of flat plate collectors feeding the generators of the double-stage LiBr–H_2O air-cooled absorption cycle. The conclusions obtained illustrated that the irreversibilities generated by the double stage thermal compressor would tend to increase with the absorption temperature up to the value of 45 ^oC. It was also reported that the double stage system had about 22% less exergetic efficiency than the single effect one and 32% less exergetic efficiency than the double effect one. The entropy generated and the exergy destroyed by the air cooled system were higher than those by the water cooled one. For an absorption temperature of 50 ^oC, the air cooled mode generated 14 % more entropy and destroyed 14% more exergy than the water cooled one.

Very recently, several novel studies have been made using energy and exergy analyses [75-77] of absorption systems. Gunhan et al.[75] proposed a novel solar assisted absorption cooling (SAAC) system and performed its exergetic analysis and assessment. The performance of the system was evaluated through exergy analysis. Water was used as refrigerant in the system. Also a parametric study was undertaken to investigate the effect of the various dead (reference) state temperatures ranging from 30 °C to 42 °C with an interval of 4 °C on the whole system efficiency. In this context, the exergy efficiency of the whole system varied between 13.1 and 43.2% corresponding to the dead state temperature range. The highest relative irreversibility occurred in the solar collector and the absorption chiller were 66.1% and 34.6% respectively, in charging and discharging processes at a dead state temperature of 34 °C. Aman et al.[76] developed a thermodynamic model for a 10 kW air cooled ammonia–water absorption chiller driven by solar thermal energy. Both energy and exergy analyses were performed to assess the performance of this residential scale cooling system. It was concluded that most exergy loss occurred at absorber, followed by generator and condenser with values of 63%, 13% and 11%,respectively. In addition, it was also reported that the exergy loss of the condenser and absorber greatly increased with temperature, while the generator was not effected that much, and the exergy loss in the evaporator was the least sensitive to increasing temperature. Wonchala et al. [77] proposed a detailed solution procedure and validated for water–lithium bromide absorption cooling machine It was concluded that the circulation ratio and the coefficient of performance increased as the temperature of the heat source increased, while the second law performance was reduced. Padilla et al. [78] conducted a parametric analysis of a combined power/cooling cycle, which combined the Rankine and absorption refrigeration cycles, uses NH_3/H_2O mixture as the working fluid and produced power and cooling at a time. The cycle heat source temperature had minimum and maximum values of 90 and 170 °C, respectively. The maximum effective first law and exergy efficiencies for an absorber temperature of 30 °C were found as 20% and 72%, respectively.

As for as the study of advanced absorption cooling systems are concerned, Kim and Ferreira[79] investigated a low temperature-driven absorption half-effect cycle for the development of an air-cooled H_2O/LiBr absorption chiller to be combined with low-cost flat solar collectors for solar air conditioning in hot and dry regions. Simulation results showed that the chillers would deliver chilled water around 7.0 °C with a COP of 0.37. Morosuk et al. [80] proposed that most of the work on absorption cooling used first and/or second law approaches. By splitting the exergy destruction in every component of the machine as endogenous/exogenous and unavoidable/avoidable form, a unique tool for an accurate identification, where efficiency gains are more likely to be achieved, was proposed. It was concluded that most of the cycle irreversibility was due to desorber and absorber; however2/3 of the total exergy losses were in fact unavoidable.

Izquierdo et al. [81] studied on the entropy generated, the exergy destroyed and the exergetic efficiency of lithium-bromide absorption thermal compressors of single and double effect, driven by the heat supplied by a field of solar thermal collectors. Chua et al. [82] performed exergy analysis for continuous operation of aqua ammonia absorption refrigeration systems to estimate the losses within the different components of an absorption system. Their results indicated that 30% of the thermal energy input was lost within the rectifier and the regenerative heat exchange; while the liquid and vapor phases in the absorber contributed to about 6% and 3% thermal energy dissipation respectively. Lostec et al. [83] performed the second law optimization of an absorption refrigeration system. Their results indicated the presence of three optimum values of COP (0.56, 0.62 and 0.69) for the absorption system minimized UA value, irreversibility and exergetic efficiency, respectively. Their results also showed that all of these three optimum COP values were lower than the maximum COP value. Also their results indicated that 33% and 34% of the exergy destruction took place inside the absorber and desorber of the absorption system, respectively.

Caciula et al. [84] carried out a simulation study of the aqua-ammonia absorption system based on the first and second laws of thermodynamic. Compound parabolic collectors were utilized in their analysis and developed a mathematical model for aqua-ammonia mixture based on mass and energy conservation equations. It was concluded that maximum COP values of 0.66 and 0.73 could be achieved at generator temperatures of 90 °C and 74 °C when operated at evaporator temperatures of -3 °C and 6 °C, respectively. Kilic and Kaynakli [85] performed the first and second laws of thermodynamic analysis of a LiBr– water absorption chiller. Their results indicated that irrespective of the working conditions, the generator presented the highest exergy loss component, i.e. 45.6% of the overall exergy loss in the chiller while the pump and refrigerant heat exchanger represented the lowest exergy loss component in the chiller. Ziapour and Tavakoli [86] developed a computer code for the simulation of a diffusion absorption system using Engineering Equation Solver software. Both first and second law analyses were performed on diffusion absorption system. It was illustrated that best system performance was obtained for the rich and poor solution concentrations of 0.35 and 0.1, respectively for a generator temperature of 100 ^{o}C. Exergy losses in the generator and absorber were found to be very high in the system compared to other components. It was also concluded that the improvement in such systems was possible by the optimization of the generator and the absorber. Recently some experimental works have also been conducted over diffusion absorption systems. Yildiz and Ersoz [87] performed both the simulation and experimentation for aqua ammonia based diffusion absorption system using helium as an inert gas.

ii) Adsorption

The adsorption cooling systems uses the properties of certain solids of adsorbing large quantities of vapor due to their very large superficial area and porosity. The difference between the absorption and adsorption chillers is the chemical process that takes place in the process, and is dependent on the pair of working fluids that are selected. Generally, adsorption chillers use water-silica, zeolite-water, activated carbon-methanol or activated carbon-ammonia, while absorption chillers uses a wider range or working fluids, with lithium bromide-water, lithium chloride-water and water-ammonia being some of the most commonly used as reported by [86,88]. The main components of an adsorption cooling systems are two adsorbent beds, a condenser an expansion valve and an evaporator where the refrigeration effect is carried out, as shown in Fig. 8. The refrigerant vapor is adsorbed over the surface of the solid material mainly in the internal surface.

Operation starts with a refrigerant saturated bed that is heated by solar collectors to free the refrigerant vapor that enters the condenser where the latent heat is removed and it is liquefied.

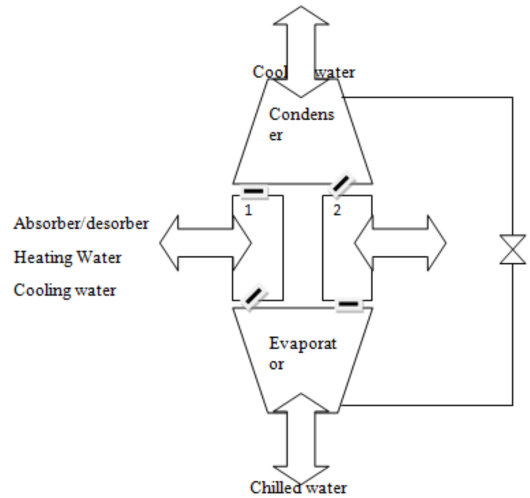

Fig.8 Block diagram for an adsorption refrigeration system.

Then the refrigerant is throttled in the expansion valve and enters the evaporator and evaporates producing the cooling effect. The refrigerant vapor then enters the second adsorber bed where it is adsorbed. The process continues until all the refrigerant being generated and as well as the adsorbent bed is saturated. Finally the heating and cooling lines are switched and heat is added to the second bed and cooling water flows to the first bed producing a semi continuous cycle. Typical operating conditions with a temperature of hot water of about 80 °C can achieve a COP of about 0.6.A simple adsorption cycle indicating all the heat transfer for a complete cycle is shown in Fig. 9 as was reported in the study of Anyanwu and Ogueke [89].

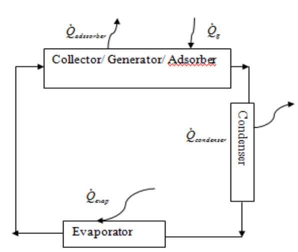

Fig. 9. Schematic of a simple adsorption

refrigeration.

The thermodynamic efficiency (exergetic efficiency) relates to the actual performance of a system to its expected maximum performance as determined from its ideal Carnot cycle. It may be written from a relation proposed by Pons et al. [90] as follows:

$$\varepsilon = \frac{COP_a}{COP_{\max}} \tag{48}$$

with

$$COP_{\max} = \frac{1 - \left(T_{adsor} / T_g \right)}{\left(T_{adsor} / T_{evap} \right) - 1} \tag{49}$$

and

$$COP_a = \frac{\dot{Q}_{evap}}{\dot{Q}_g} \tag{50}$$

where $\dot{Q}_{evap}$ and $\dot{Q}_g$ are the heat transferred during refrigeration and the heat used to generate refrigerant during generation, respectively [89].The COP and thermodynamic efficiency depended on the mass of adsorbate generated and the generation temperature, T_g. Above a certain generator temperature, the mass of adsorbate generated was no longer a direct function of the temperature. Thus, an improved COP and consequently the thermodynamic efficiency would only come from an adsorbent that would readily desorb the adsorbate in its pores up to a low level of about 20–25 g of adsorbate per kg of adsorbent for the range of obtainable temperature using a conventional flat plate solar collector. This, however, was with the understanding that the refrigerant would have a high latent heat of vaporization. The activated carbon/ammonia for 0, -10 and -20 ^{o}C evaporator temperatures and 47 ^{o}C condenser temperature gave thermodynamic efficiency values above 100% at generator temperatures of about 70 ^{o}C.

b) Open sorption systems

A desiccant based solar cooling system directly conditions incoming ventilation air. The process consists of three stages: the first stage includes dehumidifying the incoming air using either a solid or liquid desiccant (which absorbs moisture); the second stage involves recovering the sensible heat from the outgoing exhaust air; and the third stage involves using an evaporative cooling process to cool the air to the desired temperature. The desiccant is subsequently regenerated using solar thermal energy (thereby removing water from the desiccant). Fig. 10 describes the solid desiccant cooling system with solar collector according to the Kim and Ferreira [5].

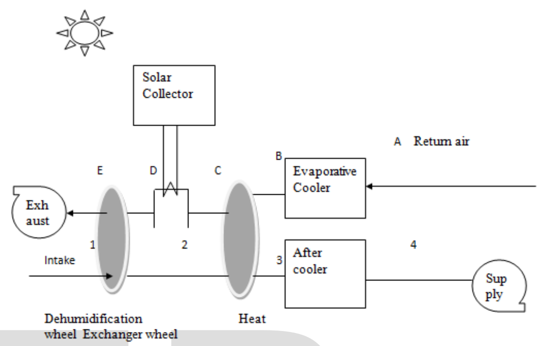

Fig. 10 A solid desiccant cooling system with solar collector[5].

The system includes two revolving wheels (i.e., dehumidification wheel and the heat exchange wheel) with the other components within an air handling unit (AHU).Desiccant systems are most effective in temperate climates as the dehumidification of the air must be effective enough to allow for evaporative cooling to take place. Desiccant based systems can also be effective when providing only dehumidification to the incoming ventilation air before the air enters the traditional air conditioning system, which would subsequently cool the air to the desired temperature [91]. Kanoglu et al. [92] presented a method for energy and exergy analyses of desiccant cooling systems. The procedure was applied to an experimental device functioning in a ventilation mode with natural zeolite as the sorbent material. They obtained a COP of 0.35 and an exergy efficiency of 11.1%. Khalid et al.[93]performed an exergy analysis on a liquid-desiccant-based, hybrid air-conditioning system that uses lithium bromide. The optimum desiccant mass-flow rate through the regenerator was found to be about 30kg/hm² for anambient temperature of40 °C. The maximum irreversibilities were generated at an ambient vapor pressure of 25mm Hg and a desiccant mass flow rate of 5kgh $^{-1}$m $^{-2}$.

3.2. Solar electric cooling systems (vapor compression systems, Peltier method)

While the output of a PV cell is typically direct current (DC) electricity, most domestic and industrial electrical appliances use alternating current(AC).Therefore, a complete PV cooling system typically consists of four basic components, as shown in Fig 11, photovoltaic modules, a battery, an inverter circuit and a vapor compression AC unit [94].

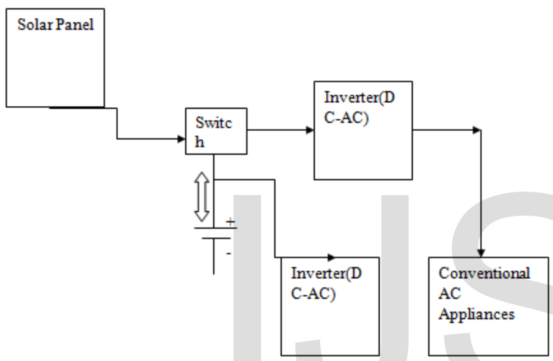

Fig.11 Schematic of a stand-alon PV system.

Abid and Hepbasli [95,96] performed two studies on a photovoltaic system for the Riyadh region of Saudi Arabia. The dynamic exergetic performance of photovoltaic modules through energy and exergy efficiencies, sustainability index, improvement potential factors, and in terms of their exergy destruction costs was evaluated. The thermodynamic variables, such as environmental temperature, module temperature, wind velocity, density and humidity of air, were considered in the analysis. It was reported that the energy efficiency varied between 34.09% and 52.14% while the exergy efficiency ranged from 0.16% to 15.23%, respectively. The exergy efficiency for the PV/T analysis changed between 1.36% and 19.86%, with an average value of 9.65%. The sustainability index was between 1 and 1.17 for the system considered in the analysis. The improvement potential factor had minimum and maximum values of 3376 W and 24559.42 W, respectively while the exergy rate of solar radiation was in the range of 3850.9 W and 25115.95 W. Ooshaksaraei et al. [97] used the first and second laws of thermodynamics to examine the steady-state and transient performance of a single-path air-based bifacial PV/T panel. They reported that the best thermal efficiency (51%-67%) was provided by the bifacial photovoltaic thermal panel with two parallel air streams. The exergy analysis indicated that a single air stream panel had the highest exergy efficiency (4.43%-10.15%). The single-path panel design was the best option in case the user was mainly interested in electrical energy. The double-path parallel design was the best option if thermal energy was the dominant desired output energy. Marletta and Evola [98] performed the second law analysis of a water-cooled PV/T module. It was correctly observed that the main drawback of this technology was that in the case of PV cells the electricity production potential increased at low temperatures, whereas the thermal energy usability was higher at high temperatures. The authors stress out the importance of the definition of the operational conditions that maximized the total exergy harvested by the system. Dupeyrat et al. [99] reported experimental results regarding the energetic and exergetic performance of a flat plate PV/T collector. Buker et al. [100] studied the energy and exergy efficiencies of a polyethylene heat exchanger loop underneath PV modules. The system was designed to act as a roof element that properly blended into the surroundings and became a heat resource for solar assisted heating and cooling technologies. Al-Shamani et al.[101] presented an overview of the research, performance, and development of PV/T collector systems. Covered and uncovered water PV/T collector types were described and analytical and numerical models for their and simulation as well as experimental works focusing on their operation were discussed. Exergy analysis showed that uncovered PV/T collectors produced the largest total (electrical plus thermal) exergy. Zhang et al. [102] assessed the dynamic performance of a novel solar photovoltaic/loop-heat-pipe (PV/LHP) heat pump system for potential use in space heating or hot water generation. They performed theoretical simulations for estimating the energy and exergy efficiencies and experimental verification. Several performance indicators were used such as the basic thermal performance coefficient (COP_{th}) and the advanced performance coefficient (COPPV/T). The mean daily electrical, thermal and overall energetic and exergetic efficiencies of the PV/LHP module were 9.13%, 39.25%, 48.37% and 15.02%, respectively, and the average values of COP_{th} and COPPV/T were 5.51 and 8.71. Gholampour and Ameri [103] studied from the energetic and exergetic viewpoint the effect of some parameters, such as the packing factor, the fin number and the fin height as well as environmental and dimensional parameters on the performance of the PV/T system with natural air flow. Sobhnamayan et al.[104] analyzed the energy and exergy efficiency of a PV/T water collector. Kamthania and Tiwari [105] presented an overview on the development and application aspects of the PV/T collector systems for the last 30 yr. The review covered a detailed description and thermal models of hybrid photovoltaic thermal systems, using air as the working fluid. Numerical model analysis and qualitative evaluation of thermal and electrical output in terms of the overall thermal energy and exergy for various silicon and non-silicon based modules were carried out. Wu et al. [106] presented a review of methodologies for calculating heat and exergy losses of conventional PV/T systems. Kasaeian et al. [107] estimated the overall performance of air PV/T collectors based on energy and exergy analysis. Simulations were performed under the climate conditions of Kerman (Iran). Both unglazed and glazed air PV/T, were considered. The overall energy efficiency of glazed/unglazed systems was about 66%/52%. The overall exergy efficiency for unglazed/glazed systems ranged between 11.2-11.6%/ 10.5-11.1%.

Another method for solar electric cooling is Peltier effect, which can be defined as presence of heating or cooling at junction of two different conductors due to electricity flow [108]. When a DC current is passed through one (or more pairs) of n and p-type semiconductor materials, the temperature of one conductor decreases and absorb the heat from its surrounding space. The absorbed heat from the space occurs when electrons pass from a p-type material to the n-type material (from low energy level to high energy level) . When a temperature gradient is achieved between the hot and cold ends of the conductor, adverse voltage is created [109]. The heat is transferred through n and p-type semiconductor from cold side to the hot side then the heat is rejected to outside. If the direction of the current is reversed, the direction of the heat flow is reversed also, and air conditioning system operates in the heating mode [110].

3. Discussion

There are certain advantages and disadvantages of these different solar cooling methods. For example, the Rankine Cycles are suitable for high capacity systems, for integration into polygeneration systems (heat, electricity and refrigeration) and its limitations are high installation cost, large system, regular maintenance required due to complications and many moving parts. Similarly, ejector refrigeration systems can utilize low temperature heat source, low operating and installation cost and the limitations are its low COP, complicated design of the ejector, and difficult to operate in a wide range of ambient temperatures. Furthermore, the advantages of absorption systems are: having only one moving part (pump) with the possibly of no moving parts for small systems (e.g. Platen-Munters cycle), possible to utilize low-temperature heat supply and its limitations are: It cannot achieve a very low evaporating temperature in using LiBr-H_2O as working media, the system is quite complicated and difficult for service. Adsorption systems have the advantage of no moving parts (except valves), low operating temperatures can be achieved and the limitations are its high weight and poor thermal conductivity of the adsorbent make it unsuitable to use for high capacities and can cause long-term problems, low operating pressure requirement makes it difficult to achieve air-tight, very sensitive to low temperature, especially the decreasing temperature during the night, and it is an intermittent system. The desiccant cooling systems are environmentally friendly, water is used as the working fluid and can be integrated with a ventilation and heating system. The driving temperature is quite low, thus possible to use low temperature solar collector. In case of the liquid desiccant, the system does not need a condenser because the refrigerant can be released to the atmosphere. The limitations are: it is difficult to have good control of the system in a humid area. It is not appropriate for areas where water is scarce.

In case of the solid desiccant, the system requires maintenance due to moving parts in the rotor wheel of the solid desiccant system. In case of the liquid desiccant, the liquid sorbent can contaminate the supplied air. Crystallization generally occurs in liquid desiccant systems due to poor process control. The advantages of PV systems are high COP, long term experience and widely available commercially, scalable from small to a large systems and disadvantages for a PV system are: installation cost is high and requires battery for energy backup., Can be noisy.

It is also important to consider local conditions when designing and installing a solar driven refrigeration system. This is predominantly due to the different characteristics of each refrigeration cycle. Furthermore, cooling load characteristics for each location or building are not always similar. There are certain limitations for each solar cooling technique imposed by the local climate. Some of them are related to insolation, or rather lack of insolation, whereas others are related to temperature levels or humidity ratios. An example of the latter is the desiccant cooling system, which functions inefficiently at high humidity. Although low humidity is preferred for the desiccant cooling system, sufficient make up water is needed as the working medium for the evaporative cooling process. Also the temperature of the heat sink significantly affects the performance of heat driven systems in general. A lower heat sink temperature can minimize the required driving temperature and improve system COP. It is therefore critical to find a good heat sink in order to achieve a reasonably low temperature, especially in a hot environment. For ejector refrigeration cycles, systems with one fixed ejector dimension can only operate within a small range of condenser temperatures. Subsequently, whenever discussing the potential for solar cooling, a thorough investigation of local conditions must be carried out. Furthermore, in areas where cooling demand is high, the system works more efficiently and requires less solar collector area per cooling effect. The ambient temperature (heat sink) affects the condenser conditions, and the condenser temperature strongly influences the performance of the absorption chiller. In an area where the ambient temperature is not extremely high, the solar-driven cooling systems work more efficiently and dry condensers (air-liquid condenser) may be used. In very hot climates, liquid-liquid condensers or cooling towers are needed. It was found that Saudi Arabia experiences, on an average, more than 2.0MWh/m^2 of global solar radiation each year on horizontal surface as reported by Sahin et al.[111]. The long-term mean values of sunshine duration and global solar radiation on horizontal surfaces are summarized in Table 1.[112]. The conclusions drawn from the table were that higher values of global solar radiation were observed in Nejran, Bisha, Al-Sulayyil, etc. area in the southern, most part of the Kingdom and relatively lower values in the northern region like Hail, Sakaka, Tabarjal, etc.

Table I. Long-term daily mean values of sunshine duration and global solar radiation on horizontal surface for Saudi Arabia[112].

Station #	City	Lat (deg.)	Lon (deg.)	Alt (m)	S (h)	H (MWhm^{-2} yr)
1	Qurayyat	31.33	37.35	2	9.0	2.03
2	Tabarjal	30.52	38.38	3	9.0	1.72
3	Sakaka	29.97	40.20	574	9.0	1.94
4	Tabuk	28.38	36.58	773	9.1	1.64
5	Tayma	27.63	38.48	820	9.2	2.04
6	Hail	27.47	41.63	1010	9.4	1.91
7	Sarrar	26.98	48.38	75	8.7	1.66
8	Al-Ula	26.62	37.85	681	9.1	2.12
9	Qatif	26.55	50.00	8	8.4	1.73
10	Maqqala	26.37	47.37	450	8.9	1.78
11	Zilfi	26.30	44.80	605	8.9	2.04
12	Unayzah	26.07	43.98	724	9.3	2.00
13	Uqtalas-Suqur	25.83	42.18	740	9.1	2.23
14	Hutatsudair	25.53	45.62	665	9.0	2.15
15	Al-Hofuf	25.50	49.57	160	8.7	2.07
16	Shaqra	25.25	45.25	730	9.2	2.21
17	Hanakiya	24.85	40.50	840	9.1	2.21
18	Riyadh	24.57	46.72	564	9.2	1.87
19	Madina	24.52	39.58	590	2.32	2.32
20	Dawdami	24.48	44.37	0	8.8	2.17
21	Derab	24.42	46.57	0	8.7	2.26
22	Al-Kharj	24.17	47.40	430	9.1	2.03
23	Harad	24.07	49.02	300	9.0	1.71
24	Yabrin	23.32	48.95	200	9.1	2.06
25	Al-Aflat	22.28	46.73	539	9.0	2.19
26	Khulays	22.13	39.43	60	8.9	2.18
27	Sayl Kabir	21.62	40.42	1230	8.9	2.46
28	Turbah	21.40	40.45	1130	9.0	2.09
29	Taif	21.23	40.35	1530	8.9	1.98
30	Sulayyil	20.47	45.57	600	9.0	2.40
31	Bisha	20.02	42.60	1020	9.2	2.56
32	Heifa	19.87	42.53	1090	9.1	2.22
33	Juarshy	19.85	41.57	2040	18.5	1.98
34	Madaylif	19.53	41.05	53	8.5	2.32
35	Al-Numas	19.10	42.15	2600	7.4	2.21
36	Kwash	19.00	41.88	350	8.5	1.70
37	Kiyad	18.73	41.40	30	8.4	1.87
38	Sirr-Lasan	18.25	42.60	2100	8.7	1.84
39	Abha	18.22	42.48	2200	8.7	2.13
40	Najran	17.55	44.23	1250	9.1	2.53
41	Sabya	17.17	42.62	40	8.5	1.83

The lower values are much higher than those in other western and European countries and hence offer an opportunity to harness the power of the sun to generate electricity. As far as studies on calculating the solar exergy of Saudi Arabian regions are concerned, only one study conducted by Hepbasli and Alsuhaibani [113] has appeared. In this context, they comprehensively reviewed various solar exergy models used in solar energy-related applications, and determined the solar exergetic values for some regions of Saudi Arabia and Turkey, which were taken as two illustrative examples, to which various models were applied and compared. The ratios of solar radiation exergy to solar radiation energy (exergy-to-energy ratio) for northeastern Saudi Arabia were obtained to be on average 0.933 for both approaches of Petela [114,115] and Spanner [116]and 0.950 for Jefer's approach[117]at outside air temperatures between 16.18 and 33.01 ◦C. The data used in the analysis related to Saudi Arabia were taken from Sahin et al. [111], who reported the experimental values obtained from the measurements on the shoreline of the Arabian Gulf in northeastern Saudi Arabia, near Dhahran. The eastern and the western parts of Saudi Arabia also experiences higher intensities of global solar radiations and hence should be explored. Fig. 12 illustrates a comparison of various solar refrigeration technologies in terms of performance and initial cost, although differing in technical maturity and commercial status.

Low temperature sorption system would be a better option for Saudi Arabia because of the cost of a solar collector system tends to increase with working temperature more rapidly than COP of a sorption machine does. 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 Research &Development 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 [5].

It is also reported that the COP is higher for a LiBr–H_2O double-effect than for a single-effect absorption chiller, but it requires thermal energy at temperatures of 140–160 ◦C [6]. In this temperature range, the performance of conventional collectors is not good enough. Due to higher efficiency values of parabolic-trough collectors (PTCs) at these temperatures, the combination of these two systems is of great interest [118]. Connection of NH_3–H_2O absorption chillers to a solar system requires solar collectors able to work efficiently at temperatures above 95 ◦C [119], such as the PTCs or high-efficiency stationary collectors. Air-conditioning and refrigeration facilities driven by a PTC solar field are still infrequent, while several test facilities using this technology have, however, appeared in the literature during the last 50 years.

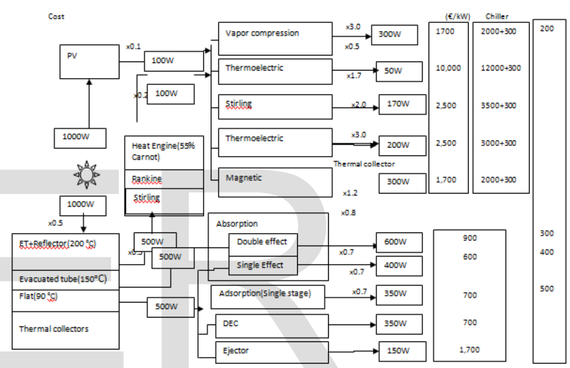

Fig.12 Performance and cost of various solar refrigeration systems[5].

5. Conclusions

Second law analysis gives a different view of the system performance and insight details of the irreversibility's that the first law analysis cannot provide. It is an effective tool, by providing information about how losses at different equipment are interdependent and where a given design should be modified for best performance of these system. The various solar cooling technologies discussed above in this article has ability to reduce the dependency on conventional energy technologies in Saudi Arabia. Cooling demand has different characteristic at various locations of Saudi Arabia. To provide cooling for one specific type of application, the right cooling system must be selected in order to meet the required cooling characteristics and temperature level. The solar Refrigeration system performance depends on the cooling & heating subsystems, and also the solar converter efficiency. A suitable type of solar collector must also be selected in order to provide the right driving temperature for the selected cycle. The dimension (size) of the solar collector strongly depends on the climatic condition of the region. Solar thermal absorption system looks to be the best option, among the solar cooling technologies due to its maturity in the market. Solar electric driven refrigeration system is also good option, but due to its high initial cost and low efficiency the solar thermal systems are preferred. The following concluding remarks can be made:

- Exergy is a way to a sustainable development, and is very useful tool for calculating the actual performance of Solar Refrigeration system.
- All sorption cycles are in the process beginning with research laboratory to the market, but much more work is needed on cost minimization and its design.
- The exergy efficiencies of a solar collector, a PV and a hybrid solar collectors are very low , which require further improvement.
- Adsorption and chemical adsorption cycles appear to function well in small - scale applications such as small refrigerators; however, these cycles have a problem of corrosion and crystallization which need to be solved..
- For solar electric cooling technology, the cost is highly dependent on system Coefficient of Performance.

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