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Systematic comprehensive techno-economic assessment of solar cooling technologies using location-specific climate data

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ABSTRACT

A methodology for assessing solar cooling technologies is proposed. The method takes into account location specific boundary conditions such as the cooling demand time series, solar resource availability, climatic conditions, component cost and component performance characteristics. This methodology evaluates the techno-economic performance of the solar collector/chiller system. We demonstrate the method by systematic evaluation of 25 feasible combinations of solar energy collection and cooling technologies. The comparison includes solar thermal and solar electric cooling options and is extended to solar cooling through concentrated solar power plants. Solar cooling technologies are compared on an economic and overall system efficiency perspective. This analysis has implication for the importance of solar load fraction and storage size in the design of solar cooling systems. We also stress the importance of studying the relation between cooling demand and solar resource availability, it was found that overlooking this relation might lead to overestimations of the potential of a solar cooling system in the range of 22% to over 100% of the actual potential.

1. Introduction

Sustainable buildings will best be achieved by considering both end use efficiency and efficient and renewable energy supply options [1]. In this paper we consider 25 feasible combinations of solar energy collection and cooling technologies (solar cooling) suitable for sunny climates. A methodology for assessing these technologies is proposed and illustrated by application to Abu Dhabi, UAE.

According to the International Energy Agency (IEA), air-conditioning is the dominant energy consuming service in buildings in many countries. Conventional cooling technologies are characterized by high energy consumption during operation and they tend to cause high electricity peak loads because of their concurrent operations during periods of high ambient temperatures [2]. The situation in Abu Dhabi is no exception; electric cooling represents 41% of the total consumption of residential and commercial sectors, in addition electric cooling contributes to more than 60% of the peak capacity [3], this poses a huge stress on the electrical system both technically and economically similar to those experienced in the US and the Spanish electrical systems [4,5].

An abundant supply of solar radiation is available in Abu Dhabi. Meteorological data shows that Abu Dhabi has a strong potential for solar energy utilization, a yearly sum of around 2044 (kWh/m^2) of global solar irradiation on a horizontal surface was recorded for the year 2008, direct normal irradiation of around 1800 (kWh/m^2) was recorded for the same year [6]. At the same time, electricity consumption in Abu Dhabi is growing considerably and is on track to outpace planned generation capacity. According to Abu Dhabi Water and Electricity Company (ADWEC), the peak electricity demand in Abu Dhabi is expected to reach 12GW by 2020, this represents more than a thee fold increase compared to 2007 [7]. Given the high availability of solar energy and the high cooling demand, it is logical to propose the use of solar energy to supply a large portion of the air-conditioning load.

In the past, solar cooling did not attract much attention; it was not until the 1970s that the interest in solar cooling increased [8,9].

Abbreviations: AC, absorption chiller; Capex, capital expenditure; CGC, cooling generation COST; COP, coefficient of performance; CSP, concentrated solar power; DNI, direct normal irradiation; Elec, electric output; ES, energy storage; ETC, evacuated tube collector; FLH, full load hours; FPC, flat plate collector; GHI, global horizontal irradiation; GridCon, connected to the electricity grid; OE, overall efficiency; SEGS, solar energy generating system; TES, thermal energy storage; Th, thermal output; Tres, central receiver CSP plant; VC, vapor compression.

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Early efforts in the field of solar thermal cooling systems are presented in [10,11]. Solar cooling literature is rich in papers that assess, develop and optimize a wide variety of solar thermal cooling options, particularly absorption cooling systems [5,12–16].

A comprehensive review of the state-of-the-art solar cooling technologies is presented in [9]; it covers solar thermal, solar electric and other new emerging technologies. Another interesting study of the economic feasibility of solar cooling is presented in [17]. The study recommends large-scale cooling options through solar power plants in combination with vapor compression chillers as a centralized solar cooling option, and a decentralized solar cooling option through a local modular solar field and absorption chillers.

Although the solar cooling literature is extensive, the research has tended to focus on the cooling equipment rather than on the system as a whole [16]. There is not much published work that assesses and compares, on a consistent basis, different solar cooling technologies, both thermal and electric, as a complete system. A few review papers have provided a wide range of proven and state-of-the-art solar cooling technologies but followed a simplified comparison approach that does not account for variables that play a key role in the overall performance of the cooling system. Often in such reviews, the focus is on the technology performance but not on the system performance that is why average efficiencies of the cooling equipment and solar field are assumed sufficient [9]. However, when designing a solar cooling system this approach is inadequate. Besides, accounting for the performance of its separate components, such analysis should address the interaction of these components with weather and supply and demand fluctuations.

In this work we propose a methodology that may be used by planners and researchers to assess various solar cooling technologies with a consistent standardized approach. This methodology extends previous effort by taking into account the cooling demand, solar resource availability, cost of components and performance parameters. Two parameters that are fundamental to the economics of solar cooling are considered: solar load fraction and storage size. Solar load fraction [1] is defined as the ratio between the net annual useful solar cooling output (kWh) and the total cooling demand (kWh) as explained in greater detail in Section 4. The method provides a quantitative measure of the two important aspects of any solar cooling technology: thermal performance and system cost.

In Section 2, we describe the methodology developed for assessing solar cooling technologies. Section 3 defines the basic requirements for any solar cooling system; solar load fraction, cooling demand time series and storage assumptions. Section 4 presents the feasible solar cooling technologies that are considered for comparison along with the detailed technical analysis carried out for evaluating the performance of each solar cooling system. Section 5 describes the approach used in ranking these systems based on financial merits. The results and lessons from applying our methodology are presented in Section 6 while Sections 7 and 8 include discussion, observations and conclusions.

2. Methodology for solar cooling system comparative assessment

A solar cooling system consists of three main blocks: the solar field, the cooling equipment and the storage. The solar field will collect the solar energy and convert it into either electrical or thermal energy, which is then used to produce cooling power through compatible cooling equipment. Storage, either cold or hot, is used to extend the operating hours of the system beyond sunshine hours. Solar resource availability and cooling demand profile are also parts of the system boundary conditions. Choosing the most suitable solar cooling application is not a simple task. Hence we propose two indices to rank solar cooling technologies: cooling generation cost and overall efficiency. The primary index is the **cooling generation cost (CGC),** which represents the price-lifecycle cost, expressed as a net present values, paid per each kWh of cooling energy produced. Embedded in this figure are the costs of the solar field, cooling equipment, thermal storage, land, maintenance, installation and financing. In addition, this figure is based on the performance of each solar cooling technology and the demand time series. Cleaning of the solar field is not included in CGC as we assume that these costs are relatively small [18] and comparable across the different technologies. **overall efficiency (OE)** is the ratio between the cooling output and the solar input (GHI) on the total area of land consumed by the solar field, thus OE takes into account both the performance of the whole system plus land utilization.

In order to evaluate the CGC and OE indices in a comprehensive and consistent manner we followed the algorithm outlined in Fig. 1:

The first step in assessing solar cooling technologies is to identify the feasible combination of solar cooling based on the requirements of the specific case; feasible combinations for our study are identified in Fig. 2. Each solar cooling technology is then assessed separately; the performance of the solar field and that of the cooling equipment is evaluated based on weather conditions and on solar resource availability, this is discussed in Section 3. Next, the desired **solar load fraction** is set; solar load fraction is defined as the ratio between the net annual useful solar cooling output (kWh) and the total cooling demand (kWh), it is explained in greater detail in Section 4. Finally, an iterative process is initiated to find out the required solar field size which is done taking into account the following key parameters that affect the effective solar load fraction:

- Solar cooling output.
- Cooling demand time series.
- Storage size.

After the solar load fraction is met the economic assessment is carried out as discussed in detail in Section 5. The resulting CGC and OE values are then used for a comparative study of all the considered solar cooling technologies.

3. Technical assessment of solar cooling systems

Feasible solar cooling system technology combinations considered in our study are presented in Fig. 2. As can be seen, the main building blocks of a solar cooling system are, solar field, cooling equipment and storage. In this section technical aspects of these blocks are presented.

Solar cooling system performance was simulated for all 25 technologies on an *hourly* basis throughout a year, taking into account the variations in solar cooling supply and the cooling demand time series, see Fig. 1. Through this simulation the required sizes of both the solar field for each technology and the corresponding chiller capacity were evaluated based on the stipulated solar fraction and the storage assumptions.

In Fig. 2, each solar collection technology is combined with the best cooling technology it could interface with, based on the operating temperatures. Solar thermal collectors, which include concentrating and non-concentrating collectors [19–21], can be used in combination with sorption cooling technologies [22]. Another alternative is to convert thermal energy into electricity and use it to drive conventional vapor compression chillers. Photovoltaic cells (PV) convert solar irradiation directly into electricity; hence PV can be used to drive conventional high performance vapor

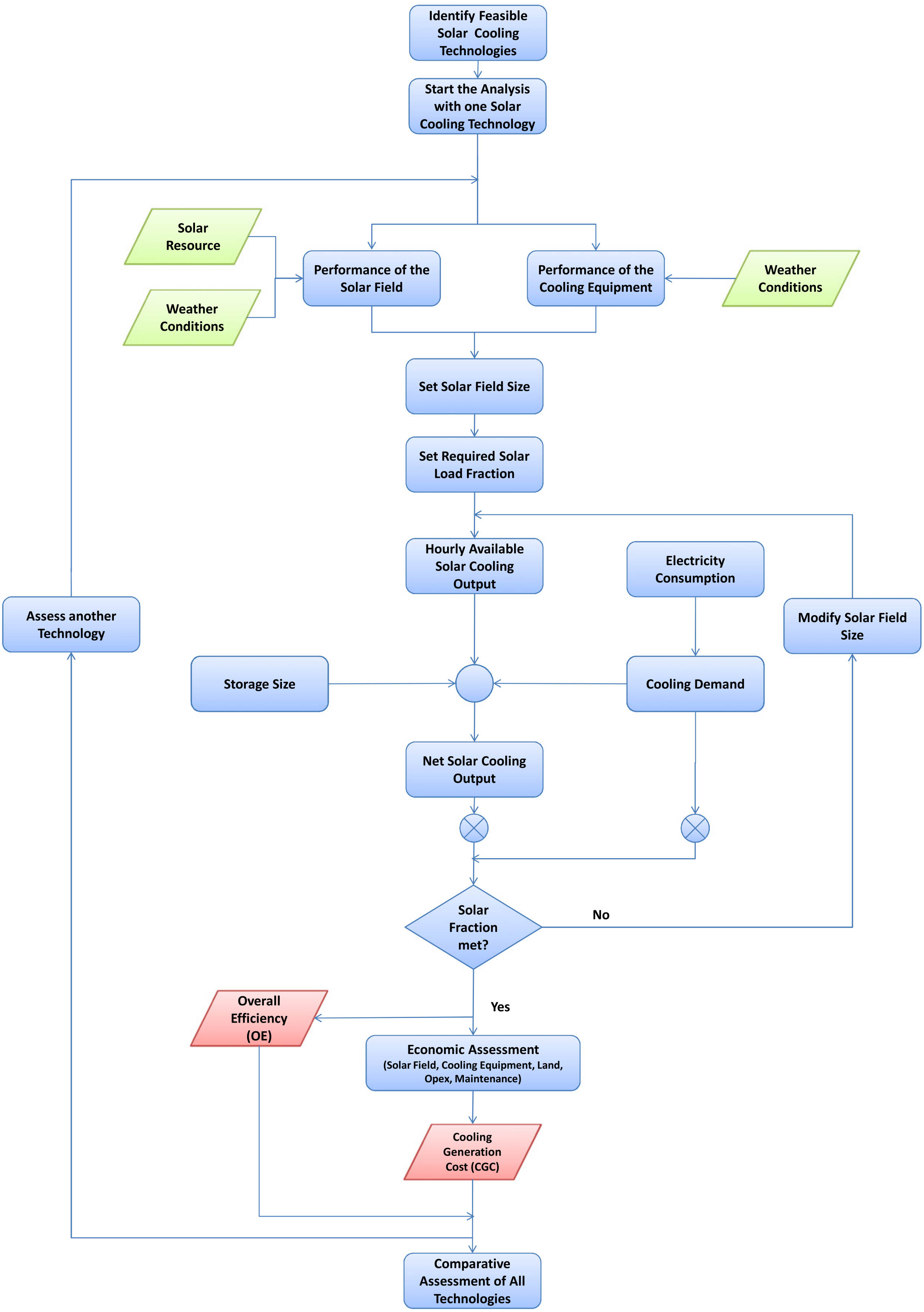

Fig. 1. Algorithm used in assessing solar cooling technologies.

compression chillers. Similarly, large-scale cooling plants could be realized through concentrated solar power plants (CSP); these technologies are compared with the aforementioned smaller scale solutions.

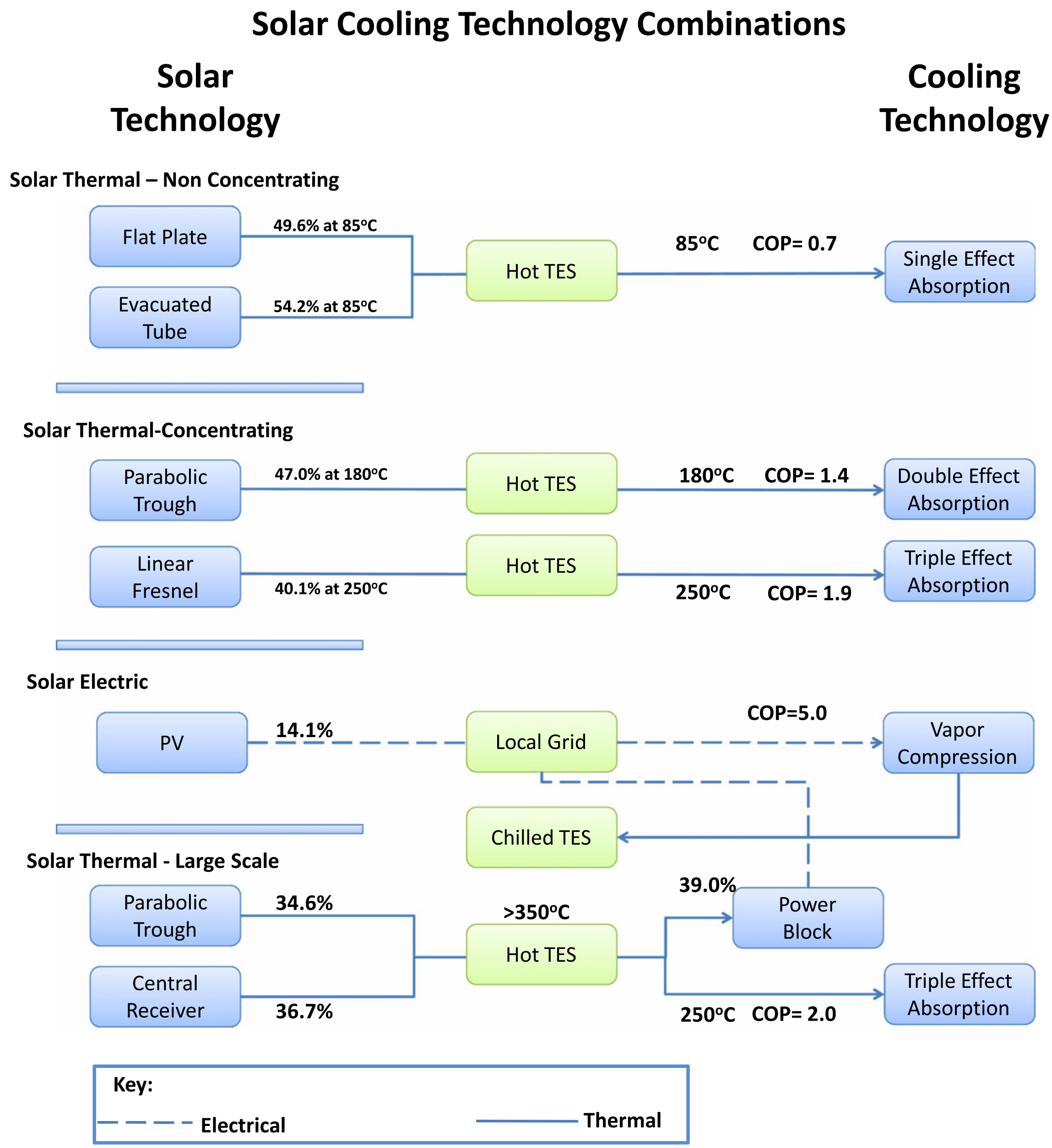

Fig. 2. Summary of solar cooling paths. Efficiency and COP values are average values shown for convenience; in simulation, these values are changing throughout the day. In addition these values are based on chosen commercially available products.

3.1. Solar collectors

The performance of solar collectors is evaluated according to their type. Since flat plate collectors (FPC) and evacuated tube collectors (ETC) have a large loss area to aperture area, their performance is greatly affected by variations in ambient condition, thus an *hourly* modeling of their performance is necessary. Photovoltaic cells (PV) are also affected by ambient conditions, especially those influencing the cell temperature (solar irradiation, ambient temperature and wind speed), thus an *hourly* performance simulation was also performed for PV cells. All performance figures needed for simulation of the solar collectors are based on manufacturers' data and standard test data of the respective collectors unless stated otherwise.

Although the performance of concentrating collectors which operate at elevated temperatures (above 180 °C) is also affected by ambient conditions, performance variations are less dramatic, thus average overall efficiency values provided by manufacturers are considered to be sufficiently accurate. In addition, the solar collectors of large scale solar cooling options are evaluated based on data from NREL [23]. The NREL report includes data for plants in operation and data based on cost and technology evolution.

3.1.1. Performance evaluation of flat plate and evacuated tube collectors

The thermal yield of FPC and ETC is found using the following equation [1]:

$$\text{Yield} = A * (I_t * \text{IAM} * \eta_o - C_1(T_a - T_{op}) - C_2(T_a - T_{op})^2) \quad (1)$$

where I_t is the total irradiation on the surface of the collector, this should be calculated for tilted surfaces [W/m^2], IAM is the Incident angle modifier which is a result of optical properties dependence on angle of incidence, η_o is the Optical efficiency of the collector at normal incidence based on area A, C_1 and C_2 is the performance constants of the collector supplied by the manufacturer [W/(m^2 K)], [W/(m^2 K^2)], respectively, T_a is the ambient temperature[°C], T_{op} is the average operating temperature[°C], A is the aperture area of the collector [m^2].

Solar irradiation on the tilted surface of the collector must be evaluated for each hour of the year. Different models are available to estimate the total irradiation on a tilted surface; we use the Hay Davies model [1] denoted by Eq. (2) below:

$$I_T = (I_b + I_d A)R_b + I_d(1-A)\left(\frac{1+\cos\beta}{2}\right) + I\rho_g\left(\frac{1-\cos\beta}{2}\right) \tag{2}$$

where I_T is the total irradiation on a tilted surface [W/m^2], I_b is the beam irradiation on a horizontal surface [W/m^2], I_d is the diffuse irradiation on a horizontal surface [W/m^2], I is the total irradiation on a horizontal surface [W/m^2], A is the anisotropy index which is the ratio between beam irradiation to the extraterrestrial irradiation, both on a normal surface, β is the tilt angle of the collector from the horizontal [rad], ρ_g is ground reflectance and is assumed to be zero since we assume a densely packed solar field, R_b is a geometric factor calculated as the ratio between beam radiation on a tilted surface to that on a horizontal surface, R_b is given by the equation:

$$R_b = \frac{\cos\theta}{\cos\theta_z} \tag{3}$$

where θ is the angle of incidence of beam irradiation on the tilted surface and θ_z is the solar zenith angle.

IAM is a function that represents the dependence of both transmittance and absorption on the angle of incidence. For FPC, the incident angle modifier is given for each collector as a curve from the SPF catalogue [24], an equation was fit to the curves given by the SPF catalogue for each collector. This equation only applies for beam radiation and circumsolar diffuse radiation. For isotropic diffuse radiation an incident angle of 60° is assumed (it is a result of integrating the curve over all the incident angles [1]).

For ETC two curves are used for incident angle modifiers, one in the transverse direction and one in the longitudinal direction. Incident angle modifier for diffuse irradiation is again calculated on the mean angle of 60°. For beam irradiation the transverse and longitudinal angles change hourly. These are calculated taking into account the tilt angle of the collector. We derived Eqs. (4) and (5) which are used to define these angles for ETCs:

$$\tan(\theta_{\text{long}}) = \frac{\sin(\theta_z)\cos(\gamma)\cos(\beta) - \cos(\theta_z)\sin(\beta)}{\cos(\theta_z)\cos(\beta) - \sin(\theta_z)\sin(\beta)\cos(\gamma)} \tag{4}$$

$$\tan(\theta_{\text{trans}}) = \frac{\sin(\theta_z)\sin(\gamma)}{\cos(\theta_z)\cos(\beta) - \sin(\theta_z)\sin(\beta)\cos(\gamma)} \tag{5}$$

The resultant IAM is the product of $\text{IAM}_{\text{trans}}(\theta_{\text{trans}})$ and $\text{IAM}_{\text{long}}(\theta_{\text{long}})$.

3.1.2. Performance evaluation of photovoltaic cells

Performance of PV cells is affected by the cell temperature. The conversion efficiency is found by evaluating the following two equations, the first equation represents the variation of PV cell efficiency with the cell temperature, and second equation is used to evaluate the PV cell temperature:

$$\eta = \bar{\eta}(1 - \beta(T_{\text{cell}} - T_{\text{STC}})) \tag{6}$$

$$T_{\text{cell}} = T_a + \frac{\tau\alpha_0 * \text{IAM} * I_t - I_t * \eta}{2 * h_o} \tag{7}$$

where η is the PV conversion efficiency, $\bar{\eta}$ is the nominal efficiency at standard testing conditions, β is the efficiency dependence on the cell temperature which is given by the manufacturer, T_{cell} is the PV cell temperature, T_{STC}: is the temperature at standard testing conditions = 25 °C, T_a is the ambient temperature, I_t is the total irradiation on the surface of the PV cell which is evaluated in the same way as that for FPC and ETC, $\tau\alpha_0$ is the absorptance–transmittance product for PV cells at normal incidence, assumed 0.9 as a rule of thumb [1], h_o is the wind convection factor given by the linear equation below [25,1]:

$$h_o = 2.8 * V + 3.0 \tag{8}$$

where V is the wind speed [m/s], IAM is the incident angle modifier for PV cells given by Ref. [1]:

$$\text{IAM} = \frac{\tau\alpha(\theta)}{\tau\alpha_0} \tag{9}$$

$$\tau\alpha(\theta) = e^{-\left(\frac{KL}{\cos\theta_r}\right)}\left[1 - 0.5\left(\frac{\sin^2(\theta_r-\theta)}{\sin^2(\theta_r+\theta)} + \frac{\tan^2(\theta_r-\theta)}{\tan^2(\theta_r+\theta)}\right)\right] \tag{10}$$

where K is the glazing extinction coefficient (4 m^{-1}) [1], L is the glazing thickness (2 mm)[1], $\tau\alpha_0$ is the absorptance–transmittance product for PV cells at normal incidence, assumed 0.9 as a rule of thumb [1], θ is the incident angle, θ_r is the refraction angle.

The refraction angle is calculated using Snell's law

$$\theta_r = \sin^{-1}\left(\frac{\sin\theta}{1.526}\right) \tag{11}$$

3.2. Cooling equipment

Similar to the solar field, performance of cooling equipment is affected by ambient conditions. The variation of COP with ambient temperature is considered to be the dominant factor, and therefore we derived equations that relate the COP of each cooling technology to the ambient temperature. This data was obtained from manufacturers' data sheets and published literature [26,27].

Air-cooled chillers are assumed for all technologies. Although air-cooled chillers have higher investment cost and running electricity cost than those with wet cooling towers, they have the advantage of not using water during operation. In dry climates where the main source of water is desalination this is a major advantage. It should be noted however that air cooling is not necessarily the best heat rejection technology especially for solar cooling applications. A thorough investigation must be carried out before deciding on the heat rejection method to be used. This should account for the actual cost of water associated with wet cooling towers on one hand and the potential improvement in efficiency on the other.

For vapor compression chillers a COP of 5 is considered at the American Refrigeration Institute (ARI) standard test conditions (35 °C). For absorption chillers nominal COP at 30 °C condenser temperature is assumed to be 0.7, 1.4 and 2.0 for single effect, double effect and triple effect absorption chiller respectively, all based on manufacturer's data. COP variation with ambient temperature is shown for the four chiller types in Table 1, these equations are applied when the difference between the nominal condenser temperature of 30 °C and the ambient temperature is greater than 4 °C, otherwise the COP is assumed to be the nominal COP.

Table 1
COP variation with ambient temperature, these equations are applied when the difference between the nominal condenser temperature of 30 °C and the ambient temperature is greater than 4 °C.

Chiller type	COP equation
Vapor compression	$\text{COP}_{\text{VC}} = 5 * (-0.03647 * T_a + 2.271)$
Single effect absorption	$\text{COP}_{\text{A-1}} = 0.7 * (-0.05046 * (T_a + 4) + 2.472)$
Double effect absorption	$\text{COP}_{\text{A-2}} = 1.4 * (-0.05002 * (T_a + 4) + 2.457)$
Triple effect absorption	$\text{COP}_{\text{A-3}} = 2.0 * (-0.013 * (T_a + 4) + 1.39)$

* T_a is the ambient temperature.
* Constants are based on manufacturer's data.

3.2.1. Chiller auxiliary energy

Another important factor taken into account for the cooling equipment is the auxiliary electrical power needed to run the absorption machines, the energy is mainly consumed by the fans of the dry cooling tower. Obviously, this energy demand differs according to the chiller technology and the performance of the air cooled condenser, the auxiliary power needed by the chiller is estimated by Eq. (12). The equation estimates the amount of electric power needed to run the dry cooling tower. This power is related to the amount of heat rejected by the chiller and the efficiency of the cooling tower at varying ambient temperatures.

$$P_{\mathrm{electric}} = \left(\eta_{\mathrm{fluid\_cooler}}\left(1+\frac{1}{\mathrm{COP}}\right) * (0.04681 * \mathrm{DTI} - .04019)\right) \tag{12}$$

Where P_{electric} is the auxiliary electric power in (kW_{electric}) for each (kW_{capacity}), DTI is the difference between the condenser operating temperature and the ambient temperature (°C), is the ratio between the amount of electricity needed to run the fluid cooler and the amount heat rejected. It assumed to be 2% based on manufacturer's data [28]. In the same equation, the relation between the efficiency of the tower and ambient temperature is given by based on [26].

We assume that each of the solar cooling configurations includes sufficient PV panels to produce the auxiliary energy needed. Hence putting all configurations on equal footing since the auxiliary energy requirements is different. In other words we assume that the annual auxiliary energy is 100% met by grid connected PV cells, whose capital cost is included in the final cost of the system; this assumption enhances system autonomy that we are ideally targeting as well as making a fair comparison between all the solar cooling configurations.

In sizing the auxiliary field, it is assumed that the total electricity output from the field through the year is equal to the total electricity demand, this implies negligible amount of seasonal storage in the grid only for auxiliary power supply.

3.3. Energy storage options: thermal and grid-connection

In order to accommodates variations in cooling supply and demand and allow overnight operation, energy storage is required. Hot TES between the solar field and the cooling equipment is possible for all solar cooling technologies except PV with VC chillers since the direct output of PV is electricity. Thermal storage of chilled water is also another option for all technologies. Another means of storage for solar technologies with electric output (PV and $\mathrm{CSP}_{\mathrm{electric}}$) is local electricity grid storage. See Fig. 2.

The storage size is assumed to be 10 full load hours[1] (FLH) for all technologies, which ensures adequate energy storage for meeting night cooling demand based on our analysis of the cooling demand time series and the solar resource availability given a 75% solar load fraction. A simple model based on energy balance is used for modeling storage, which is sufficient for our current analysis needs.

It is worth mentioning that for the PV–VC system chilled water storage is only possible after the chiller. Thus the chiller has to be sized to be able to accommodate the variations in the solar input during the day (this is handled by hot storage in other systems). Thus the size of the chiller in this case must be more than twice the capacity needed for other technologies with hot thermal storage.

For the storage in the local grid, we assume that the energy fed to the grid during sunshine hours could be retrieved at nighttime. But excess energy generated could not be retrieved in the next solar day, and is fed to the grid with no revenue. This means that storage is done only on a daily basis and seasonal storage is not allowed. This was assumed to make a fair comparison with other non-electric technologies since excess energy generated from these systems could be also used for different purposes and generate revenue. For example, thermal energy could be used for desalination or industrial process heat applications and so on. Since technical and economic potential of the use of excess is region-specific and can vary significantly it is not included in CGC.

4. Solar load fraction, storage and seasonal trend of cooling demand

The primary motivations for promoting solar cooling are to (i) reduce greenhouse gas emissions, (ii) reduce peak electricity loads in strained infrastructures and (iii) create semi-autonomous buildings. Since these motivators exist to varying degrees in all potential solar cooling applications they were used in defining the system requirements for our comparative analysis.

All solar cooling technologies presented in this case study are sized to achieve a ***solar load fraction*** of 75% of the total cooling demand for the sake of comparison. Solar load fraction is defined as the ratio between the net useful solar cooling output (kWh) and the total cooling demand (kWh).

Solar load fraction is calculated using the formula below:

$$\text{Solar load fraction} = \frac{\sum_{n=1}^{n=365} \text{Useful cooling yield}(n)}{\sum_{n=1}^{n=365} \text{Cooling demand}(n)} * 100\% \tag{13}$$

Useful cooling yield is calculated as follows:

$$\text{Useful cooling yield} = \left\{ \begin{array}{l} Y(n), D(n) > Y(n) \\ D(n), D(n) \le Y(n) \end{array} \right\} \tag{14}$$

where $D(n)$ is the average cooling demand of day (n) in [kWh] and $Y(n)$ is the average cooling yield of day (n) in [kWh].

In Fig. 3, the total cooling energy generated by the solar cooling system in excess of the cooling demand is depicted as a function of the solar load fraction for four representative solar cooling technologies (this curve is based on evaluating Eqs. (13) and (14) for different solar load fractions). Fig. 3 shows how difficult it is to increase the target solar load fraction without generating a considerable amount of excess energy. It also becomes harder to satisfy the peak cooling demand. This is a direct consequence of the discrepancy between the cooling demand and the solar resource availability.

Storage plays a key role in reducing excess energy production. It also provides the system with a degree of independence from the solar input, the level of this independence is related to the size of storage employed. A very high capacity of thermal storage is required to completely decouple supply from demand. In such a case a solar load fraction of unity could be approached much easier, refer to Fig. 3. The problem however is that such large sizes of storage are difficult to justify from an economic point of view [5]. In our systems storage size of 10 FLH was assumed.

Proper design of a solar cooling system is largely a matter of finding the optimal solar load fraction. The design is a compromise between the degree of autonomy that we want to achieve in the operation of a cooling system and the efficient utilization of capital needed to build the solar cooling system. The optimal solar load fraction may be different from one system to another but for the purpose of this study we assumed a constant solar load fraction throughout to facilitate a transparent comparison, and to produce easily interpreted results. We believe the results in terms of CGC ranking would not much different with a rigorous optimization under a range of reasonable component cost scenarios.

[1] One full load hour (FLH) of storage means that the storage has enough energy to allow operation for one hour at full load with no solar or other input.

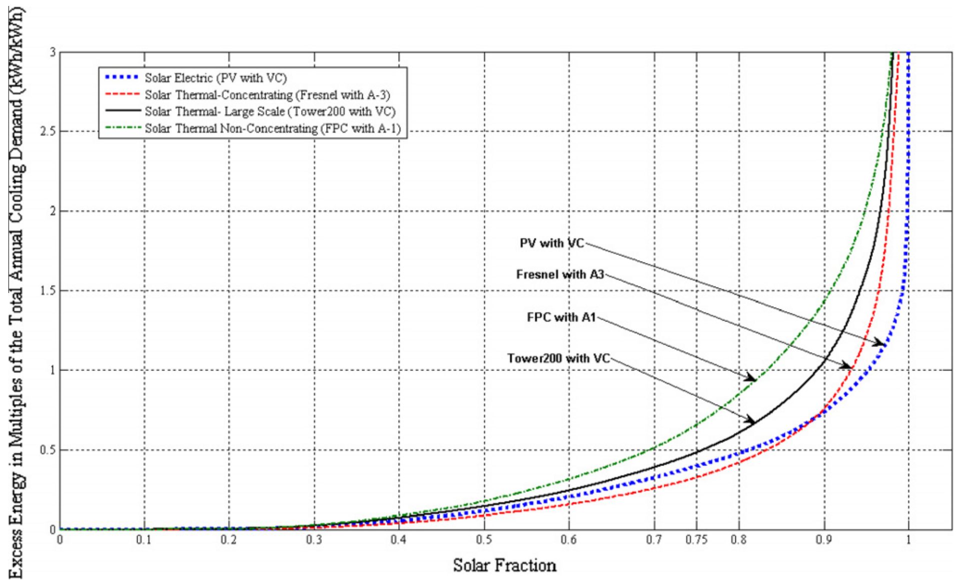

Fig. 3. Excess energy versus solar load fraction. Higher solar load fractions generate more energy in excess of the demand which leads to higher CGC and lower OE.

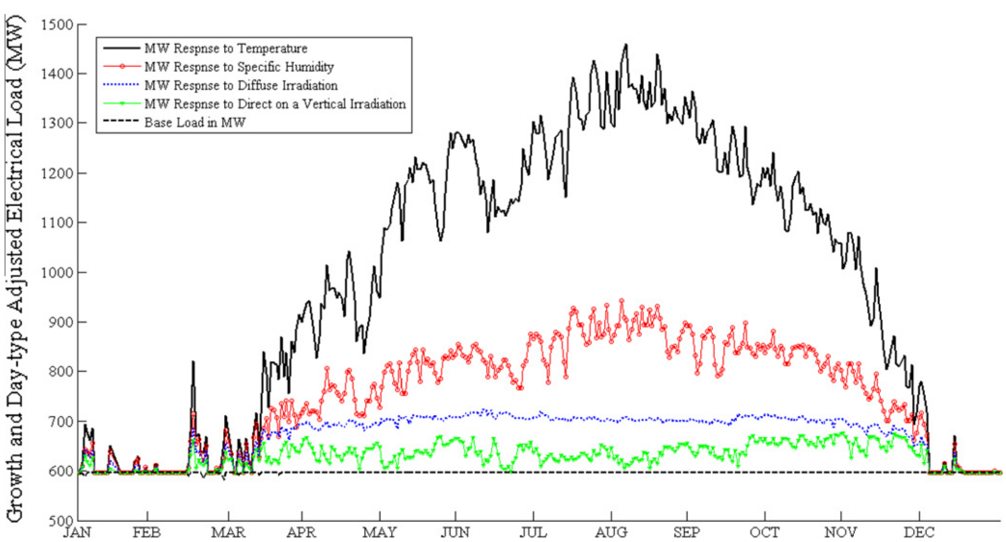

Fig. 4. Contributions of different weather parameters to the electricity consumption [3].

To illustrate the analysis method, the cooling load profile of Abu Dhabi Island is used to represent the cooling demand presented to our cooling system. The maximum cooling capacity is assumed to be 1000 tons (equivalent to 3517 kW of cooling capacity, where the conversion factor is 3.517 (kW/ton)), thus the cooling profile is scaled accordingly. The cooling demand load profile is inferred from electricity consumption data in another study [3]. Fig. 4 depicts the seasonal variation of electrical demand of Abu Dhabi Island. The seasonally varying component of demand is mainly cooling; the cooling demand is further divided into responses, represented by areas between the curves, to ambient temperature, specific humidity, vertical component of direct normal irradiation and diffuse irradiation.

5. Economic analysis of alternative solar cooling options

The solar cooling technologies are ranked according to the cooling generation cost (CGC), this cost represents the price paid for each k Wh of cooling, analogous to the levelized energy cost used in the analysis of utility generation and demand-side management options. CGC takes into account the capital cost of the solar field, the cooling equipment, the thermal storage, the land and the problems of supply–demand mismatch and how performance of the system changes with conditions. It also includes the costs of annual maintenance throughout the life time of the plant. The life time of the plant is embedded in the finance parameter α.

$$\mathrm{CGC}\left[\frac{\$}{\mathrm{kWh}}\right]\frac{\text{Specific annual payments}\left[\frac{\$}{\mathrm{yr\ kW}}\right]}{\text{Number of full load hours per year}\left[\frac{\mathrm{h}}{\mathrm{yr}}\right]} = \frac{[\alpha + M] * \mathrm{Capex} + \Sigma O}{\mathrm{FLH}} \tag{15}$$

where Capex[2] is the capital cost [\$/kW cooling capacity], α is the ratio between the capital investment and the annual fixed payments, M is the annual maintenance cost [% of Capex per year], ΣO is the annual operation cost [\$/kW cooling capacity], FLH is the utilizable cooling yield expressed as the number of full load hours per year [hr/yr].

[2] Component costs for Eq. (15) are reported in the Appendix as percent of Capex.

- Capex is the sum of the investment costs of the solar field, the cooling equipment, the storage and the land cost; this also includes installation and commissioning costs. System integration costs are not included in this cost.
- Financing conditions are embedded in α. α represent the ratio between the annual payments and the present worth of the capital investment. This is assuming a uniform payment series and a fixed interest rate over a 25 yr analysis period, the value of α is assumed, in this example, to be 10%.
- Maintenance is assumed to be constant throughout the life time of the plant (25 yr), it is normally provided in the contract as a percentage of Capex. Values between 0.5% and 3% of Capex are reported in the literature for the solar field and the cooling equipment [29,16]. A maintenance cost of 1.5% for solar technologies with thermal output is assumed, 2% for solar thermal technologies with electric output, and 1% for PV. For cooling equipment, a value of 2% is used [16].
- Operational cost are mainly dominated by the cost of electricity needed to run the several pumps and fans of the system, since we assume that this demand is met by extra PV cells which price is included in Capex, operational costs are assumed to be zero.
- The number of full load hours translates to the total amount of cooling generated by each technology in k Wh, normalized by the total chillers' capacity in kW. This number is the same for all cooling technologies by virtue of the fact that we have assumed that all technologies should satisfy a solar fraction of 75% of the cooling load.

6. Comparative assessment of solar cooling technologies

We used a comprehensive selection of solar cooling options, from manufacturers of solar fields and cooling equipment to perform a comparative analysis of the possible systems. The results of the analysis are presented in Appendix A without the disclosure of the manufacturer due to the confidential nature of the data provided to us by them. Solar processes are generally characterized by high initial cost and low operating costs [19]. The high capital cost of utilizing solar power has always been the major obstacle to wide deployment of solar technologies. On average, the capital investment (Capex) accounts for more than 85% of the CGC. The contributions of the main components of the capital cost are shown in Fig. 5.

In the paper we refer to some of solar cooling configurations as "large scale" options. These options are based on performance analysis of plants with capacities ranging from 13 MWe to 400 MWe, presented in [23]. Mostly these plants are non-modular and/or their performance figures are based on a full capacity plant, thus a distinction is important from small scale options. In addition, due to economy of scale these options may be expected to provide cheaper CGC than other smaller scale options.

Because the solar field constitutes roughly 75% of the capital cost (Capex), investing in the most efficient available cooling technology is almost always justified. Moreover, the high Capex fraction represented by the solar field makes the ranking of various solar cooling combinations sensitive to the solar collectors prices. This explains the spread of different products from the same category over the cost range from around 4.0 (¢/kWh) to over 30.0 (¢/kWh).

Fig. 6 depicts the ranking of all the solar cooling options based on their CGC. Among small scale options, Fresnel-based and PV-based solar cooling options are the most economical. The CGC of FPC and ETC is relatively high although their specific collector cost is not. This is a direct consequence of the relatively low performance of the single effect absorption chiller connected to these collectors, especially at high ambient temperatures. From the same figure, the potential of large-scale cooling plants in providing a cheap cooling option is evident.

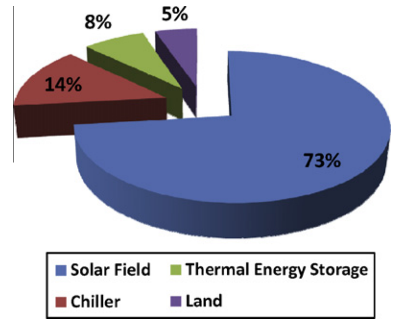

Fig. 5. Average Capex breakdown, (land cost considered is of desert in Abu Dhabi).

Overall efficiency (OE) is the ratio between the cooling output and the solar input (GHI) incident on the total area of land consumed by the solar field. Thus OE takes into account both the performance of the whole system plus land utilization. Overall efficiency hence represents the amount of cooling that each technology produces per each square meter of land, and not of collector. Thus OE accounts for the different packing factors of solar collector types. Fig. 7 depicts the overall efficiency of all solar cooling technologies considered. It is shown that PV technologies represent the most efficient options followed by Fresnel concentrating technologies, while non-concentrating technologies have poor overall efficiencies is spite of their good land utilization and ability to use UAE's abundant summer diffuse irradiation, this is due to the performance of the cooling equipment.

In Table 2, the effects of land cost on the resultant CGC is presented, land costs are based on Abu Dhabi standard land costs, however desert land cost is just an estimate since there is no standard price for it. In the land cost calculation we assume that the major part of the land is occupied by the solar field, area for the cooling equipment is not included.

Two observations could be made from Table 2. First, large-scale cooling plants have poor land utilization (ratio between aperture area and area of land needed). This is because the solar field is usually laid out in order to minimize losses due to shading and blocking resulting in a high land surface requirement and consequent cost. Second is that solar cooling technologies that have low overall efficiency need larger solar fields which also results in a higher land cost, this is the case for non-concentrating collectors with absorption chillers.

It should be mentioned that land utilization efficiency greatly depends on the collectors' layout and tilt angle, when land utilization is important it is reasonable to sacrifice a portion of the overall output to enhance land utilization. Efficient use of land or roof area becomes a significant issue when the cost of land is high, when the available land is limited or when the site preparation costs are high.

7. Seasonal balancing of supply and demand

It was found that the amount of cooling that can be supplied by each technology is not the only factor that should be considered in deciding which solar cooling technology to choose. In fact the cooling demand time series is a very important variable that is often overlooked. Unless seasonal storage is feasible (almost never), an overestimation of the potential of a solar cooling system results if the seasonal demand time series is not considered. Based on our case study this overestimation is in the range of 22–100% more than the actual potential. At solar fractions higher than 90%, the

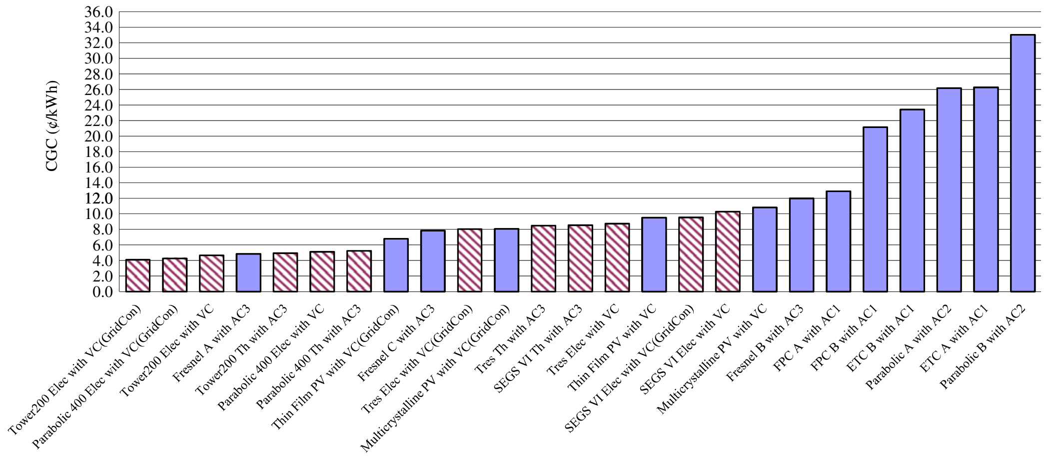

Fig. 6. CGC (hatched bars represent large scale options, where: Tres, SEGS VI, Tower200 and Parabolic400 are the names of these plants as defined in Appendix B).

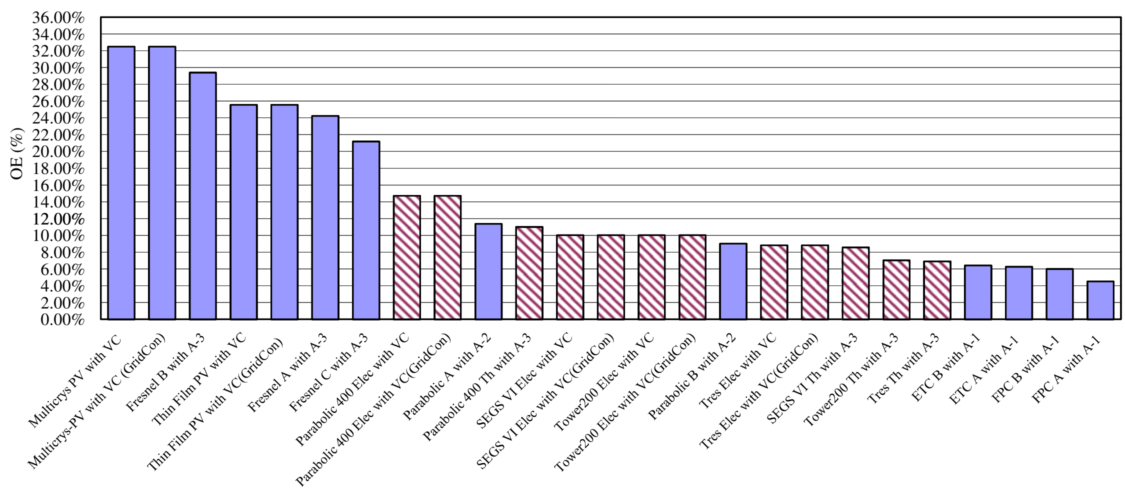

Fig. 7. Yearly averaged overall efficiency (hatched bars represent large scale options, where: Tres, SEGS VI, Tower200 and Parabolic400 are the names of these plants).

Table 2
Effects of land cost on CGC based on Abu Dhabi land costs (similar technologies in terms of land utilization efficiency are grouped).

Technology	CGC in (¢/kWh) for free land	CGC in (¢/kWh) for desert land (10$/m^2)	CGC in (¢/kWh) for suburban land (280$/m^2)	CGC in (¢/kWh) for core urban land (800$/m^2)
ETC A with A-1	25.69	26.26	41.74	71.54
ETC B with A-1	22.84	23.41	38.8	68.43
FPC A with A-1	12.14	12.91	33.71	73.77
FPC B with A-1	20.56	21.16	37.34	68.52
Parabolic A with A-2	25.57	26.11	40.75	68.94
Parabolic B with A-2	32.3	32.99	51.43	86.95
Fresnel A with A-3	4.57	4.74	9.18	17.74
Fresnel B with A-3	11.74	11.88	15.54	22.6
Fresnel C with A-3	7.54	7.73	12.82	22.62
Thin film PV with VC	9.35	9.52	14.16	23.09
Multicrystalline PV with VC	10.67	10.81	14.57	21.82
Thin film PV with VC(GridCon)	6.62	6.79	11.43	20.36
Multicrystalline PV with VC(GridCon)	7.94	8.08	11.84	19.09
SEGS VI Elec with VC	9.82	10.29	23.15	47.93
Parabolic 400 Elec with VC	4.79	5.12	13.88	30.76
SEGS VI Elec with VC(GridCon)	9.06	9.53	22.4	47.17
Parabolic 400 Elec with VC(GridCon)	3.95	4.27	13.03	29.91
SEGS VI Th with A-3	7.96	8.43	21	45.22
Parabolic 400 Th with A-3	4.78	5.14	14.93	33.78
Tres Elec with VC	8.21	8.76	23.36	51.49
Tower200 Elec with VC	4.2	4.67	17.55	42.36
Tres Elec with VC(GridCon)	7.5	8.04	22.65	50.78
Tower200 Elec with VC(GridCon)	3.62	4.1	16.98	41.78
Tres Th with A-3	7.81	8.39	23.99	54.03
Tower200 Th with A-3	4.27	4.84	20.16	49.68

amount of useful cooling energy was found to be as low as 50% of the total cooling energy generated. Similar conclusions are obtained in [16] and [5].

Fig. 8 illustrates the relation between global irradiation on a horizontal surface (GHI), direct normal irradiation (DNI) and the daily average cooling demand. Although the average annual yield of GHI and DNI are comparable, 2044(kWh/m^2) and 1800(kWh/m^2), respectively, their relation with the cooling demand is different. The seasonal relation between GHI and the daily average cooling demand is shown in Fig. 8, although there is a time lag in the order of weeks. For Abu Dhabi, the relation between DNI and the daily average cooling demand is not as strong as that with GHI. The daily variations in DNI are large and on average DNI values are lower during peak cooling demand in summer time than in wintertime due to the higher humidity in the summer.

Non-concentrating solar thermal technologies and PV cells benefit from the fact that they can effectively utilize GHI which, in many cases, may be better correlated with the demand [1]. This characteristic helps in mitigating the electrical demand during summer time, where the peak electrical demand often occurs, and thus reducing the seasonal variation in electrical demand. For concentrating collectors the case is different, these technologies can only use DNI in which the daily average DNI and electricity demand for cooling are less well correlated.

Having a discrepancy between the supply and demand adds to the complexity of solar cooling. The simplest approach is to oversize the solar field to meet the peak demand in summer time but this leads to dumping considerable amounts of energy during times of low cooling demand (winter). In order to accommodate these discrepancies between the supply and demand without having to dump much of the costly energy produced, seasonal thermal storage could be a solution, however the very high storage volumes needed to accomplish that makes this solution unfeasible [5], except possibly in cases of large scale district cooling systems with in-ground storage. Therefore a hybridization scheme is adopted to allow solar cooling to provide cooling when there is enough solar irradiation, and the gaps are filled by other conventional means (hybridization is implicitly adopted since the solar load fraction is always less than 100%).

The cooling loads of high performance buildings, on the other hand, may exhibit less seasonal variation than the current Abu Dhabi building stock. Thus a valid assessment is seen to hinge on a system model that accounts for solar resource, the type of collector technology, the cooling plant performance characteristic, the sensitivity of component performance to weather, and the seasonal cooling load variations.

8. Conclusions

An assessment of solar cooling technologies was presented in this work. The methodology is based on assessing the performance of each solar cooling technology as a system taking into account cost and performance parameters in addition to important boundary conditions of weather and cooling demand.

The assessment was applied to 25 solar cooling technologies based on the climatic conditions and cooling demand time series of Abu Dhabi, UAE. The technologies were ranked, on one hand, from an economical perspective and from a performance perspective on the other. Results show that large-scale cooling plant options are the most economical. On a smaller scale, Fresnel concentrators and thin film PV cells are the most economically viable. In terms of overall efficiency however multicrystalline PV cells with vapor compression chillers were the most efficient option of all. Solar resource availability is a major factor in determining the most suitable solar cooling technology for a certain location. Not only the amount of annual solar yield (DNI or GHI), but also the distribution of this resource through the year and its relation with the cooling demand time series are important in the analysis. In addition, the distribution of global irradiation used by non-concentrating technologies may be completely different from the distribution of direct irradiation used by concentrating technologies, depending on the location. In assessing the viability of solar cooling technologies, it is important to consider the distribution of these parameters throughout the cooling season and their interaction with the cooling demand time series.

From the study, we concluded that two very influential parameters determine the most economical solar cooling option; the cost of the solar collection technologies and the performance of the refrigeration technologies. Hence, choosing the most efficient cooling equipment added to the technical developments in the field of cooling technologies would result in significant reduction in investment costs of solar cooling technologies. The same is true for cost evolution of solar harvesting technologies.

It follows then that the choice of the heat rejection mechanism (dry versus wet cooling) in a solar cooling system is very

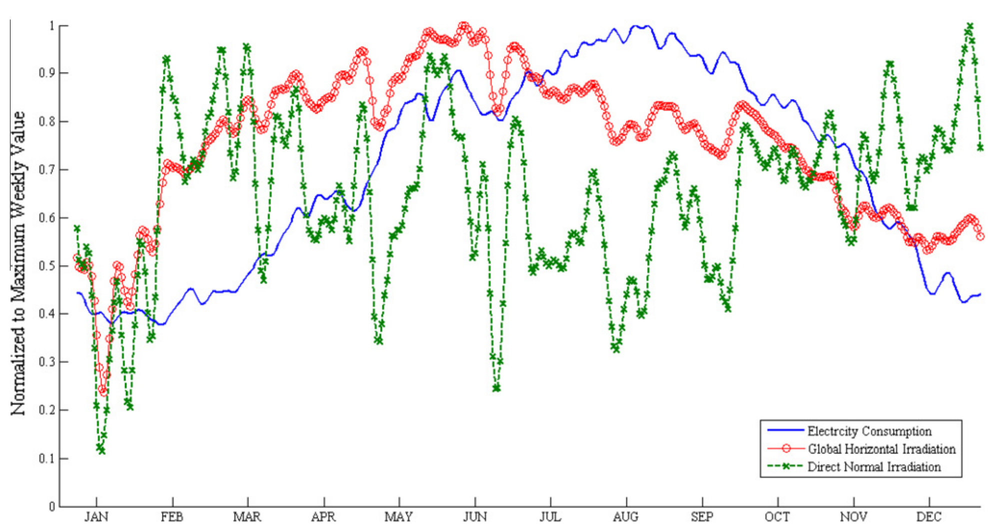

Fig. 8. Seven-day moving averages of electricity consumption, global horizontal irradiation and direct normal irradiation for Abu Dhabi.

important, because it directly affects cooling equipment performance. In addition we found that the degradation in the performance of absorption chillers in such hot conditions is severe. This may suggest that these technologies are either not very well suited to hot climates and that more effective condenser cooling technologies must be used or that more effort is needed to develop absorption technologies optimized for solar thermal power. Finally, we should point out that the performance of the solar cooling system is greatly dependent on the chosen solar load fraction and storage size. Proper selection and optimization of these parameters is important in the design of a solar cooling system, future work is focused on this aspect.

To sum up, the methodology presented, although it is an extension of previous efforts, combines several important considerations. First it is comprehensive in the sense that it compares several solar cooling technologies, both thermal and electric, and that it analyses the solar cooling as a system and not as disconnected components. Second it takes into consideration important boundary conditions such as the cooling demand and supply variations through the year. Third the variation in performance of the cooling equipment (change of COP with ambient temperature) and the performance of solar field (efficiency versus solar resource and ambient conditions) are considered. Fourth, auxiliary power requirement for the cooling equipment, which is a function of the ambient temperature, condenser efficiency and COP of the chiller, is calculated for each of the considered technologies. And fifth, PV panels are used to supply this auxiliary power in order to facilitate a fair comparison between the options. The auxiliary power requirement is important because it varies substantially among options; using PV-powered auxiliaries ensures that all of the compared options are truly providing the same annual solar load fraction.

Appendix A. Summary of results and cost break down

Technology	Capex ($/kW capacity)	CGC (¢/kWh)	CAPEX of CGC	Main of CGC	Solar/ Capex	Chiller/ Capex	TES/ Capex	Land/ Capex	Aux PV/ Capex	Excess energy ratio	Overall efficiency	Field size (m^2)
ETC A with A-1	8610	26.26	87.36%	12.64%	91.26%	3.90%	2.31%	2.50%	0.03%	44.87%	6.27%	45,380
ETC B with A-1	7681	23.41	87.41%	12.59%	90.22%	4.37%	2.59%	2.78%	0.04%	45.14%	6.41%	45,120
FPC A with A-1	4262	12.91	87.98%	12.02%	80.60%	7.87%	4.67%	6.78%	0.07%	46.68%	4.51%	61,000
FPC B with A-1	6946	21.16	87.47%	12.53%	89.02%	4.83%	2.87%	3.24%	0.04%	46.70%	5.99%	47,470
Parabolic A with A-2	3232	9.52	90.48%	9.52%	59.95%	22.66%	15.40%	1.99%	0.00%	34.31%	25.57%	13,600
Parabolic B with A-2	3672	10.81	90.50%	9.50%	65.07%	19.95%	13.55%	1.43%	0.00%	35.09%	32.51%	11,040
Fresnel A with A-3	8574	26.11	87.48%	12.52%	90.64%	3.58%	3.32%	2.37%	0.08%	51.59%	11.38%	31,480
Fresnel B with A-3	10,824	32.99	87.43%	12.57%	92.10%	2.84%	2.63%	2.37%	0.07%	51.59%	9.03%	31,550
Fresnel C with A-3	1581	4.74	88.92%	11.08%	59.58%	17.63%	17.99%	3.91%	0.90%	30.26%	24.25%	13,250
Thin film PV with VC	3912	11.88	87.71%	12.29%	83.94%	7.13%	7.27%	1.30%	0.36%	30.27%	29.41%	12,360
Multicrystalline PV with VC	2560	7.73	88.20%	11.80%	74.68%	10.89%	11.11%	2.76%	0.56%	30.25%	21.18%	16,160
Thin film PV with VC(GridCon)	3297	10.29	85.33%	14.67%	77.06%	8.89%	8.63%	5.42%	0.00%	39.15%	10.03%	18,610
Multicrystalline PV with VC(GridCon)	2801	8.43	88.53%	11.47%	73.15%	9.95%	10.15%	6.24%	0.51%	30.25%	8.57%	18,190
SEGS VI Elec with VC	1675	5.12	87.15%	12.85%	56.21%	17.49%	19.02%	7.28%	0.00%	39.15%	14.73%	12,680
SEGS VI Th with A-3	1726	5.14	89.50%	10.50%	56.69%	16.15%	18.45%	7.88%	0.83%	30.25%	11.01%	14,160
Parabolic 400 Elec with VC	2817	8.76	85.72%	14.28%	72.90%	10.40%	9.49%	7.21%	0.00%	39.14%	8.83%	14,920
Parabolic 400 Th with A-3	2790	8.39	88.64%	11.36%	72.14%	9.99%	9.58%	7.77%	0.51%	30.26%	6.91%	15,940
Tres Elec with VC	1528	4.67	87.09%	12.91%	54.95%	19.18%	14.15%	11.72%	0.00%	39.15%	10.02%	11,900
Tres Th with A-3	1624	4.84	89.48%	10.52%	55.54%	17.16%	13.31%	13.12%	0.88%	30.27%	7.03%	14,160
Tower200 Elec with VC	2295	6.79	90.09%	9.91%	84.43%	12.76%	0.00%	2.81%	0.00%	34.31%	25.57%	13,600
Tower200 Th with A-3	2735	8.08	90.19%	9.81%	87.37%	10.71%	0.00%	1.91%	0.00%	35.09%	32.51%	11,040
SEGS VI Elec with VC(GridCon)	3012	9.53	84.17%	15.83%	84.34%	9.72%	0.00%	5.94%	0.00%	39.15%	10.03%	18,610
Parabolic 400 Elec with VC(GridCon)	1356	4.27	84.60%	15.40%	69.41%	21.60%	0.00%	8.98%	0.00%	39.15%	14.73%	12,680
Tres Elec with VC(GridCon)	2550	8.04	84.45%	15.55%	80.55%	11.49%	0.00%	7.96%	0.00%	39.14%	8.83%	14,920
Tower200 Elec with VC(GridCon)	1311	4.10	85.27%	14.73%	64.01%	22.34%	0.00%	13.66%	0.00%	39.15%	10.02%	11,900

Appendix B. Detailed description of solar cooling options

System name	Solar field description	Solar system output	$T_{operation}$ (°C)	ES type	Chiller type
ETC A with AC1	Evacuated tube collectors from manufacturer (A)	Thermal	85	Hot thermal storage	Single effect absorption chiller
ETC B with AC1	Evacuated tube collectors from manufacturer (B)	Thermal	85	Hot thermal storage	Single effect absorption chiller
FPC A with AC1	Flat plate collectors from manufacturer (A)	Thermal	85	Hot thermal storage	Single effect absorption chiller
FPC B with AC1	Flat plate collectors from manufacturer (B)	Thermal	85	Hot thermal storage	Single effect absorption chiller
Fresnel A with AC3	Fresnel, linear Fresnel concentrator from manufacturer (A)	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Fresnel B with AC3	Fresnel, linear Fresnel concentrator from manufacturer (B)	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Fresnel C with AC3	Fresnel, linear Fresnel concentrator from manufacturer (C)	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Multicrystalline PV with VC	Multicrystalline photovoltaic cells	Electric	N/A	Chilled water storage	Vapor compression electric chiller
Multicrystalline PV with VC(GridCon)	Multicrystalline photovoltaic cells	Electric	N/A	Electricity grid	Vapor compression electric chiller
Parabolic 400 Elec with VC	Parabolic 400 Ele, is a parabolic trough concentrated solar power plant with an equivalent capacity of 400 MWe. Performance figures are based on NREL report [24]	Electric	>350	Hot thermal storage	Vapor compression electric chiller
Parabolic 400 Elec with VC(GridCon)	Parabolic 400 Ele, is a parabolic trough concentrated solar power plant with an equivalent capacity of 400 MWe. Performance figures are based on NREL report [24]	Electric	>350	Electricity grid	Vapor compression electric chiller
Parabolic 400 Th with AC3	Parabolic 400 Ele, is a parabolic trough concentrated solar power plant with an equivalent capacity of 400 MWe. Performance figures are based on NREL report [24]	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Parabolic A with AC2	Parabolic trough collectors from manufacturer (A)	Thermal	180	Hot thermal storage	Double effect absorption chiller
Parabolic B with AC2	Parabolic trough collectors from manufacturer (B)	Thermal	180	Hot thermal storage	Double effect absorption chiller
SEGS VI Elec with VC	SEGS VI, solar energy generating systems, is a parabolic trough concentrated solar power plant with an equivalent capacity of 30 MWe. Performance figures are based on NREL report [24]	Electric	>350	Hot thermal storage	Vapor compression electric chiller
SEGS VI Elec with VC(GridCon)	SEGS VI, Solar Energy Generating Systems, is a parabolic trough concentrated solar power plant with an equivalent capacity of 30 MWe. Performance figures are based on NREL report [24]	Electric	>350	Electricity grid	Vapor compression electric chiller
SEGS VI Th with AC3	SEGS VI, solar energy generating systems, is a parabolic trough concentrated solar power plant with an equivalent capacity of 30 MWe. Performance figures are based on NREL report [24]	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Thin film PV with VC	Photovoltaic cells based on thin film technology	Electric	N/A	Chilled water storage	Vapor compression electric chiller
Thin film PV with VC(GridCon)	Photovoltaic cells based on thin film technology	Electric	N/A	Electricity Grid	Vapor compression electric chiller

(*continued on next page*)

Appendix B (*continued*)

System name	Solar field description	Solar system output	$T_{operation}$ (°C)	ES type	Chiller type
Tower200 Elec with VC	Tower200, is a central receiver concentrated solar power plant with an equivalent capacity of 200 MWe. Performance figures are based on NREL report [24]	Electric	>350	Hot thermal storage	Vapor compression electric chiller
Tower200 Elec with VC(GridCon)	Tower200, is a central receiver concentrated solar power plant with an equivalent capacity of 200 MWe. Performance figures are based on NREL report [24]	Electric	>350	Electricity grid	Vapor compression electric chiller
Tower200 Th with AC3	Tower200, is a central receiver concentrated solar power plant with an equivalent capacity of 200 MWe. Performance figures are based on NREL report [24]	Thermal	250	Hot thermal storage	Triple effect absorption chiller
Tres Elec with VC	Tres, or Solar Tres, is a central receiver concentrated solar power plant with an equivalent capacity of 13.7 MWe. Performance figures are based on NREL report [24]	Electric	>350	Hot thermal storage	Vapor compression electric chiller
Tres Elec with VC(GridCon)	Tres, or Solar Tres, is a central receiver concentrated solar power plant with an equivalent capacity of 13.7 MWe. Performance figures are based on NREL report [24]	Electric	>350	Electricity grid	Vapor compression electric chiller
Tres Th with AC3	Tres, or solar Tres, is a central receiver concentrated solar power plant with an equivalent capacity of 13.7 MWe. Performance figures are based on NREL report [24]	Thermal	250	Hot thermal storage	Triple effect absorption chiller

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