Performance evaluation of a solar absorption air conditioning system under arid climatic conditions

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ABSTRACT

The main objective of the present work is to evaluate the performance of a solar absorption air conditioning system for a supermarket under arid climatic conditions (e.g., Hurghada city (27.26 ºN, 33.81 ºE), Egypt). To achieve this objective, TRNSYS software as a simulation tool based on the mass and energy conservation equations is used. Model validation showed that an acceptable agreement between the simulated and actual results with an overall annual error 8.4%. Reported results indicated that the annual cooling load and maximum instantaneous cooling load are 463 Wh/m^{2}and 173 W/m^2. Also, the results revealed that solar collector area of 900 m^2, storage tank volume of 25 m^3 and collector flow rate 5 kg/h.m^2 with annual solar fraction of 0.52 are required.

Key words: Solar absorption chiller, Commercial building, evacuated-tube collector

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1 Introduction

Energy consumption becomes more and more huge with the worldwide economy developments. Residential and commercial sector buildings consume about 20.1% of the total delivered energy consumed worldwide. However, total world delivered commercial sector energy consumption grows by an average of 1.6%/year from 2012 to 2040 and is the fastest-growing energy demand sector [1]. Among commercial sector buildings, supermarkets are the most energy-intensive commercial buildings. Their energy is consumed by heating and cooling systems, lights, refrigerators, computers, and other equipment in the buildings. Refrigerators, air conditioning systems and lights consume around 80% of electricity energy of supermarkets and other utilities for the remainder [2,3].

Energy costs of supermarkets are typically approximately 1% of sales [4]. Thus, reducing energy use in supermarkets would save money and would have tremendous positive impact in our environment and energy security. In addition, energy efficiency in supermarkets creates good, skilled, and needed jobs in construction and technology, such as engineers, commissioning agents, energy managers, and building operators. Also, energy efficiency can improve the bottom line, increase profits, and put the facility in a more price-competitive position. With growing concerns over global warming and other environmental issues, many supermarket owners want to demonstrate to customers that they are responsible environmental stewards.

The main problem facing the owners of commercial buildings is how to reach the highest level of comfort conditions at the lowest possible energy cost. One of the most expensive parameters to reach a comfortable environment within commercial buildings is to be maintain a high level of indoor air quality due to consume a high amount of high grade energy as a result of using a conventional air conditioning systems, which are working with refrigerants that have side effect on ozone layer.

In the resent work, improving the energy efficiency of a supermarket involves upgrades to air conditioning systems. These upgrades can attract and retain more customers, leading to an increase in sales. Therefore, the attention of many owners of commercial building is to use alternate air conditioning systems using environmentally friendly working fluids to reduce both energy consumption and emissions to achieve the highest possible profitability. By 2030, hydro, wind, solar PV, bioenergy, geothermal, concentrating solar and marine power between them provide nearly 40% of electricity supply [5]. Thus, solar absorption air conditioning system becomes an interesting alternative to be used due to the tremendous solar potential in account that the thermal cooling load increases when the solar radiation is maximum specially in arid (hot and humid) climatic conditions.

Various investigations are reviewed to find the status of the solar cooling systems performance evaluation. [6–8] conducted comprehensive reviews on the solar cooling systems and technologies. The reviews focused on the conducted research related the solar thermal system and /or associated cooling system technologies. The annual performance of a solar cooling system with the air-cooled chiller was investigated for residential cooling application [9]. They reported that the investigated absorption chiller can meet about 65% of the total cooling load of the building with an average COP_{th} of about 0.61. Furthermore, 28% of the titled solar radiation was converted into cooling capacity by the solar air conditioning system.

The solar driving variable effect LiBr-water absorption cooling system was theoretically investigated [10]. The daily performance of this system was calculated and analyzed. Variable effect chiller had high COP under high driving temperature, ensured a competitive overall efficiency. Their results showed that average chiller COP of 0.88 and solar COP of 0.35 were obtained. The effects of solar collector area, storage tank volume and cut-off driving temperature on the system performance were analyzed. The optimal solar collector area and tank volume were obtained.

Both experiments on a hybrid solar system integrating a concentrator and computer simulations of a hybrid solar cooling system incorporating flat plate collectors and a concentrator were carried out by [11]. The dynamic simulation of the entire system model was performed using TRNSYS software. The system was investigated for a small-scale user, consisting of a single-floor residential house. Different time bases (day, week year), concentrator and flat plate collector areas, climatic conditions and energy tariffs were considered to investigate the energy and economic performance of the system. The main results were 70% concentrator efficiency, 50% Primary Energy Saving, 1.6 m2 /kW the optimal concentrator area per absorption chiller capacity and Simple Payback period between.

A techno-economic analysis and performance comparison of single-, double- and triple-effect absorption chillers driven by parabolic trough collectors and waste incineration plants for trigeneration of cold, heat and power in a Danish study was presented in [12]. Regardless of the chiller type, the use of parabolic trough collectors solved the summer supply problem. Moreover, double- and triple-effect systems led to a reduction of 45% and 50% in the price of the hybrid system in comparison with the single-effect design. Also, the hybrid system integrated with triple-effect showed the best performance by decreasing the required flow rate of municipal solid waste, and the number of required collectors, and consequently, decreasing the costs.

The feasibility of an absorption cooling system in a hot and dry region was carried out using the climate conditions of Duhok city located in the North of Iraq by using the TRNSYS17 programs [13]. The study outcomes were the capacities of the solar absorption cooling system components; i.e. the single effect Li-H2O absorption chiller of 35.17 kW cooling capacity, the evacuated tube solar collector of 95 m^2 area and the thermal storage tank of 3 m^3 to serve a floor space of 270 m^2.

The evacuated tube collector performance and explored its thermal efficiency in Kuwait representing a hot and severe climate experimentally was investigated [14]. Additionally, the simulated heat generated from the evacuated tube collector was used for solar cooling of a house providing 80% of air-conditioning demand.

A H2O-LiBr absorption cooling system driven with solar energy, and liquefied petroleum gas (LPG), to air condition low income houses in coastal areas of Mexico was simulated in [15]. The results indicated that dependency on a solar field of evacuated tube collectors with an area ranging from 207 to 220 m^2 according to the climatic region concerned (warm and dry vs. warm and humid climate), 60% of the power required by the system can be achieved from solar thermal energy.

The above review revealed that various investigations were carried out on solar absorption air conditioning systems under certain climatic conditions of various cities over the world to save energy and protect environment. Thus, the objective of the present work is to predict the performance of a solar absorption cooling system for a supermarket under arid climatic conditions (e.g., Hurghada city (27.26 ºN, 33.81 ºE), Egypt).

2 Building and System Description

The proposed system consists of three sub-systems: a commercial building, absorption air conditioning system and solar field, as shown in Fig. 1. The specific objectives of the present work are:

- to simulate the commercial building using TRNBUILD program to estimate the annual cooling load under climatic conditions of Hurghada (27.26 ºN, 33.81 ºE).
- to simulate the performance of solar assisted absorption air conditioning system under climatic conditions of Hurghada using TRNSYS simulation studio.
- to predict the thermal performance of proposed system using both TRNBUILD and TRNSYS simulation studio.

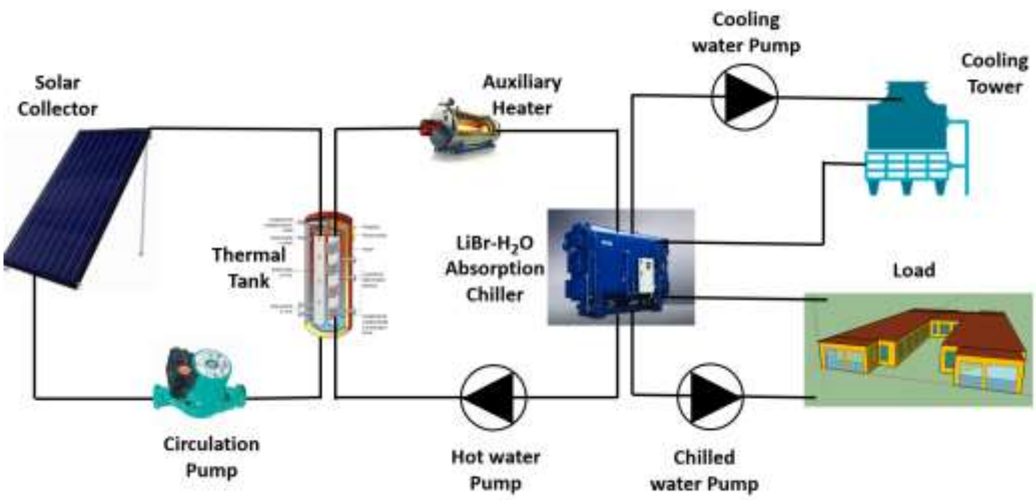

Fig. 1 Schematic diagram of a solar driven air-conditioning system.

2.1 Building Description

The considered supermarket is located at Hurghada (27.26 ºN, 33.81 ºE). It is one story supermarket with floor area and height of 1021 m^2 and 3.1m, respectively as illustrated in Fig. 2.

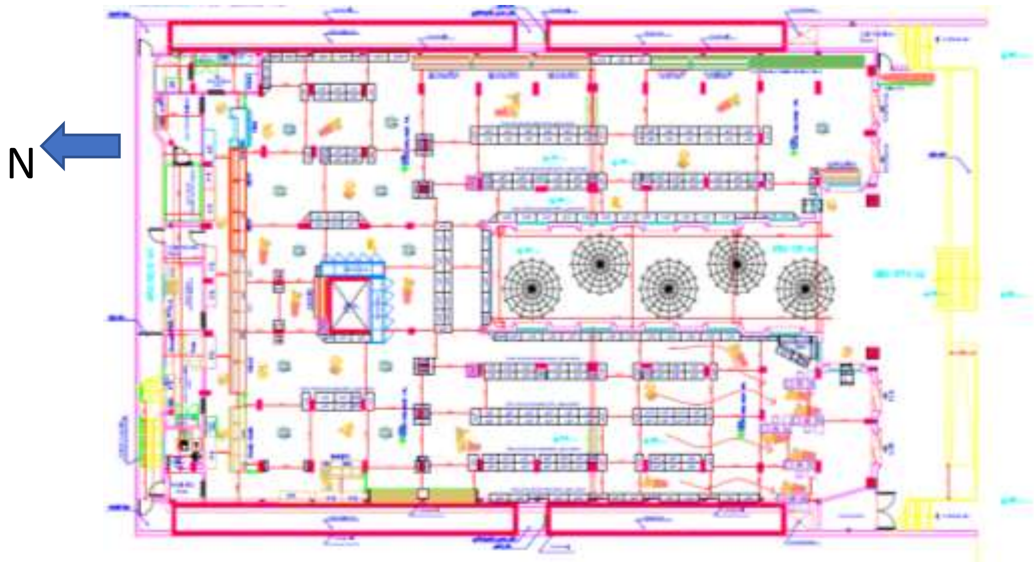

Fig. 2 The supermarket plan.

The supermarket has U-shape with one entrance gate and one exit gate. Figure 3 shows the supermarket 3D-model, which is drawn in sketch-up program. The supermarket is the ground floor of multi-floor air-conditioned building. Therefore, the ceiling floor is not exposed to the sun and considered as adiabatic boundary next to air-conditioned space. The external wall has overall thermal transmittance (U-value) of 2.48 W/m^{2}K. The shading effect of neighbor buildings is taken into consideration. The gates are four glazing doors (two at entrance and with thermal transmission (U-value) of 2.89 W/m^{2}K and solar gain coefficient (g-value) of 0.789%. The doors are automatically opened and closed using occupancy sensors. The infiltration rates due to doors opening and cracks are taken into consideration with average air change of 3.0 Air Change Per hour (ACH).

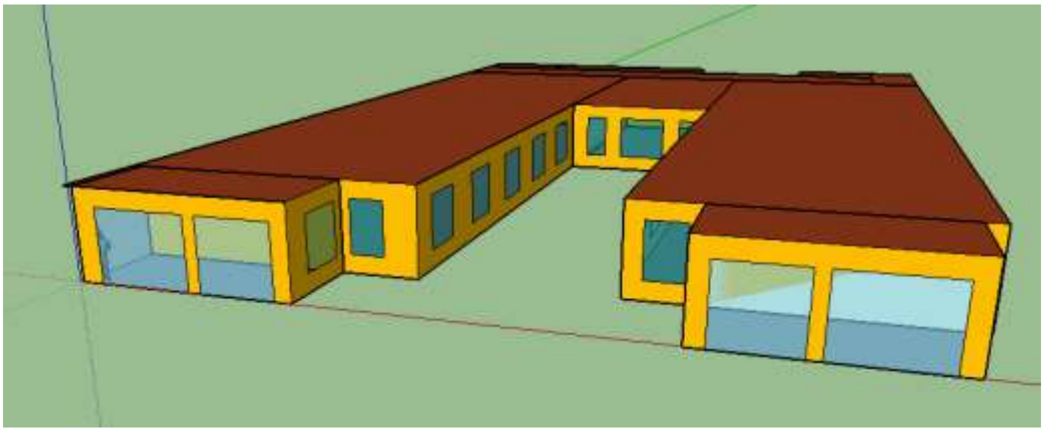

Fig. 3 Supermarket 3D-Model.

The gains from each equipment are calculated using the following equations based on ASHRAE Fundamental (2017) [16] using the following equations.

$$Q_{R_{gain}} = N * P_{rated} * F_R * F_u \quad \text{Eq.(1)}$$

$$Q_{S_{gain}} = N * P_{rated} * F_S * F_u \quad \text{Eq.(2)}$$

$$Q_{L_{gain}} = N \ * P_{rated} * F_L \ * F_u \qquad \text{Eq.(3)}$$

$$P_{ele} = N \ * P_{rated} * F_u \qquad \text{Eq.(4)}$$

Where $Q_{R_{gain}}, Q_{S_{gain}}, Q_{L_{gain}}, P_{rated} and\ P_{ele}$ are the radiant heat gain, the sensible convention heat gain, the latent heat gain, the rated power and the electric power drawn by each piece of equipment, respectively. N, F_R, F_S, F_L, F_u are the number of same pieces of equipment, the radiant factor, the sensible factor, the latent factor, and the usage factor, respectively. All pieces' schedule is always ON with usage factor of 25% over the hour (i.e. actual operation with full capacity is 15 min per hour) except last four pieces of equipment of the bakery section. The capacities of the existing pieces of equipment are listed in Table 1.The daily operation schedule of bakery section is between 3:00 – 7:00 AM while it keeps OFF at remaining hours. The only item has latent load is Bakery Display Cabine with latent ratio of 0.05.

The occupancy schedule of the supermarket was obtained from stored hourly database of the customer along a year of 2016. The schedule is adjusted to be a fraction between 0 and 1 representing the ratio of the number of customers at any hour to the maximum customer number of 90 clients per hour. The occupancy schedule varies from season to season and workday to weekends (Fridays and Saturdays) as shown in Fig. 3.

As per the installed lighting in the supermarket, the maximum lighting intensity is 34 W/m2. During the day, the lighting is adjusted to be always ON for 18 hours between 7:00 AM to 1:00 AM of the next day, while it works with half capacity during the remaining 6 hours between 1:00 AM to7:00 AM.

Table 1 The capacities and heat gains from the existing Equipment.

Equip. Name	kW	Equip. No	Sensible Radiant Ratio	Sensible Convective Ratio
Commercial Multideck Refrigerator	2.2	1	0.06	0.19
Air Curtain	0.2	4	0.60	0.00
Commercial Multideck Refrigerator AC	0.5	10	0.06	0.19
Stainless Steel Refrigerator	0.75	1	0.30	0.15
Under Counter Refrigerator	2.2	3	0.30	0.15
Meat Mincer	1.87	1	1.00	0.00
Slice Machine	0.7	3	1.00	0.00
Stretch Packing Machine	0.175	3	0.41	0.14
Chicken Electric Grill	3	1	0.29	0.00
Bakery Display Cabine	0.2	1	0.09	0.00
Hot Meal Cabine	2.2	2	0.10	0.41
Mix Display Frozen Refrigerator 6D	5	1	0.19	0.22
Mix Display Frozen Refrigerator 5D	4.5	1	0.19	0.22
Casheir Machine	0.2	5	0.60	0.00
Air Curtain B	0.2	4	0.60	0.00
Stainless Steel Vertical Refrigerator	2.3	2	0.30	0.15
Ice Maker	1.5	1	2.50	0.00
Fish Griddle	2	1	0.04	0.00
Frayer	4	1	0.02	0.00

Ventilation blower A	2.2	1	0.60	0.00
Ventilation blower B	0.1	2	0.60	0.00
Water Electric Heater	3	1	0.38	0.12
Ventilation blower C	0.1	2	0.60	0.00
Deck Pizza Oven	15	1	0.03	0.00
Ferment Cabine (Steam Cabine)	1.5	1	0.01	0.00
Electric Rings	12	1	0.27	0.00
Ventilation blower D	2.2	1	0.60	0.00
Ventilation blower E	0.1	1	0.60	0.00

Fig. 3.a Occupancy schedule of 1st quarter.

2.2 Existing Air-conditioning System

The existing air-conditioning system is a vapor compression type. The system consists of 5 units of 20 TR (70 kW_{th}) each. The standard coefficient of performance (COP) based on the catalogue is 2.2. Moreover, the thermostats are being set at 22°C. Regarding the electricity consumption, it is recorded by the authority of electricity and the invoice released monthly. These invoices were used to validate the simulation model consisting of building model served by conventional air-conditioning vapor compression system.

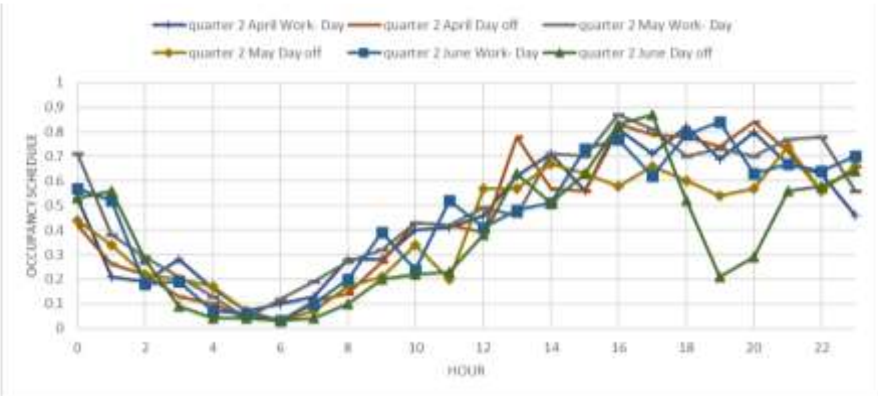

Fig. 3.b Occupancy schedule of 2nd quarter.

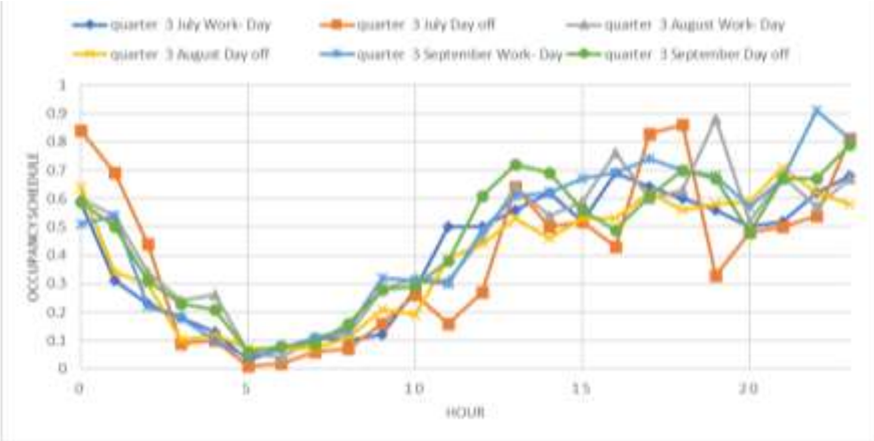

Fig. 3.c Occupancy schedule of 3rd quarter.

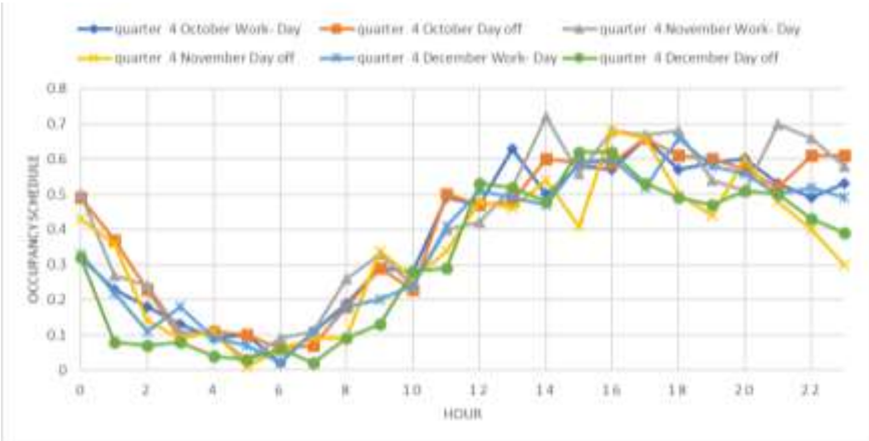

Fig. 3.d Occupancy schedule of 4th quarter.

2.3 Alternative Air-conditioning System Description

The alternative air-conditioning system is a solar absorption air conditioning system. For the absorption chiller thermal analysis, the following assumptions are made; (a) no pressure drop occurred through the chiller heat exchangers, i.e. pressure changes only through the pump and expansion devices, (b) heat losses or heat gain from or to the chiller are ignored, (c) the absorbent (lithium bromide) is non-volatile in the temperature ranges considered, (d) conditions of the refrigerant (water) at the exits of the condenser and evaporator is saturated.

Fig. 4 shows the TRNSYS simulation studio environment project that was built to predict the performance of the proposed system. The solar and absorption refrigeration subsystems is linked with cooling load that is modeled by TRNBUILD using Type 56 in order to meet the cooling load requirement of the building [17].

The single effect absorption chiller (Type 107) uses an external absorption chiller performance data file for the chiller performance modeling at the prevailing conditions of hot water supply, cooling water, and chilled water inlet temperature [18]. This model requires knowledge of the effect of chilled water inlet temperature cooling water inlet temperature hot water inlet temperature and part load ratio on the performance of the absorption chiller.

The absorption chiller is a Yazaki absorption chiller, model WFC – SC50, with a rated cooling capacity 175.8 kW when producing chilled water outlet temperature 7°C. The chiller unit is connected to water supply from the hot water storage, chilled water storage and wet cooling tower. Type 71 is used to model the evacuated tube collector [19]. The solar system components beside the collector are differential controller, thermal stratified storage tank and circulating pump. Table 2 lists the technical specifications of Yazaki WFC – SC50 absorption machine.

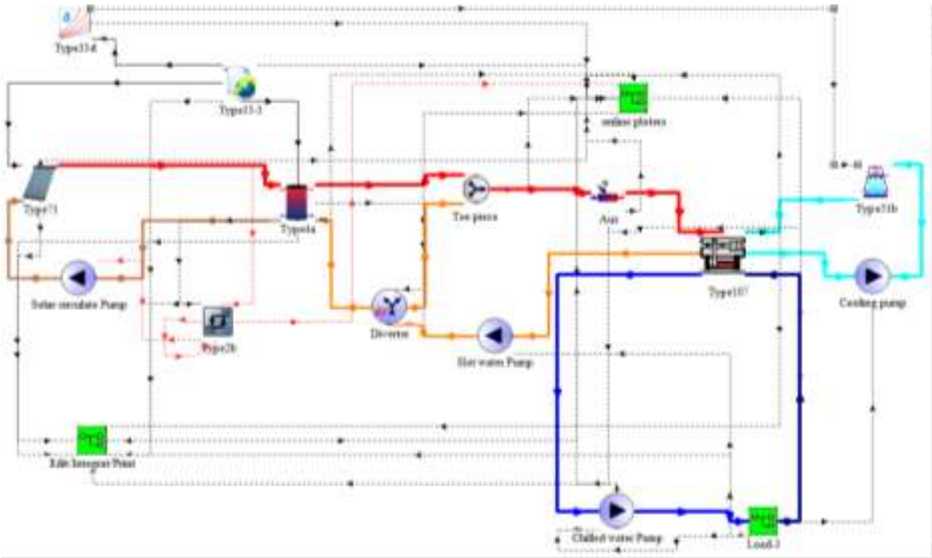

Fig. 4 TRNSYS model of the alternative air-conditioning system.

3 Results and Discussion

The building cooling load is simulated using TRNBUILD software and validated with actual consumption in the building to select the proper absorption machine that removes this load. Then, energetic analysis of solar assisted absorption refrigeration machine is carried out using TRANSYS software to optimize the solar collector surface area and storage tank volume. Finally, a detailed performance evaluation of the overall system is conducted under arid climatic conditions (e.g., Hurghada city which represent the Egyptian coastal cities). In the following sections, the results obtained for the investigated parameters are discussed.

Table 2 Technical specifications of absorption machine.

Rated cooling capacity		175.8 kW
Rated COP		0.7
Hot water	Rated Inlet temperature	88°C
	Rated Outlet temperature	83°C
	Rated flow rate	12 l/s
	Max. operating pressure	588 kPa
Cooling water	Rated Inlet temperature	31°C
	Rated Outlet temperature	35°C
	Rated flow rate	25.5 l/s

	Max. operating pressure	588 kPa
	Heat rejection	427 kW
Chilled water	Rated Inlet temperature	12.5°C
	Rated Outlet temperature	7°C
	Rated flow rate	7.64 l/s
	Max. operating pressure	588 kPa

Model validation is based on actual electricity consumption for equipment, light and existing cooling system as shown in Fig. 5, which confirms that an acceptable agreement between the simulated and actual results with an average error 8.4%. Variation of hourly ambient air dry-bulb temperature in Hurghada city is shown in Fig. 6. The dry-bulb temperature increases with local solar time until its maximum value of 40.5^{o}C in summer and then decreases. Fig. 7 shows the variation of hourly ambient air relative humidity in Hurghada city. It is clear from this figure that, ambient air relative humidity decreases with local solar time to its minimum value in summer and then increases.

Variation of total solar radiation with time in Hurghada city is shown in Fig. 8. The total solar radiation increases with time until its maximum value of 1000 W/m^2 in summer and then decreases. The annual supermarket cooling load is 463 Wh/m^2 where its hourly values are shown in Fig. 9. It is can be noted that, the building cooling load increases reaching the maximum value of 176 kW (see Fig. 10) in summer and then decreases. This is due to increase of the ambient dry bulb temperature and solar radiation during summer season. Also, it can be seen values of cooling load in winter despite the low temperatures this is a result of the presence of lighting, equipment and occupancy gains that lasts throughout the year.

Fig. 11 presents the effect of solar collector area in annual useful collected energy, auxiliary energy, collector efficiency and solar fraction. It is evident from the figure that the annual collected energy by the solar collector increases with collector area and the auxiliary heater energy decreases until they reach stability after 900 m^2 of area collector. This is due to decreasing the collector efficiency with collector are. Then, due to increase the collected energy the solar fraction increases with solar collector area until it reaches stability after the value 900 m^2 of area collector.

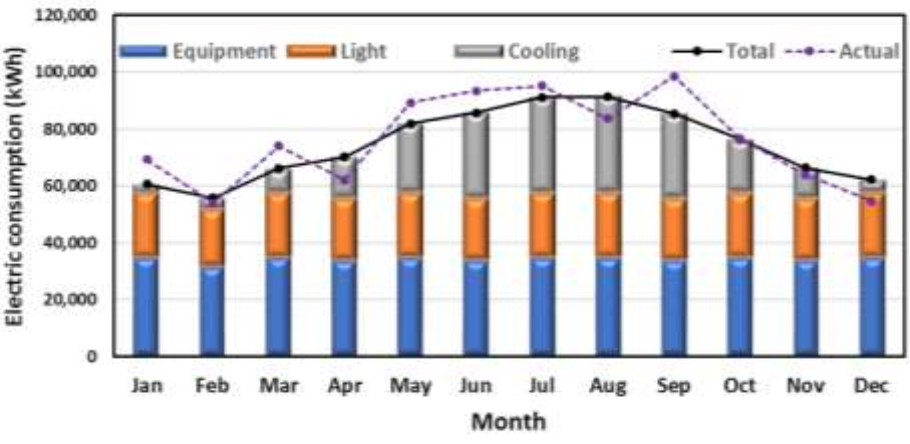

Fig. 5 Monthly Electricity consumption of the supermarket served by conventional vapor compression system.

The variation of annual useful collected energy, auxiliary energy, collector efficiency and solar fraction with a specific solar collector mass flow rate is shown in Fig. 12. The increase in specific solar collector mass flow rate leads to an increase in solar fraction. The solar fraction increasing to the maximum value at the collector mass flow rate 5 kg/h.m^2. After 5 kg/h.m^2 the collector mass flow rate is ineffective in solar fraction due to decreasing the exit temperature from the collector which leads to decrease the temperature of storage tank temperature. The decreasing in storage tank temperature required an auxiliary heating to reach the required temperature for the generator.

Effect of storage tank volume in annual useful collected energy, auxiliary energy, collector efficiency and solar fraction is shown in Fig. 13. It is clearly that, the variation of annual useful collected energy, auxiliary energy, collector efficiency and solar fraction is small with storage tank volume to 25m^3 and the stability of all energy rates and solar fraction is reached.

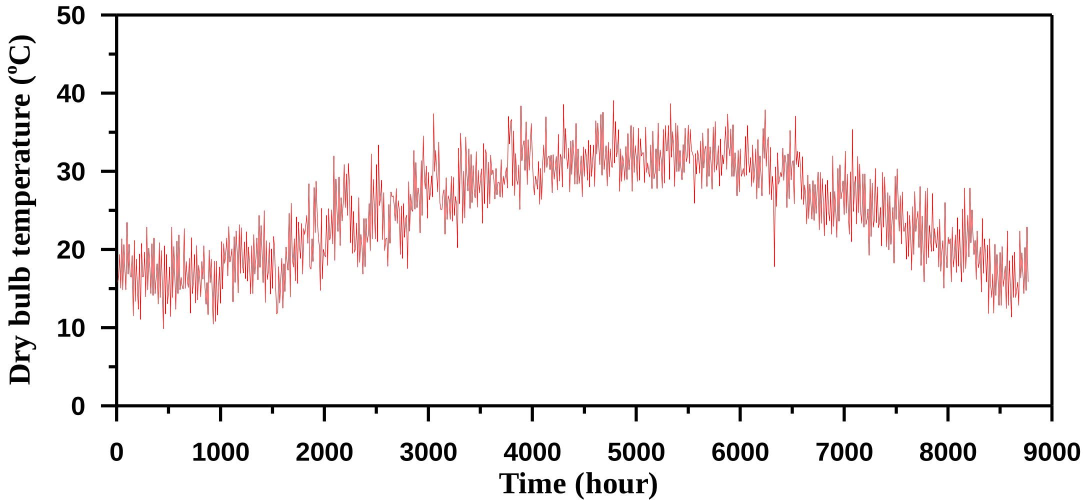

Fig. 6 Hourly dry-bulb temperature of Hurghada.

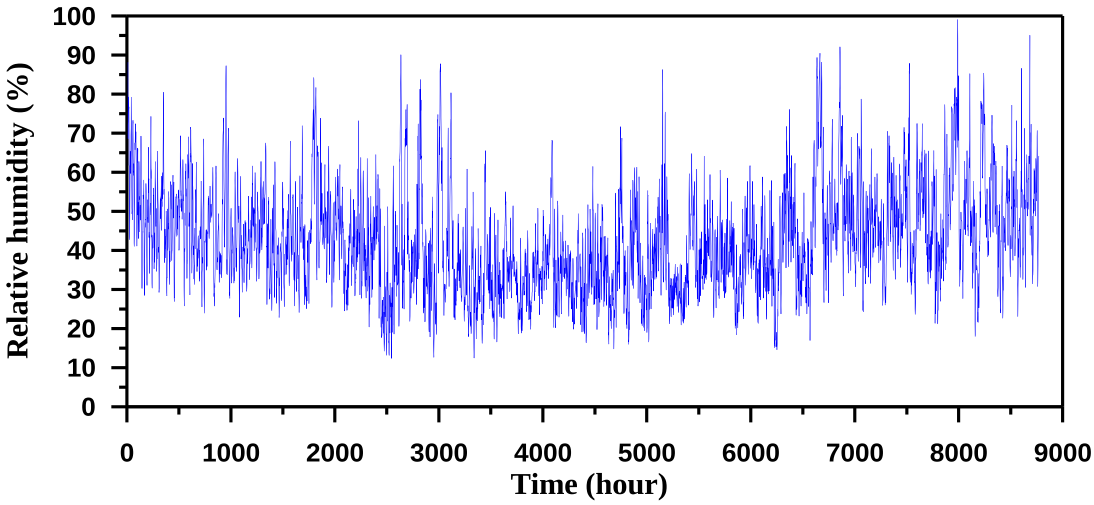

Fig. 7 Hourly relative humidity of Hurghada.

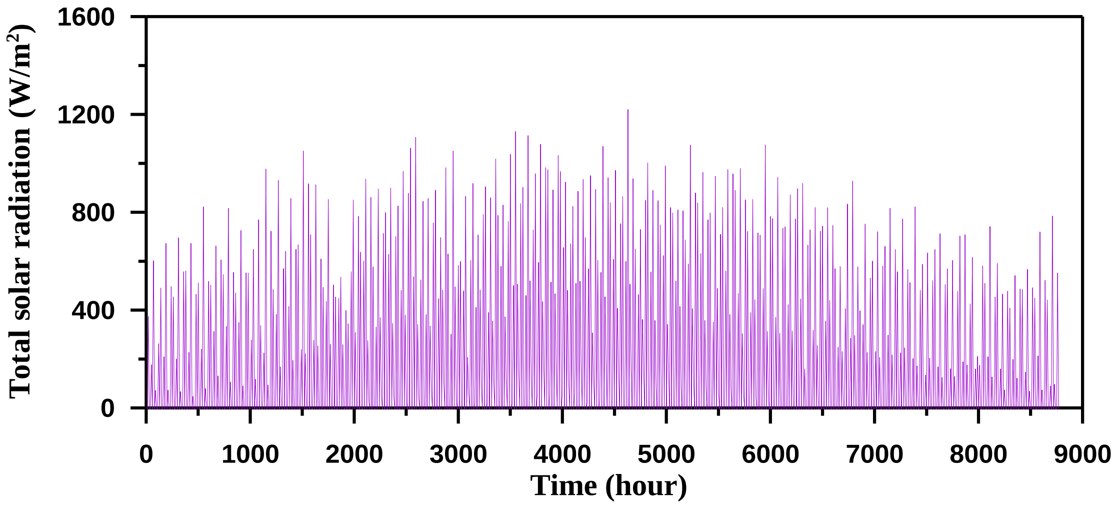

Fig. 8 Hourly total solar radiation of Hurghada.

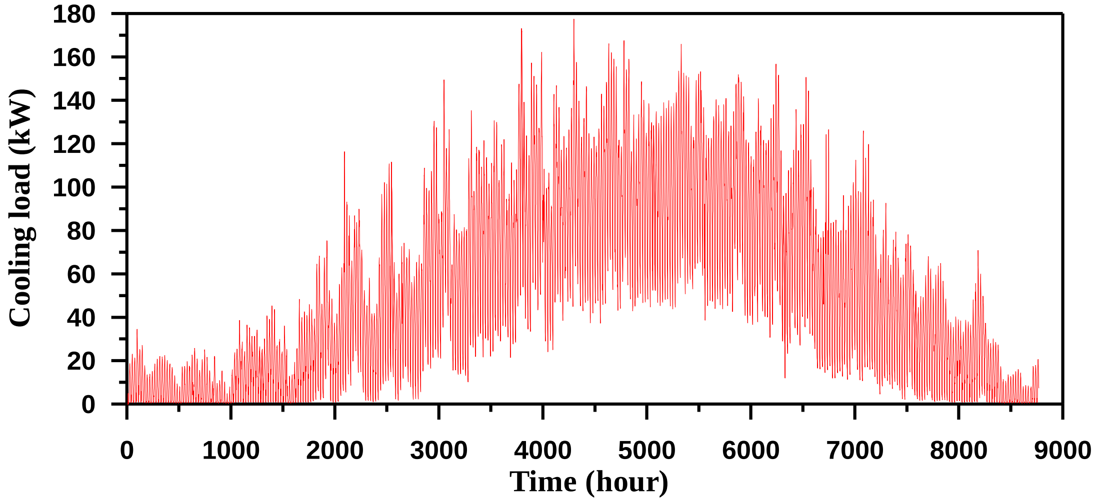

Fig. 9 Supermarket annual cooling load.

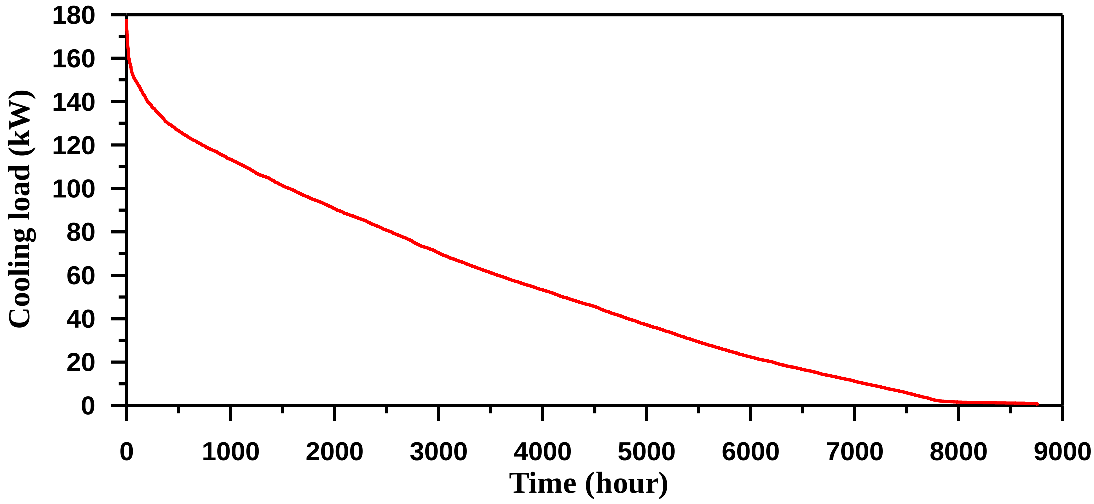

Fig. 10 Duration curve of the supermarket annual cooling load.

Variation of monthly performance of the proposed solar cooling system at the optimal collector area, collector flow rate and storage tank volume are illustrated in

Fig. 14 and Fig. 15. The variation of chilled water energy, cooling water energy and generator hot water energy with month is shown in Fig. 14. The chilled water energy, cooling water energy and generator hot water energy increases with month to it maximum value in summer months and the decreases. This due to increase in cooling load during summer seasons as shown in Fig. 9.

Fig. 15 shows the variation of monthly useful collected energy, auxiliary heater energy and solar fraction. it is can be seen from the figure that, the required energy rate for generator increases with month which leads to an increase in auxiliary heater energy in winter due to the low value of useful collected energy in this season. This leads to decreasing the value of solar fraction during winter and then increases during summer season due to increasing the useful collected energy and decreasing the auxiliary heater energy.

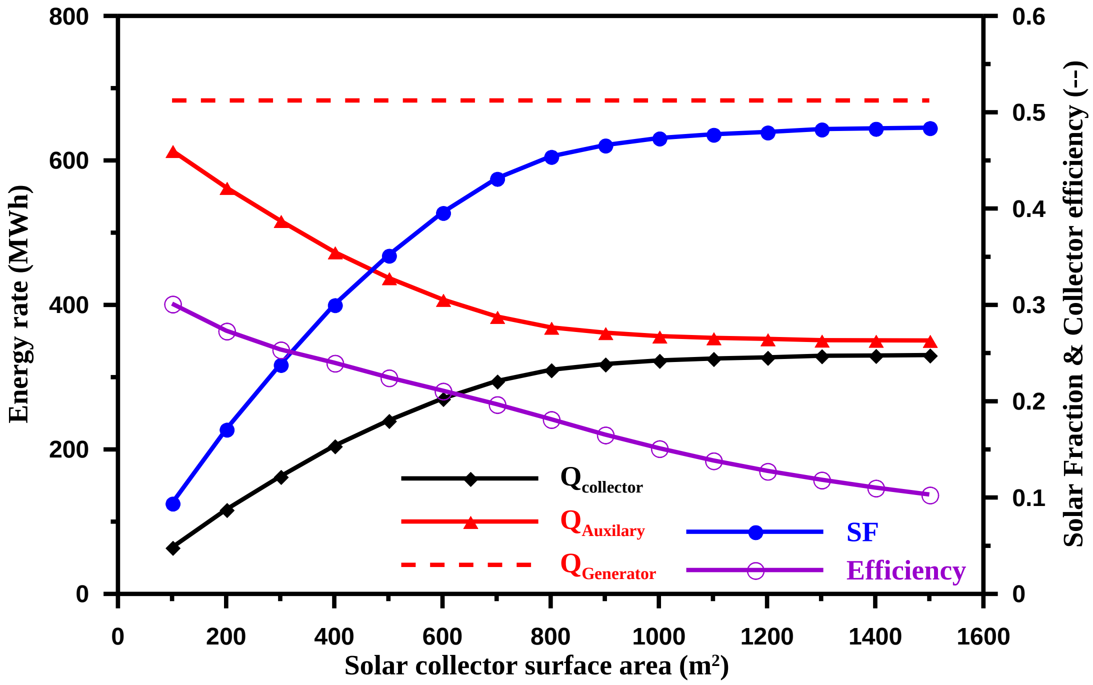

Fig. 11 Effect of solar collector area in energy rate, solar fraction and collector efficiency.

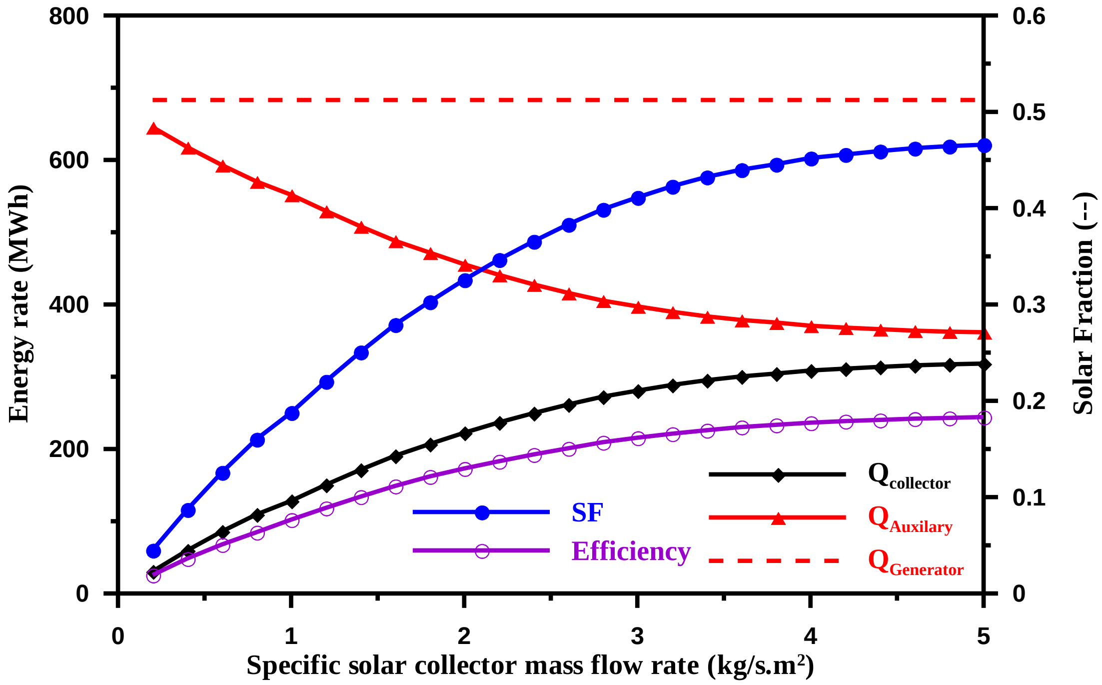

Fig. 12 Variation of energy rate, solar fraction, and collector efficiency with solar collector flow rate.

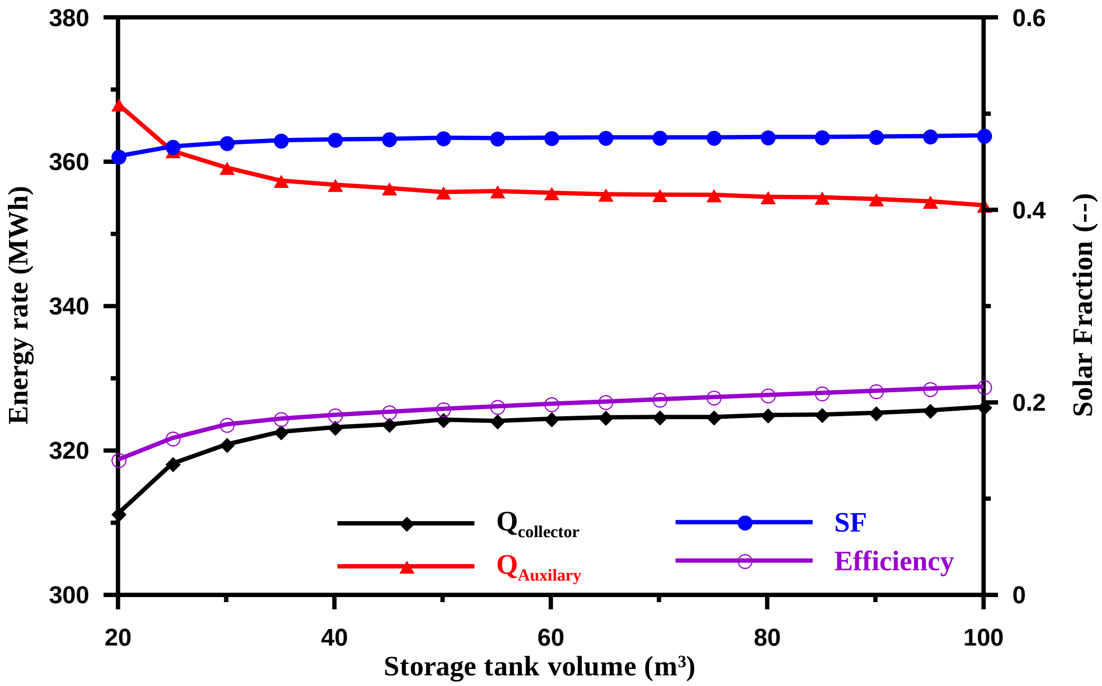

Fig. 13 Energy rate, collector efficiency and solar fraction verses storage tank volume.

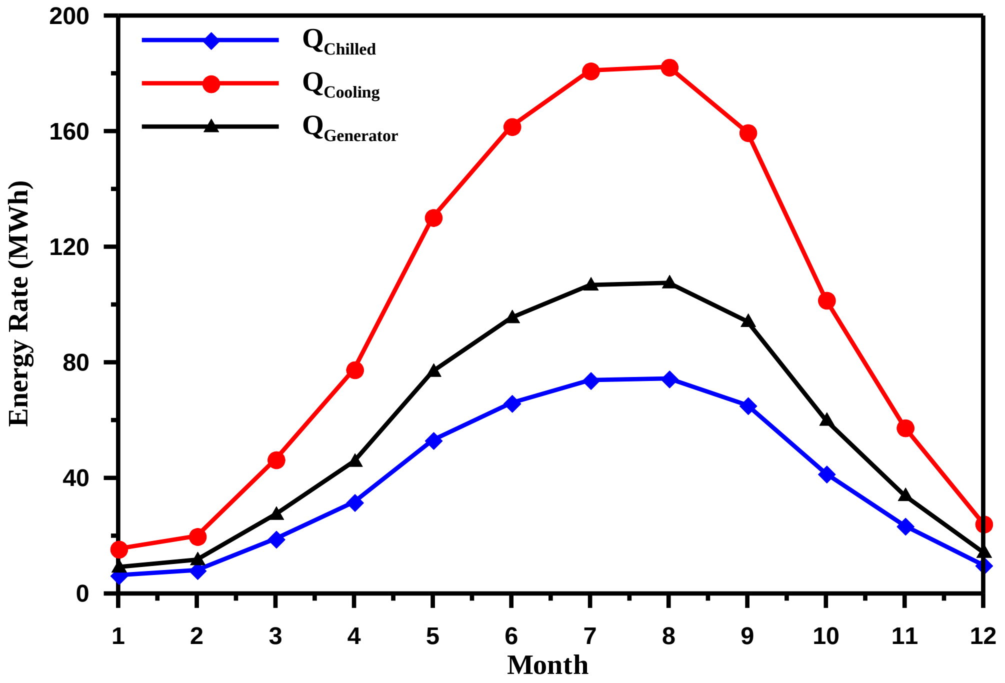

Fig. 14 Monthly chilled, cooling and generator energy rates for proposed solar cooling system.

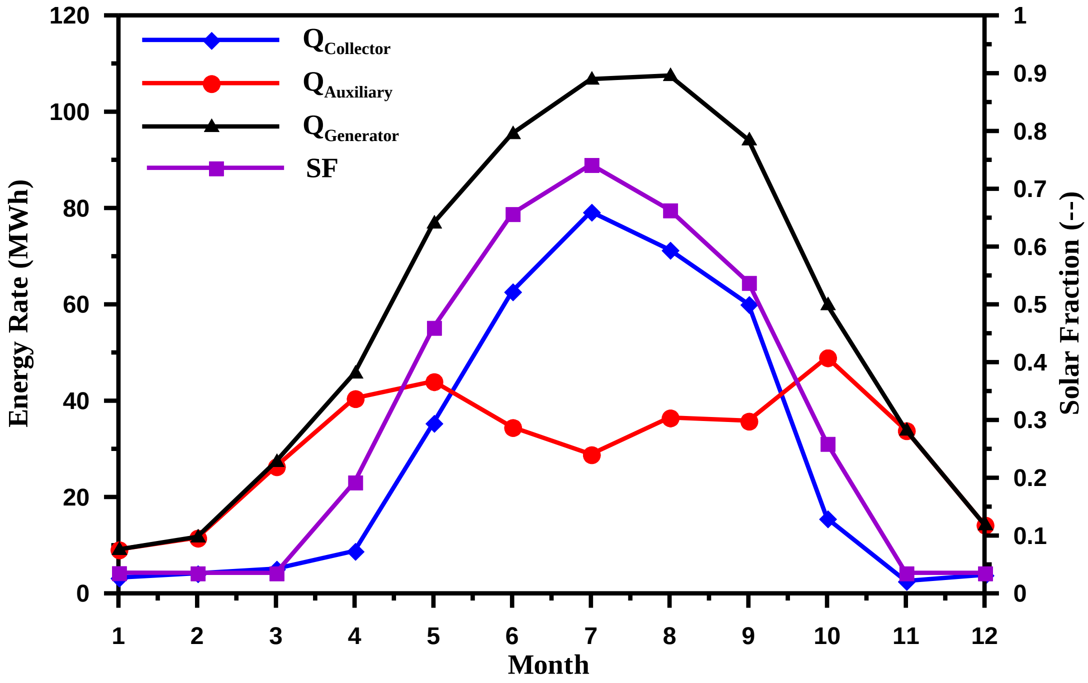

Fig. 15 Monthly chilled, cooling, generator energy rates and solar fraction for proposed solar cooling system.

4 Conclusions

The present investigation aimed to predict a thermal performance of solar air conditioning system for supermarket under arid climatic conditions (Hurghada, Egypt). The estimated load using TRNBUILD showed an acceptable agreement with actual load with an average error 8.4%. The results indicated that the annual cooling load and maximum instantaneous cooling load are 463 Wh/$m^2$and 173 W/m^2. Based on the theoretical investigations throughout the present work, the evacuated tube solar collector area of 900m^2, tank volume of 25m^3, collector flow rate of 5 kg/h.m^2 and 180 kW auxiliary heater can achieve 0.52 annual solar fraction.

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