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Simulation of Solar Lithium Bromide–Water Absorption Cooling System With Double Glazed Flat Plate Collector for Adrar

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Abstract

Adrar is a city in the Sahara desert, in southern Algeria known for its hot and dry climate, where a huge amount of energy is used for air conditioning. The aim of this research is to simulate a single effect lithium bromide–water absorption chiller coupled to a double glazed flat plate collector to supply the cooling loads for a house of 200m² in Adrar. The thermal energy is stored in an insulated thermal storage tank. The system was designed to cover a cooling load of 10.39KW for design day of July. Thermodynamic model was established to simulate the absorption cycle. The results have shown that the collector mass flow rate has a negligible effect on the minimum required collector area, but it has a significant effect on the optimum capacity of the storage tank. The minimum required collector area was about 65.3 m², which could supply the cooling loads for the sunshine hours of the design day for July. The operation of the system has also been considered after sunset by saving solar energy.

Keywords: solar, cooling, absorption, double glazed flat plate, bromide-water, energy, cooling, Adrar;

1. Introduction

The use of absorption cooling systems is becoming increasingly important, with the environmental problems caused by a mechanical compression systems (use of CFC and HCFC), and the soaring energy prices; absorption solar systems seems to be an alternative to conventional cooling systems, notably with their possibility to use renewable energy and reduce fuel consumption. Solar cooling systems consist of a solar collector and a storage system for absorption. The main components of an absorption chiller include four heat exchangers; generator, condenser, evaporator and absorber. The LiBr–H₂O system operates at a generator temperature in the range of 63–95°C with water used as a coolant in the absorber and condenser. The COP of this system is between 0.6 and 0.8, which is higher than that for NH₃–H₂O systems [1]. Tsilingiris [2] presented a theoretical microcomputer model to investigate the operational behavior of a simple design solar LiBr–H₂O absorption cooling system with 7 kW cooling load capacity for small residential applications in Greece. Ghaddar et al. [3] modelled and simulated a solar absorption system for a typical house in Beirut. The absorption cycle was simulated by a thermodynamic model, and the storage tank was assumed to be well mixed. Hourly values of the direct and diffuse components of solar radiation incident on the collectors and the values of ambient temperature, wind speed and direction were derived directly from actual hourly measured weather data files.

Nomenclature

A _c	collector surface area (m ²)	W	work (KW)
COP	coefficient of performance	X	concentration of LiBr–water solution (%)
C _p	specific heat capacity of fluid (kJ/kg °C)	Greek symbols	
dt	time step	η	efficiency
FR	flow ratio	(τ×α)	the transmittance–absorptance products
F _R	heat removal factor	Subscripts	
h	enthalpy (KJ/Kg)	a	absorber , ambient
I _g	instantaneous solar radiation incident (kW m ⁻²)	c	condenser
m	mass flow rate (Kg/s)	d	diluted
Q	heat transfer rate (KW)	f	refrigerant
T	temperature (°C)	Fi	inlet fluid (°C)
t	solar time (h)	g	generator

U_L	overall heat loss coefficient ($W/m^2 \text{ } ^\circ C$)	P	pump
		u	useful

The cooling power at the evaporator was modeled as a variable load with an average capacity of 10.5 kW. The delivery water temperature to the generator was above 65 °C and the temperature in the storage tank was not allowed to exceed 95 °C to prevent change of phase occurrence. Florides et al. [4] modeled a complete system that was comprised of a solar collector, a storage tank, a boiler for auxiliary energy and a LiBr–water absorption system, the system was modeled with the TRANSYS simulation program and the typical meteorological data file of Nicosia, Cyprus. Assilzadeh et al. [5] simulated and optimized a solar LiBr–water absorption cooling system with an evacuated tube collector by the TRANSYS program for Malaysia’s climate. They proposed that, to achieve continuous operation and increase the reliability of the system, a 0.8 m³ hot water storage tank is essential, and the system needs a collector area of 35 m² sloped at 20°C for a 3.5 kW cooling load. Adrar is one of the hottest cities in Algeria in the latitude of 27.82° N and longitude of 0.18°W. In this region an enormous amounts of energy is used for cooling residential places in a year, which is costly and harmful to the environment. The daily average of global solar radiation of Adrar is about 2.456 MJ for the summer months, and its ambient temperature sometimes reaches above 45 °C; therefore, solar absorption cooling chillers are suitable in this area, so the solar cooling system has been simulated with typical weather data of solar radiation, ambient temperatures of Adrar. The given solar cooling system consists of a double glazed flat plate collector, an insulated hot water storage tank and a single effect lithium bromide–water absorption chiller. For this research, a typical house (area about 200 m²) has been selected, which has its maximum cooling load of 10.39 kW for July. A computer program has been written in FORTRAN to simulate and design the solar cooling system to supply the cooling demand of this house from when the cooling loads are more considerable. Although stationary collectors, like the flat plate and evacuated tube, have been the most conventional collectors used in simulation of solar absorption systems [2–6]. A thermodynamic model is suitable to simulate the absorption cycle, but the basic assumptions and input parameters must be considered [3,7,8,9]; so, some typical and applied input values, such as the solution pump flow rate, the efficiency of solution heat exchanger and variable temperatures of the generator and condenser, have been used to model the absorption cycle. The cooling load of residential places is fairly high even after sunset that is why the operation of the solar system not only has been simulated during sunshine hours but also after sunset by storage of solar energy as well. The minimum required collector area and optimum storage tank capacity have been obtained for various water mass flow rates of the collector.

2. System description

The water heated by the solar collector is stored in the storage tank and pumped to the generator to separate the refrigerant from the absorbent, then the superheated refrigerant is condensed in the condenser by the cold water of the cooling tower, water follows through the expansion valve and arrived to the evaporator to produce cold required. The vapor is lead to the absorber where it is absorbed by the rich solution coming from the generator; finally the rich solution is pumped to the generator via a heat exchanger. This cycle operates at low pressure to evaporate water at low temperatures. Fig. 1 shows the schematic diagram of the solar lithium bromide–water absorption cooling system.

2.1. Absorption cycle

The performance of an absorption cycle can be simulated well by a thermodynamic model [3, 7, 8, 9]. The basic assumptions model of the absorption cycle is as follows:

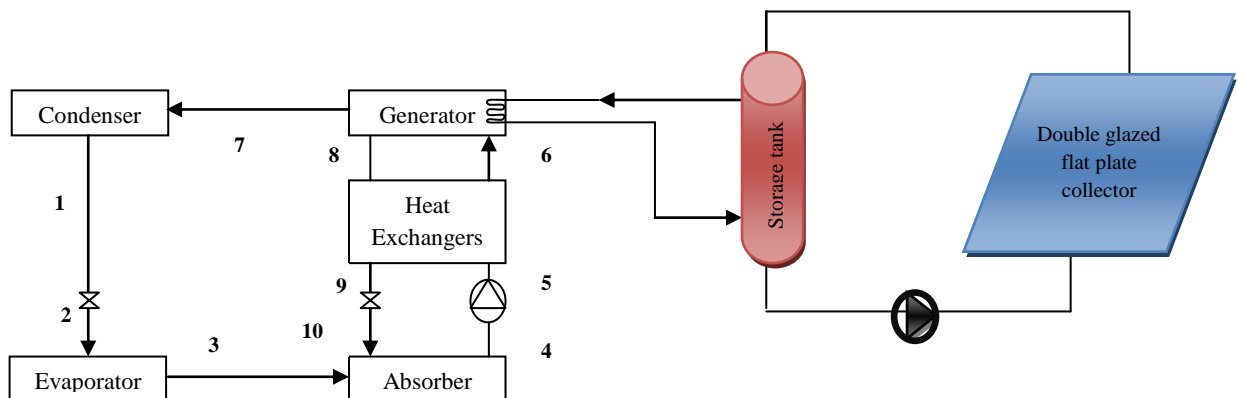


Fig. 1, Schematic diagram of solar absorption cooling system

- There is saturated refrigerant at the condenser and evaporator outlets.
- There is no departure of chemical substances from the cycle to the environment.
- The kinetic and potential energy effects are neglected.
- The refrigerant (water) at the outlet of the condenser is saturated liquid.
- The refrigerant (water) at the outlet of the evaporator is saturated vapour.
- The Lithium bromide solution at the absorber outlet is a strong solution and it is at the absorber temperature.
- The outlet temperatures from the absorber and from generators correspond to equilibrium conditions of the mixing and separation respectively.
- Pressure losses in the pipelines and all heat exchangers are negligible.
- Heat exchange between the system and surroundings, other than in that prescribed by heat transfer at the generator, evaporator, condenser and absorber, does not occur.
- The system produce chilled water, and generator is driven by hot water.

At the absorber, two mass balances can be made:

$$m_f + m_g - m_a = 0 \quad (1)$$

$$m_g X_c - m_a X_d = 0 \quad (2)$$

We derive an expression m_g and m_a as a function of m_f and refrigerant concentrations.

$$m_a = m_f \frac{X_c}{X_c - X_d} \quad (3)$$

$$m_g = m_f \frac{X_d}{X_c - X_d} \quad (4)$$

The enthalpy balance for each component exchanging heat or work with the external environment is as follows:

$$Q_g = m_7 h_7 + m_8 h_8 - m_6 h_6 \quad (5)$$

$$Q_c = m_1 (h_1 - h_7) \quad (6)$$

$$Q_e = m_1 (h_3 - h_2) \quad (7)$$

$$Q_a = m_4 h_4 - m_3 h_3 - m_{10} h_{10} \quad (8)$$

$$W_p = m_6 (h_5 - h_4) \quad (9)$$

The specific flow solution (FR), which is the ratio of the mass flow of the rich solution (m_a), delivered by the pump and steam (m_f) desorbed by the generator [2] can be written:

$$FR = \frac{m_a}{m_f} = \frac{X_c}{X_c - X_d} \quad (10)$$

The coefficient of performance (COP) of the system is equal to:

$$COP = \frac{Q_e}{Q_g + W_p} \quad (11)$$

$$COP = \frac{(h_3 - h_2)}{h_7 + (FR - 1)h_8 - FR(h_6 + h_4 - h_5)} \quad (12)$$

The equations necessary for the calculation of thermodynamic and physical properties of the binary solution (LiBr /H₂O) are given by ASHRAE [10]

2.2. Solar collector

The role of double glazed flat plate collector is to convert solar radiation into heat energy through a heat transfer fluid (water). The output power of the double glazed flat plat is given by the formula:

$$Q_u = A_c \times Fr \times [(\tau \times \alpha) \times I_G - U_L \times (T_{fi} - T_a)] \quad (13)$$

The thermal efficiency of the solar collectors is the ratio of useful energy obtained in collector to solar radiation incoming to collector. It can be formulated as:

$$\eta = \frac{Q_u}{I_g \times A_c} \quad (14)$$

2.3. Thermal storage tank

In this type of installation in which a storage unit is placed in series with the solar collector, the temperature of the fluid entered T_{fe} in the collector is equal at each moment to the fluid temperature T_{stock} in the storage volume. We can make the simplifying assumption that the variation of T_{stock} between two instants t and $t + dt$ has a negligible influence on the value of the useful flux Q_u . We can then calculate the useful flux between t and $t + dt$ T_{fe} assuming constant and equal to $T_{stock}(t)$. Then we have:

$$T_{stock}(t+dt) = T_{stock}(t) + \frac{Q_u \times dt}{m_{stock} \times cp_{stock}} \quad (15)$$

3. Assumptions and parameters of calculation

Table 1 shows the characteristics of the double glazed flat plate collector. Since the performance of the system depends on the weather data, it is necessary to assume variable temperatures of the generator and condenser of the absorption cycle, so it has been assumed that the generator temperature is 5°C lower than the top layer temperature of the storage tank, the condenser temperature is 35°C during the design days, which are simply available by using a cooling tower. The absorption cycle has been simulated by typical and applied input parameters as the evaporator temperature of 5 °C, the solution heat exchanger efficiency of 70% and a constant solution pump flow rate. Fig. 2 present double glazed flat plate collector. The thermodynamic model showed that the absorption cycle could operate moderately well with delivery temperatures of 63–95 °C to the generator as shown by the results of Ghaddar et al. [3]. At low inlet temperatures of generator as 63 °C, very low absorber temperatures are required to absorb the refrigeration vapor of the evaporator; consequently, either it may be impractical or both a large cooling tower and a large heat exchanger of the absorber may be required. The heat losses of the connection equipments between the collector, storage tank and absorption system have been neglected.

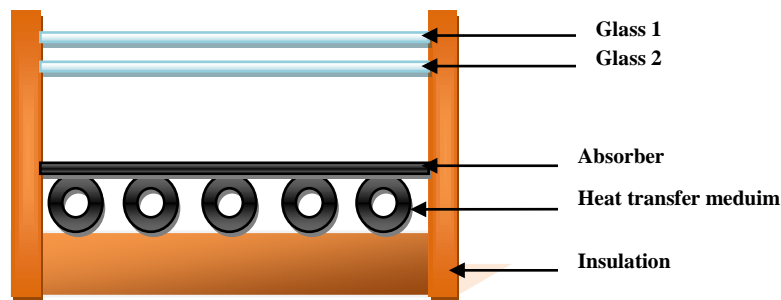


Fig. 2, Schematic diagram of double glazed flat plate collector

Table 1. Characteristic of the double glazed flat plate collector

Component	dimension
thickness glass1	0.002m
thickness glass2	0.002m
space between glass	0.007m
Emissivity of the glass	0.88
Emissivity of the absorber	0.7
Inside diameter of tube	0.035 m
absorption coefficient of the absorber	0.88

4. Results and Discussion

The system performance depends on the weather data; therefore, the variations of direct solar radiations, ambient and wind velocity have been used to simulate the solar absorption cooling system. The cooling load variations during the design days are presented in Fig. 3; the cooling loads varied from 3.1 KW to a peak of 10.39 KW at 13h. Fig. 4 shows the instantaneous solar radiation incident on a horizontal surface for the design day of July; the value of I_g in Adrar is about 200W/m^2 in the morning and it arrived to 1100W/m^2 at 13h. The ambient temperature of the design day of July is shown in Fig. 5. It has been assumed that the initial temperature ($T_{s,\text{initial}}$) of the storage tank and the inlet water temperature to the collector are equal to the ambient temperature. Fig. 6 shows the temperature distribution of the storage tank for the design day of July; the temperature of the storage tank varies from the temperature of 49°C to a maximum of 80°C at 15h, then it decrease to 67°C at the end of the day.

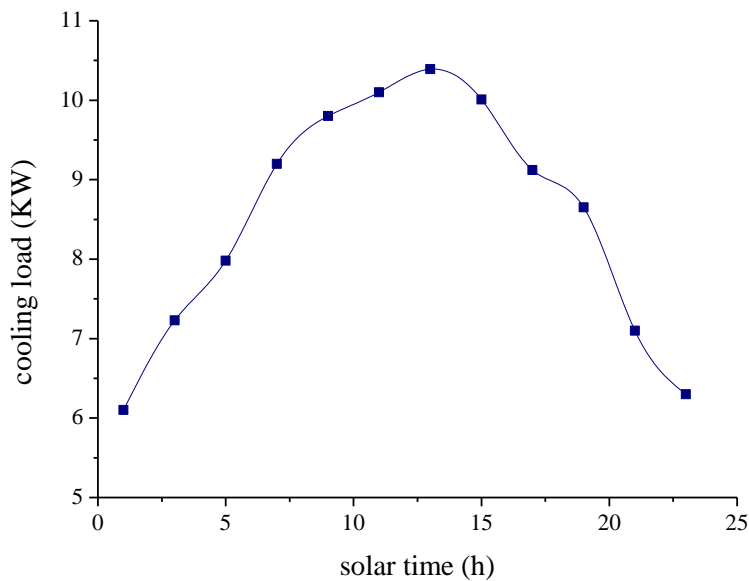


Fig. 3. Cooling load variations for design day of July

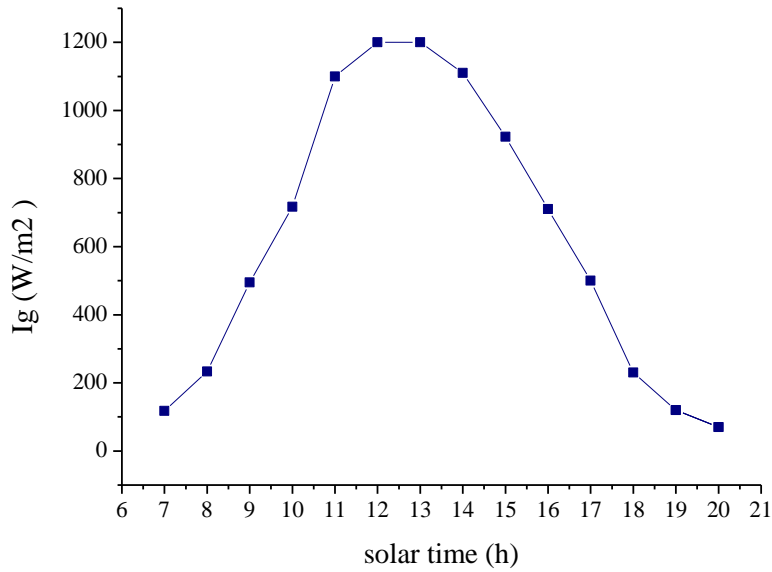


Fig. 4, Instantaneous solar radiation incident for design day of July

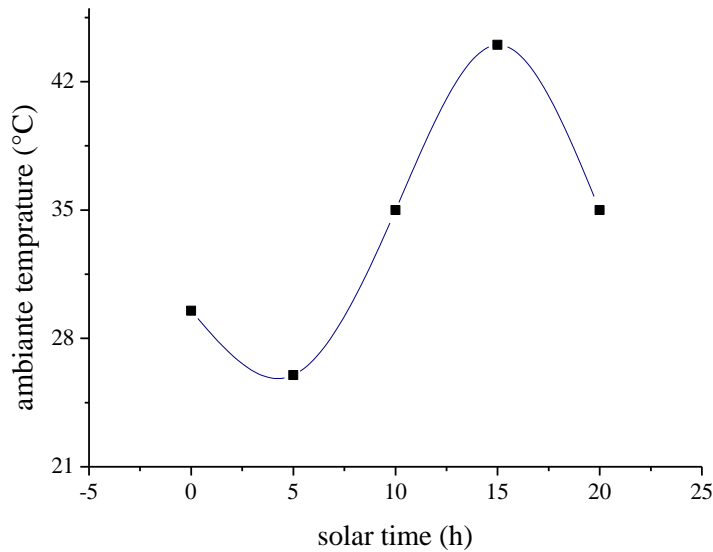


Fig.5, Variations of ambient temperature for design day of July

Figs. 7 and 8, respectively, present the minimum required collector area and the optimal capacity of the storage tank for the design day of July. It may be noted that the effect of the collector mass flow rate on the minimum required collector area is negligible because the heat loss at the double glazed flat plate collector are relatively low especially when we work with low temperatures. Against the optimal capacity of the storage tank is influenced by the change of the collector mass flow rate. Reducing the collector mass flow rate lowers the collector heat removal factor but also tends to increase the thermal capacity of the storage tank, which may improve the overall system performance [11]. At the beginning of the day the collector efficiency is low (about 0.45), it begins to increase with increasing of the solar direct radiation and the temperatures of the ambient to a value of 0.77.

The use of a storage tank increases the operation time of the absorption cycle, the simulation was made for a high flow, and therefore. The effect of the collector mass flow rate and tank volume on the thermal capacity of the storage tank has been considered. Fig. 8 shows that the obtained storage tank capacity is between 0.68 and 1.23m³ for 10.39 kW of cooling loads, which is in good agreement with other researches with various weather conditions and residential cooling load capacities [2–5]. The storage water temperature is about 67°C at the end of each day of operation, and the heat losses of the insulated storage tank are very low. Thus, the results showed that it is possible to assume the minimum initial temperature of the storage tank and inlet water temperature to the collector are about 56°C at the beginning of collector operation.

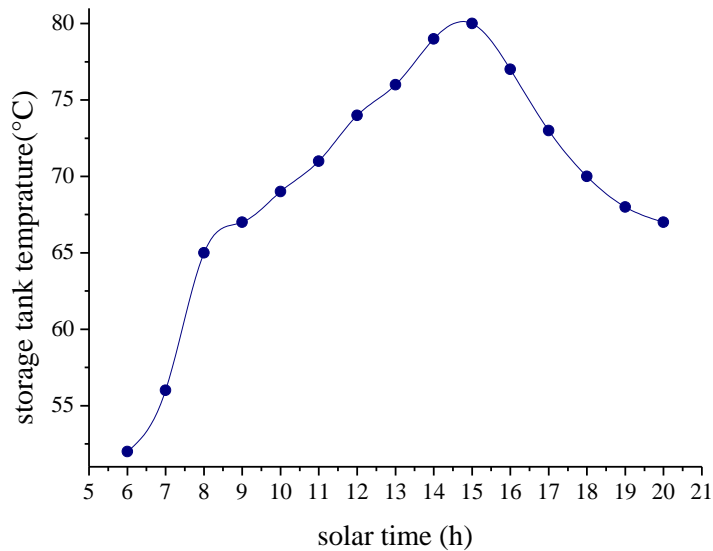


Fig. 6, Temperature distribution of storage tank for design day of July

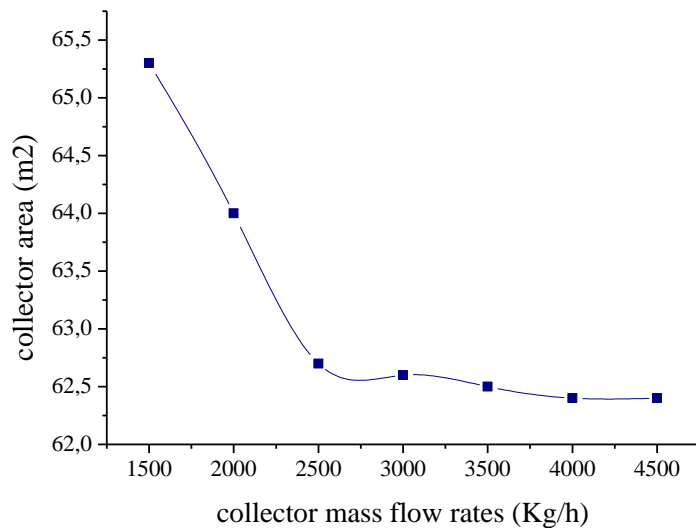


Fig. 7 Minimum required collector area at various collector mass flow rates for design day of July

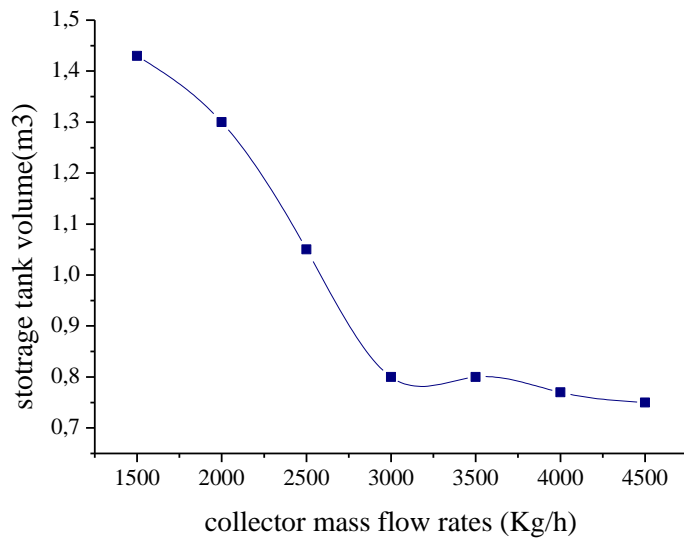


Fig. 8, optimal capacity of storage tank at various collector mass flow rates for design day of July

Fig. 9 shows that the collector efficiency during the design day of July. The result shows that the variation in collector efficiency is low; it stabilizes for most of the day at the value of 0.77; at the end of the day it decreased with the decrease of the solar direct irradiation to 0.45, this result is similar to the results of other researches Fig.10 shows the evolution of the COP during the design day for July the COP values are between 0.27 and 0.85 for generator temperatures between 56 and 80 ° C ,the COP value is maximum when cooling loads are important; absorber and the condenser is cooled by a cooling tower in series. The same result was obtained by Wijeyesundera [12] for COP amounts similar to the given values in this research. Figs. 11 present the minimum required collector area for operation over the sunshine hours for the design day of July. The results show that the minimum required collector area of the storage tank increases extremely when the operation of the absorption cooling system continues after sunset.

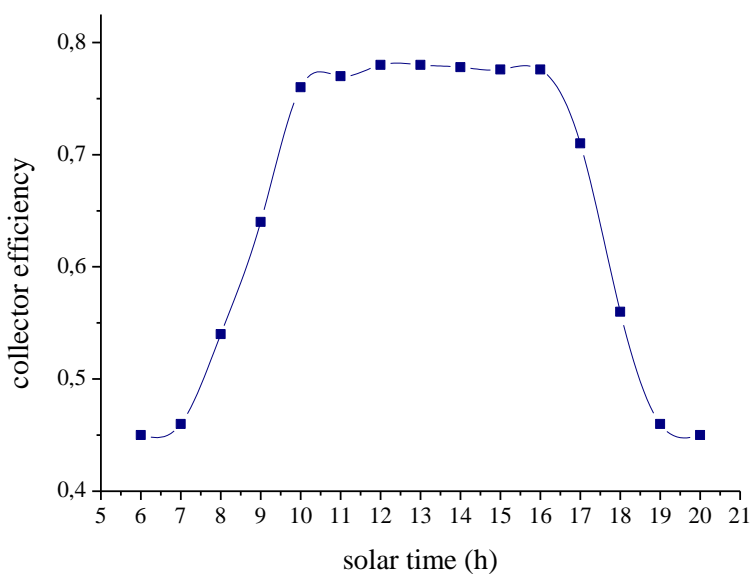


Fig. 9, Variations of collector efficiency during design day for July

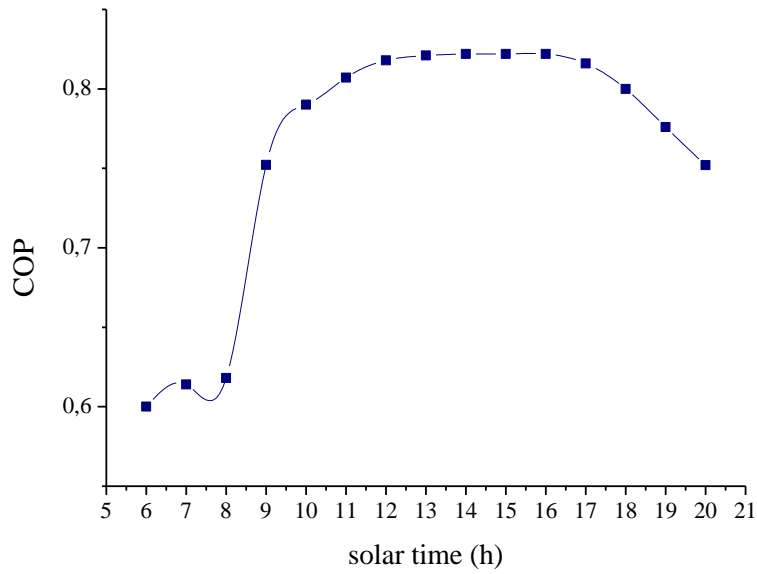


Fig. 10, Variations of COP during design days for July

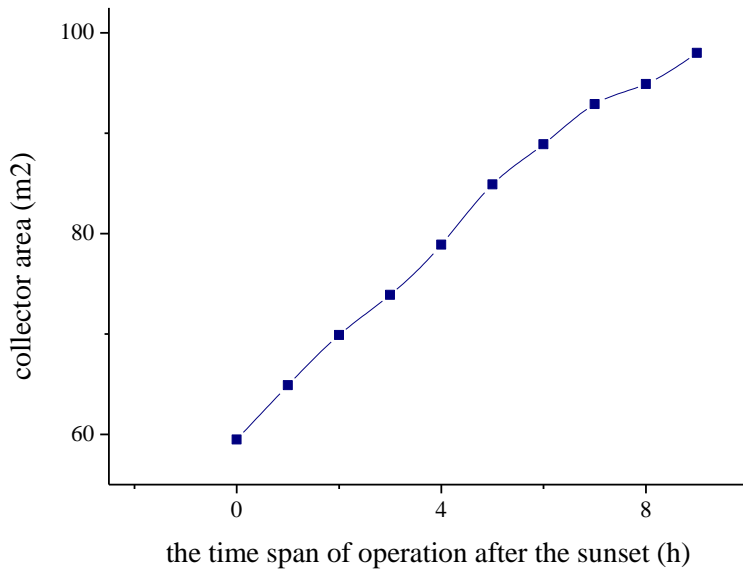


Fig. 11, Minimum required collector area for operation over sunshine hours for design day of July

Fig.12 shows that the optimal capacity of the storage tank has been increased after the sunset in comparison with the given value for operation during sunshine hours. A large storage volume is requested to cover cooling loads with the given value of operation. The auxiliary energy can also be used to supply the required heat of the generator of the absorption cycle instead of saving more solar energy in the tank after sunshine hours when the cooling load decreases considerably.

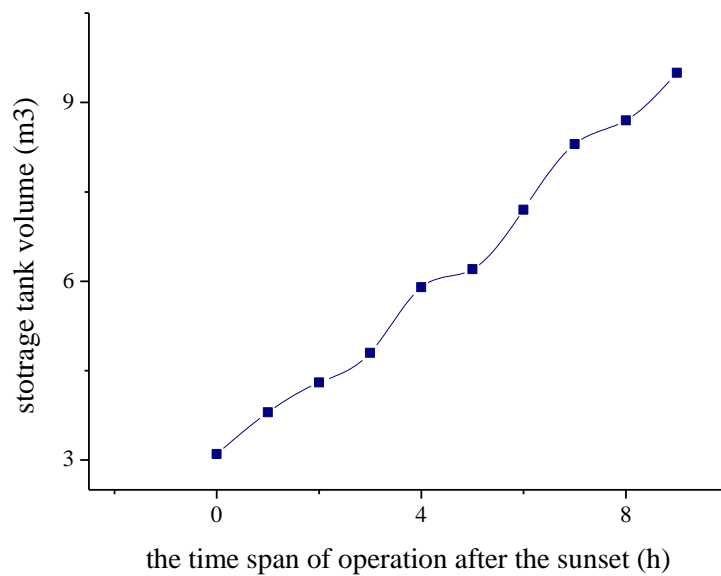


Fig. 12, Optimal capacity of storage tank for operation over sunshine hours for design day of July

5. Conclusion

The use of solar double glazed flat plate collector and insulated thermal storage tank with a single effect lithium bromide–water absorption cooling system is an effective solution to Adrar compared to other types of solar collectors such as flat plate collector. For a cooling load of 10.39 KW a large collector area and storage tank capacity are required for the design day of July are needed to cover the cooling loads during the day and after the sunset. For a house in Adrar of 200m², the minimum required collector area was about 62.7m² minimum for the collector mass flow rate of 2500 kg/h. The results showed that the collector mass flow rate did not affect the surface but it has a significant effect on the optimal capacity of the storage tank. The optimal capacity of the storage tank decreases when cooling loads increase. High values of (COP) of the system are related to increase cooling loads. For the operation of the system during the night, and the results are shown that the optimal capacity of the optimal capacity of the storage tank increases significantly.

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