Çukurova Üniversitesi Mühendislik Mimarlık Fakültesi Dergisi, 31(1), 371-379 ss. , Haziran 2016
Çukurova University Journal of the Faculty of Engineering and Architecture, 31(1), pp. 371-379, June 2016

Performance Analysis of Solar Powered Absorption Refrigeration System for Mersin Province

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Geliş tarihi: 04.02.2016 *Kabul tarihi: 09.03.2016*

Abstract

In this study, performance analysis of solar-powered absorption refrigeration (SPAR) system is evaluated using hourly atmospheric air temperature and solar radiation data in Mersin city. Ammonia-water (NH_3-H_2O) as refrigerant-absorbent pair and evacuated tube solar collector are selected for the design of the SPAR system. First, hourly cooling loads of the selected building are calculated. Then, thermodynamic analysis of the SPAR system is conducted according to this cooling load and required solar collector area is calculated for the air-conditioned space. Obtained results show that the maximum coefficient of performance (COP) value is observed in May, while the lowest value is seen in July and August. When the maximum solar radiation rate is observed to be 0,878kW/m^2 on 23 June at 01:00 p.m., the COP is calculated as 0,434. Finally, the optimum collector area is determined to be 50 m^2 for air-conditioned space area of 30 m^2.

Keywords: Absorption refrigeration system, Ammonia-water, Evacuated tube solar collector, Solar radiation, Thermal energy

Mersin İli için Güneş Enerji Destekli Absorpsiyonlu Soğutma Sisteminin Performans Analizi

Özet

Bu çalışmada, Mersin iline ait saatlik atmosfer hava sıcaklığı ve güneş ışınım verileri kullanılarak güneş enerji destekli absorpsiyonlu soğutma (SPAR) sisteminin performans analizi yapılmıştır. Güneş enerji destekli absorpsiyonlu soğutma sisteminin tasarımında soğutucu-soğurgan çifti olarak amonyak-su (NH_3-H_2O) ve vakum tüplü güneş kollektörü tercih edilmiştir. İlk olarak, seçilen binanın saatlik soğutma

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yükü hesaplanmıştır. Sonra, bu soğutma yüküne göre sistemin termodinamik analizi yapılmış ve soğutulacak bölge için gerekli güneş kollektör alanı hesaplanmıştır. Elde edilen sonuçlara göre maksimum etkinlik katsayısı, (COP) Mayıs ayında, minimum etkinlik katsayısı, (COP) ise Temmuz ve Ağustos aylarında elde edilmiştir. Birim alana gelen maksimum güneş ışınım değeri, 23 Haziran günü saat 13.00'de 0,878 kW/m^2 olduğunda, COP değeri 0,434 olarak hesaplanmıştır. Sonuç olarak, 30 m^2'lik soğutulacak bölge için optimum güneş kollektör alanı 50 m^2 olarak hesaplanmıştır.

Anahtar Kelimeler: Absorpsiyonlu soğutma sistemi, Amonyak-su, Vakum tüplü güneş kollektörü, Güneş ışınımı, Isıl enerji.

1. INTRODUCTION

Solar energy can be used in three main ways, and it is important to distinguish between these three types: passive heat, solar thermal and photovoltaic energy. Solar photovoltaic system have the widespread application due to its high power to weight ratio, simple, compact in size, less maintenance and no moving parts [1]. Solar thermal technologies contribute significantly to the hot water production in many countries, and increasingly to space heating and air-conditioning as well as industrial processes [2]. In summer, particularly under tropical climate, air conditioning has the highest energy expenditure in buildings [3]. In this context, improvements in energy efficiency should be achieved or alternative air-conditioning technologies should be developed for residential and commercial buildings. One of the important methods to reduce the energy consumption is the use of solar air-conditioning systems. Solar energy usage for comfort cooling purpose in buildings offers advantages of using an inexhaustible and free heat source to meet comfort cooling needs most of the time. Considering that comfort cooling demand increases with the intensity of solar radiation. For this reason, solar air-conditioning has been considered as a logical solution [4].

In Turkey, the yearly average solar-radiation is 3.6 kWh/m^2 day, and the total yearly radiation period is 2610 hours. Especially, the south coast cities of Turkey which attract intensive attention as a touristic places have considerable population and hot climate. Therefore, air conditioning is vitally important due to the indoor comfort conditions especially in summer season [5].

Due to these reasons, solar energy can be used in the application of absorption refrigeration systems. Solar-powered absorption refrigeration (SPAR) system can be both economic and ecological applications by combining the need for comfort cooling of buildings with a high rate of availability of solar energy [6,7]. Absorption refrigeration systems are thermally activated and they do not require high input shaft power. Therefore, where power is unavailable and expensive, available waste heat, geothermal and solar heat can be used as a heat source for absorption machines. Since no chlorofluorocarbons (CFCs) are used, absorption refrigeration systems are appropriate for preserving the environment. In addition, they do not contribute emissions to the ozone depletion or to the global warming. Although absorption refrigeration systems seem to provide many advantages, their COP is too low and initial investment cost is very high [4,8]. The most important parameter is the performance of solar collectors in the design of absorption cooling machine. If the fluid temperature reaches 150°C, the performance of absorption refrigeration machines can be doubled leading to a high COP value comparing to the case of the fluid temperature of 90°C where single-acting systems are used causing a lower COP value. The one of the main difference between absorption refrigeration and conventional mechanical vapor compression system is the use of absorbent refrigerant pair instead of compressor. In this way the system does not need a high power input for the mechanical compressor.

The use of vapor absorption systems by means of solar thermal energy has been generated a high interest in the last decades [9]. Thermodynamic analysis and establishment of the design parameters before constructing the SPAR system are substantially important in terms of evaluation of the economic performance. The objective of the present study is to analyze the performance of the SPAR system with an air-conditioned space of 30 m^2 for Mersin province in Turkey. First, hourly cooling loads of the selected building are calculated. According to this cooling load, the thermodynamic analysis of the SPAR system is conducted and later optimum size of solar panel is determined for the air-conditioned space.

2. MATERIAL AND METHODS

Schematic diagram of the SPAR system is presented in Figure 1. It consists of three main parts: a solar collector, a storage tank and a vapor absorption air conditioning system. Absorption refrigeration system which uses ammonia-water (NH_3-H_2O) as a working fluid consists of four heat exchangers: generator, absorber, condenser and evaporator. An evacuated tube collector is preferred as a solar collector. Because of its design, evacuated tube solar collector captures almost all of the solar radiation. Its efficiency is much higher than the flat plate collectors.

The mass and energy balance equations are used in the analysis of the SPAR system. All components of the system are counted to have an individual control volume [4,10]. Equations (1) and (2) should be used for mass balance where ṁ (kg/s) represents mass flow rate and x represents mass concentration of the NH_3.

$$\sum \dot{m}_i - \sum \dot{m}_0 = 0 \tag{1}$$

$$\sum (\dot{m}\cdot x)_i - \sum (\dot{m}\cdot x)_0 = 0 \tag{2}$$

The first law of thermodynamics for steady flow process is presented in Eq. (3). In this equation, h (kJ / kg) is enthalpy, Q (kW) is heat transfer and W (kW) is the shaft power.

$$\sum (\dot{m}\cdot h)_i - \sum (\dot{m}\cdot h)_0 + \left[\sum Q_i - \sum Q_0\right] + W = 0 \tag{3}$$

Equations of (4), (5) and (6) are used in order to determine heat transfer capacity of the generator. In these equations, $\dot{m}_3$, $\dot{m}_4$ and $\dot{m}_7$ represent mass flow rates of the rich solutions of heat exchanger, the weak solution returning from the generator and the ammonia vapor exiting from generator, respectively. Q_G indicates the heat entered to the generator. Here, h_3, h_4 and h_7 symbolize the enthalpies of the rich solution, the weak solution and ammonia vapor, respectively.

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \tag{4}$$

$$\dot{m}_3\, x_3 = \dot{m}_4\, x_4 + \dot{m}_7 \tag{5}$$

$$Q_G = \dot{m}_7 h_7 + \dot{m}_4 h_4 - \dot{m}_3 h_3 \tag{6}$$

Equations (7), (8) and (9) are obtained from equations (4) and (5). Here, *f* which is stated below is the mass flow rate ratio.

$$f = \frac{\dot{m}_3}{\dot{m}_7} = \frac{1 - x_4}{x_3 - x_4} \tag{7}$$

$$\frac{\dot{m}_3}{\dot{m}_7} = \frac{\dot{m}_4}{\dot{m}_7} + 1 \tag{8}$$

$$\frac{\dot{m}_4}{\dot{m}_7} = f - 1 \tag{9}$$

Equations (10) and (11) are obtained by applying laws of mass and energy balances to the condenser. Q_C indicates the amount of heat exiting from the condenser, while $\dot{m}_8$ represents the saturated liquid. And, h_7 and h_8 symbolize enthalpy of ammonia entering and exiting the condenser, respectively.

$$\dot{m}_7 = \dot{m}_8 = \dot{m}_{ref} \tag{10}$$

$$Q_C = \dot{m}_{ref} (h_8 - h_7) \tag{11}$$

Equations (1) and (3) are applied between inlet and outlet of evaporator to obtain equations (12) and (13) assuming that there is a steady flow and no shaft power is needed. Q_E indicates the heat taken from the air-conditioned space. Here $\dot{m}_9$ represents liquid ammonia after expansion valve, $\dot{m}_{10}$ represents ammonia vapor exiting from evaporator and h_9 and h_{10} symbolize the enthalpy of liquid ammonia and ammonia vapor respectively.

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{ref} \tag{12}$$

$$Q_E = \dot{m}_{ref} (h_{10} - h_9) \tag{13}$$

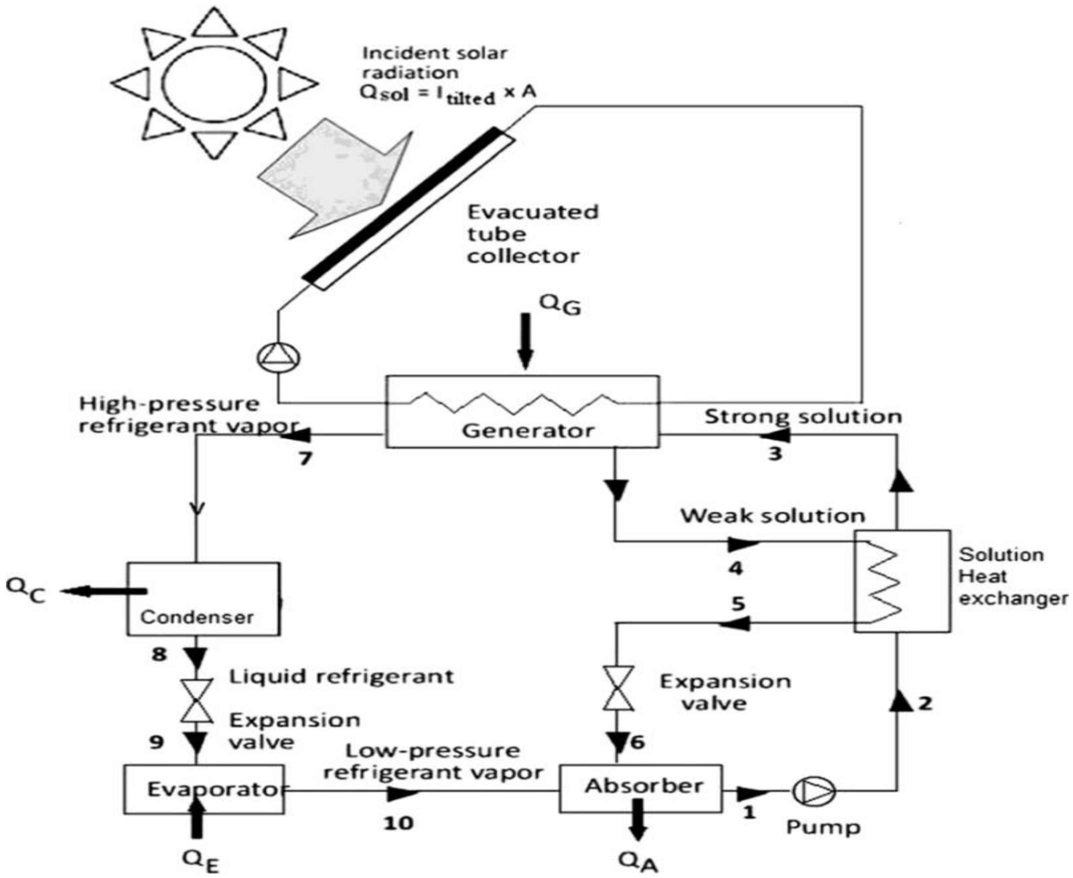

Figure 1. Components of the solar powered absorption refrigeration system [4]

Equations of (14), (15) and (16) are derived using one dimensional continuity and first law thermodynamic equations to the absorber. Here, $\dot{m}_1$ and $\dot{m}_6$ represent the rich solution that exits in the absorber and the weak solution returning from the generator, respectively. Secondly, h_1 and h_6 symbolize the enthalpy of the rich solution leaving the absorber and the weak solution returning from the generator, respectively. Q_A is the heat taken out from the absorber. Lastly, q_A (kj/kg) is the heat dissipated per unit mass.

$$\dot{m}_1 = \dot{m}_{10} + \dot{m}_6 \tag{14}$$

$$Q_A = \dot{m}_1 h_1 - \dot{m}_{10} h_{10} - \dot{m}_6 h_6 \quad (15)$$

$$q_A = f \cdot h_1 - h_{10} - (f - 1) \cdot h_6 \quad (16)$$

The energy balance applied to the entire system shows that the sum of incoming and out coming heat transfers must be zero. The Eq. (17) is obtained when the system is considered to be steady-state process and hence heat losses and the pump work are neglected.

$$Q_C + Q_A = Q_G + Q_E \quad (17)$$

The cooling coefficient of performance COP is obtained by ratio of evaporator heat load and generator heat load.

$$COP_{cooling} = \frac{Q_E}{Q_G} = \frac{\dot{m}_{10} h_{10} - \dot{m}_9 h_9}{\dot{m}_7 h_7 + \dot{m}_4 h_4 - \dot{m}_3 h_3} \quad (18)$$

Q_{sol} (kW) is solar radiation received with the solar collector, A is the surface area of solar collector, I (kW/m^2) is the solar radiation, $Q_{sol\text{-}gen}$ (kW) shows the heat transferred from the solar collector to the generator and η symbolizes the thermal efficiency of solar collector.

$$\eta = \frac{Q_{sol\text{-}gen}}{Q_{sol}} \quad (19)$$

Here, τ and α represent coefficient of the transmission and absorption coefficient, respectively, U is the overall heat transfer coefficient (W/m^2K), F is the solar-collector efficiency factor, T_W (°C) and T_A (°C) indicate temperatures of the mean water and the atmospheric air, respectively. Typical values of τα and U for the evacuated tube collector are commonly taken as 0.84-0.86 and 0.8 W/m^2 K, respectively.

$$Q_{sol} = I \cdot A \quad (20)$$

$$Q_{sol\text{-}gen} = F \bullet A[(\tau\alpha)I - U(T_W - T_A)] \quad (21)$$

By using Eqs. (20) and (21), solar collector efficiency can be written as:

$$\eta = F \bullet (\tau\,\alpha) - \frac{F\,U\,(T_W - T_A)}{I} \quad (22)$$

3. RESULTS AND DISCUSSION

Average values of solar radiation, *I* and atmospheric air temperature, T_A are obtained by taking the mean hourly values of temperature and solar radiation between 1999 and 2008. These data are taken from the weather station of Turkish State Meteorological Service situated in the province of Mersin (with coordinates of 36,8 latitude and 34,63 longitude at an altitude of 3,4 m). By considering human comfort, the indoor air temperature, T_i is taken as 24°C.

Schematic diagram of the air-conditioned space is shown in Figure 2. It is located south and west exposure. There are two windows with 2m×2m size on both exposures. External wall thickness is 0,2 m and internal wall thickness is 0,1 m. Air-conditioned space floor area is 30 m^2 and its size is 5m×6m. Air-conditioned space's height is 3 m and so its volume is 90 m^3. Wall transmission is taken as 25 m^2. Window and skylight solar areas are taken as 8 m^2.

In the calculation of heat gain, overhead lighting load is taken as 750 W and electric equipment load is taken as 500 W. Four people are supposed to live in the air-conditioned space and 130 W loads per person are taken. Safety factor is 15%. It is assumed that there are no heat gains through the doors, floor and roof, since they have the same temperature as the air-conditioned space. Heat gains due to infiltration and ventilation are ignored. The hourly comfort cooling load capacity of air conditioned space, Q_E is determined by using the Cooling Load Hourly Analysis Program (HAP) 4,4. HAP is a computer tool which assists engineers in designing HVAC systems for commercial buildings. It is a tool for estimating loads and designing systems, as well as for simulating energy use and calculating energy costs. It uses the ASHRAE-endorsed transfer function method for load calculations and detailed 8,760 hour-by-hour energy simulation techniques for the energy analysis.

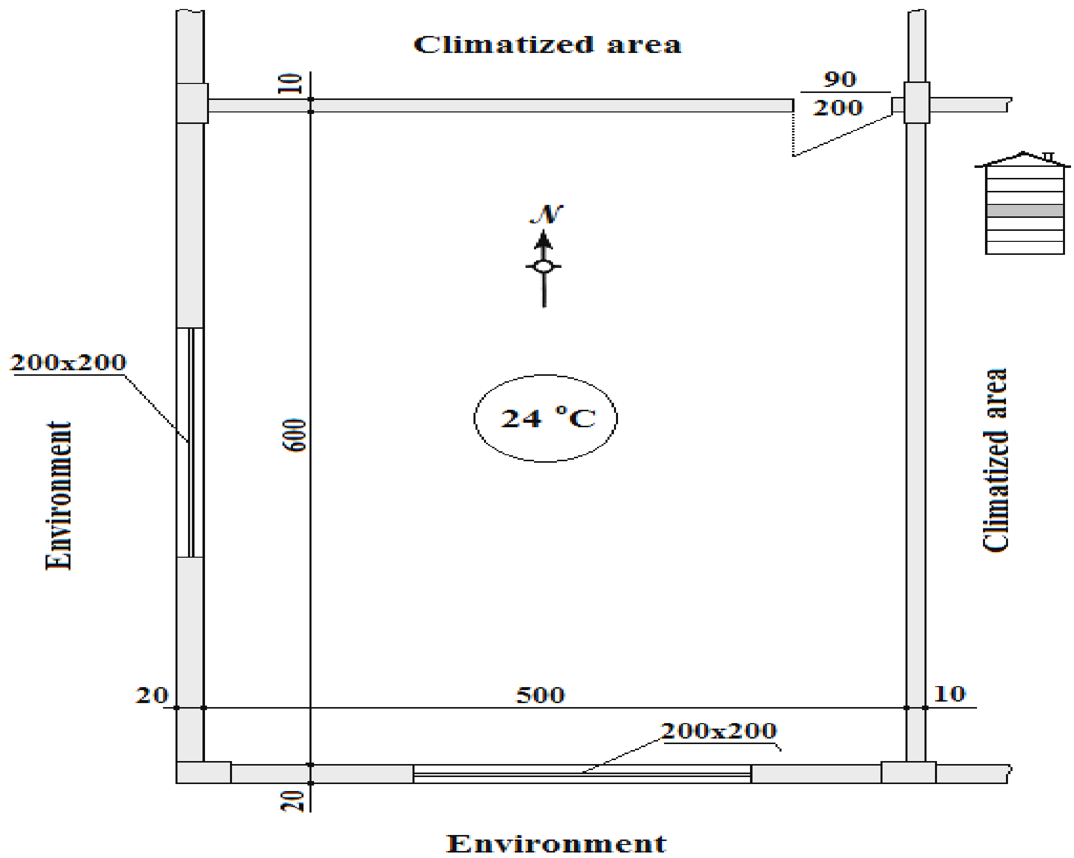

Figure 2. Schematic representation of the air-conditioned space

The hourly cooling loads, Q_E on the 23rd days of five-months are presented in Figure 3. The hourly comfort cooling load, Q_E takes different values during the day. It reaches its highest value at 03:00 p.m. in August and September; at 04:00 p.m. in May, June and July. The Q_E values vary between 2,60 kW and 5,19 kW in May, between 3,80 kW and 8,59 kW in June, between 4,06 kW and 9,06 kW in July, between 3,88 kW and 9,61 kW in August, between 3,69 kW and 10,27 kW in September. Amongst five-months, the highest Q_E values are found in September.

The SPAR system is designed for the air-conditioned space described in the previous section. NH_3 temperature of generator outlet, T_G is 110°C and evaporation temperature, T_E is 10°C. Condensing and absorber temperatures, T_C and T_{AB} are assumed to be the same because they release their heat to the same heat sink.

Condensing temperature, T_C is thought to be 10°C higher than the atmospheric air temperature, T_A. For 23rd of five selected months, $COP_{cooling}$ value of the SPAR system varies during the day as seen in Figure 4. $COP_{cooling}$ value varies ranging from 0,41 to 0,49. The highest values of $COP_{cooling}$ occur in May, while the lowest values of $COP_{cooling}$ take place in July and August.

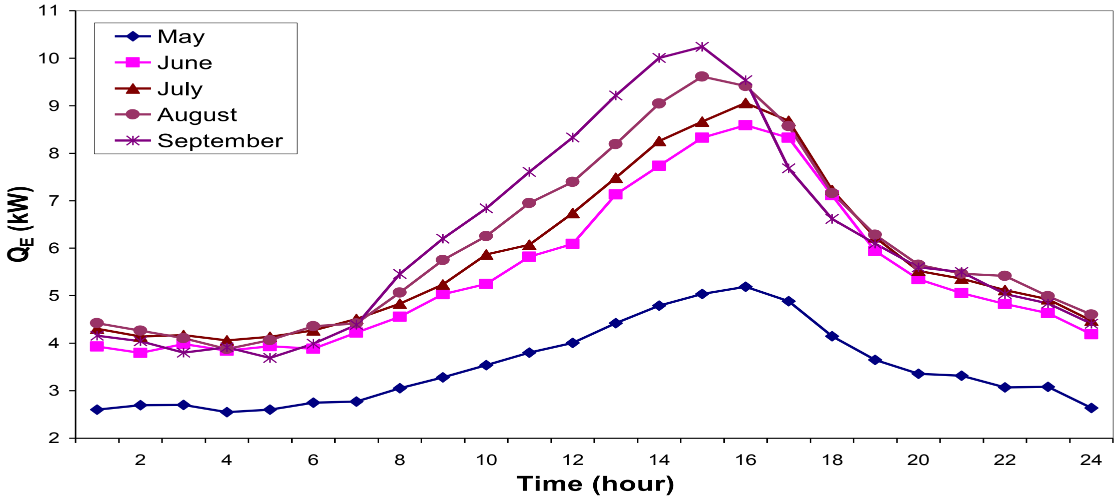

Figure 3. Variations of hourly values of Q_E on the 23rd days of five-months

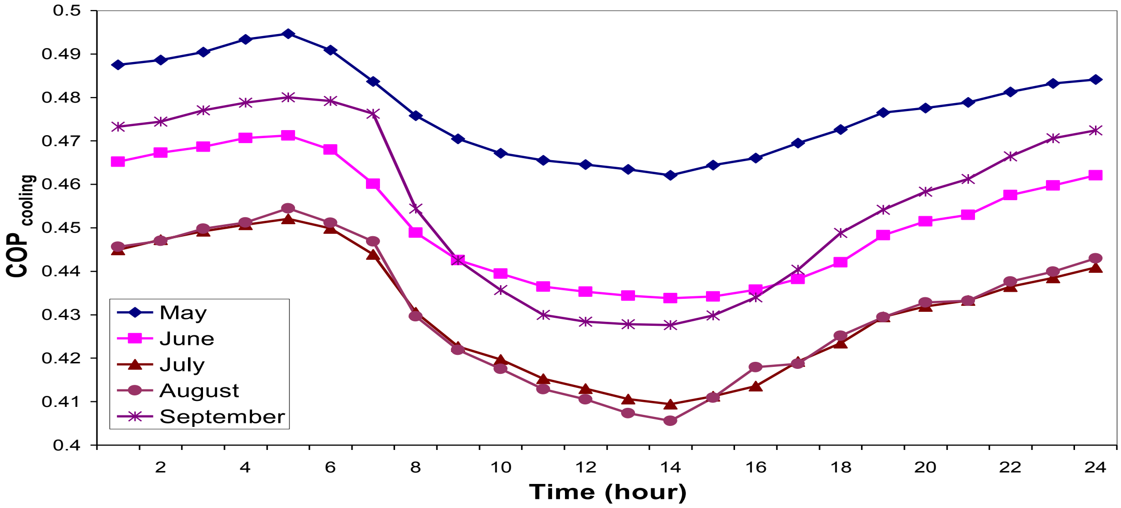

Figure 4. Variations of hourly values of $COP_{cooling}$ on the 23rd days of five-months

Proper solar collector areas, *A* are needed for providing the necessary comfort cooling capacity between 7:00 a.m. and 7:00 p.m. Because the low levels of solar radiation effects occur during morning and evening hours the necessary solar radiation collector area, *A* is obviously calculated quite large. Therefore, ensuring air-conditioning with solar energy, physically and economically to be adequate in terms of time, for example, taking the SPAR system in operation between 9:00 am and 4:00 pm the optimum solar collector area, *A* should be determined accordingly. The required solar collector area, *A* is presented in Figure 5 for duration of time between 08:00 a.m. and 05:00 p.m. As seen in the figure, the required collector area takes the lowest values in May due to the lowest values of the atmospheric air temperature, T_A and the higher values of the solar radiation, *I*. The required collector area is higher in September due to the lowest values of the solar radiation, *I* and the higher values of the atmospheric air temperature, T_A. According to this, the optimum solar collector area required for the air-conditioning system is determined as 50 m^2.

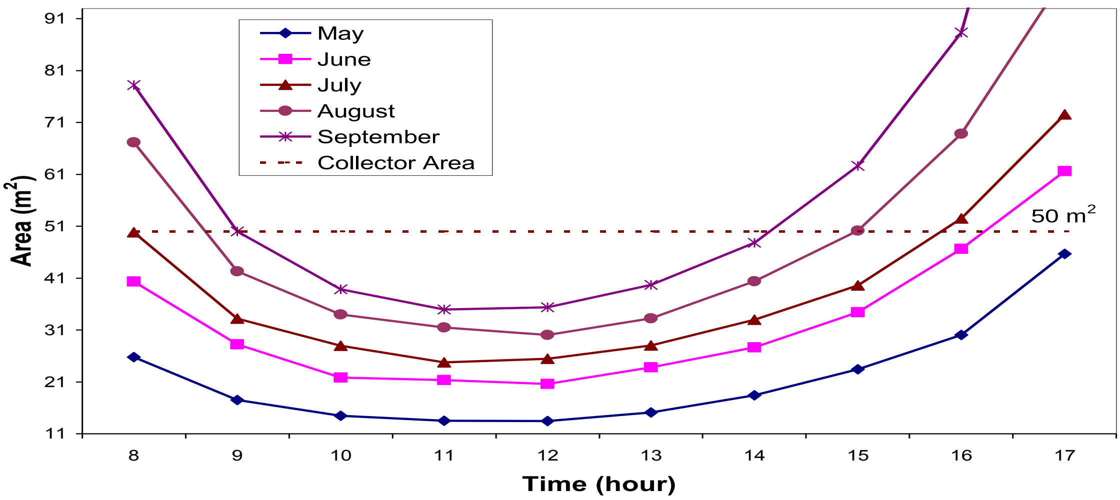

Figure 5. The collector area required for air-conditioning between 08:00 a.m. and 05:00 p.m.

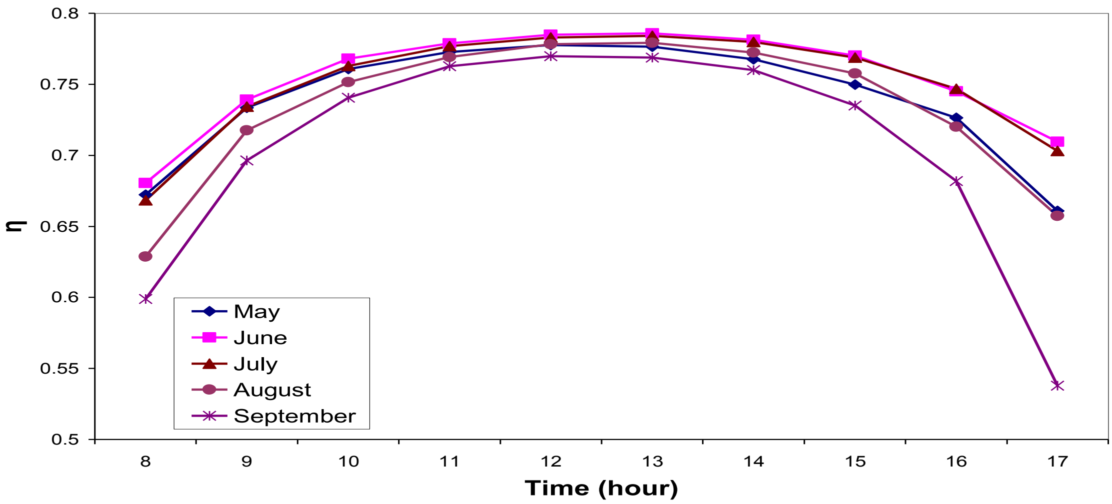

Figure 6. Variation of hourly efficiency of solar collector on the 23rd days of five-months

Hourly efficiency of solar collector on the 23rd days of five-months is shown in Figure 6. Solar collector efficiency is directly proportional to the atmospheric air temperature, T_A and solar radiation, *I*. Therefore, solar collector efficiency, η increases towards the noon time and then it starts to decrease. The solar radiation is more decisive than the atmospheric air temperature, T_A on the efficiencies of solar collector. Levels of atmospheric air temperatures, T_A are lower in May comparing to the value obtained in September but solar radiation, *I* level is higher in May comparing to September. Therefore, solar collector efficiency, η is higher in May than in September. Between the hours ranging from 08:00 a.m. to 05:00 p.m., the value of solar collector efficiency, η increases from 0,66 to 0,78 in May, from 0,68 to 0,79 in June, from 0,67 to 0,78 in July, from 0,63 to 0,78 in August and finally from 0,54 to 0,77 in September.

4. CONCLUSION

This study investigates the SPAR system by using hourly atmospheric air temperature, T_A and solar radiation, I data in Mersin province of Turkey. The obtained results show the benefits and features of using solar energy in the air-conditioning technology. The following conclusions can be drawn from the present study:

- Atmospheric air temperature, T_A is the most important factor affecting $COP_{cooling}$. Atmospheric air temperature, T_A is inversely proportional to the value of $COP_{cooling}$.
- COP values vary during the day of months considered, for example, ranging from 0,46 to 0,49 in May, from 0,43 to 0,47 in June, from 0,41 to 0,45 in July, from 0,40 to 0,45 in August, from 0,43 to 0,48 in September.
- The highest COP values occur in May, while the lowest COP values occur in July and August.
- The maximum solar efficiency is observed on the 23rd of June at 01:00 p.m. This solar efficiency is calculated as 0,8, while solar radiation, I is 878 W/m^2.
- According to the results, the optimum required collector area is determined as 50m^2 for air-conditioned space area of 30 m^2.

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