

PERFORMANCE EVOLUTION OF SOLAR-POWERED VAPOR ABSORPTION REFRIGERATION SYSTEM

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REFERENCE NO	ABSTRACT
SOLR-04	To decrease the energy consumption, solar-powered vapor absorption refrigeration system (VAR) may be an alternative, since Mediterranean region have high solar energy potential. In this study, in order to prove that this kind of alternative air conditioning systems can work in Mediterranean region, the performance analysis of solar-powered VAR is evaluated using hourly atmospheric air temperature and solar radiation data in Mersin, Turkey. Firstly, hourly comfort cooling loads, Q_{EV} of the selected space is taken from literature. Then, the heat capacity of the each components of the solar-powered VAR system is analysed according to this comfort cooling load and required solar collector area, A is calculated for the air-conditioned space. To decrease and compare required solar collector area, A, two working fluids which are water-LiBr and water-LiCl are chosen for solar-powered VAR. The results are presented in figures and tables.

Keywords:
Absorption refrigeration, solar cooling, water-LiBr, water-LiCl

1. INTRODUCTION

Air-conditioning the buildings constitutes a big part of energy consumption of whole world just like %30-40[1]. Especially developing countries demands more energy to consume because of increasing population, rising life and comfort standards. Parallel to this occasion, nonrenewable sources have been consumed rapidly causing increment of environmental pollution. Vapor compression systems which are mainly used system to air conditioning the buildings consumes quite large energy since compressing vapor is harder than pumping liquid. So, if there is available heat energy, it is very convenient using the system which driven by heat to get refrigeration or air-conditioning. In this context, Vapor Absorption Refrigeration (VAR) systems are the first systems to come to mind. Solar energy, a futuristic energy sources for countries that has solar energy potential, is one of the energy sources for VAR systems.

Researchers have been interested in single effect solar powered VAR system for years. Some of them investigates the solar VAR systems experimentally [2-4]. And some of

them conducted simulations to investigate solar powered VAR systems [5, 6].

Tierney [7] and Ming Qu et al. [8] investigated solar powered double effect VAR systems. Bellos et al [9] compared solar powered VAR systems with water-LiBr and water-LiCl in exergy approach. Bellos et al. [10] evaluated the solar powered VAR systems with different collector types.

In this study, performance evolution of solar-powered vapor absorption refrigeration system is conducted on hourly basis with different working fluids (water-LiBr, water-LiCl). Singled and double effect VAR systems are studied.

2. SYSTEM DESCRIPTIONS

The evacuated tube solar collector receives energy from sunlight then transferred it to the generator in single effect VAR system or high pressure generator (HPG) in double effect VAR system by a heat exchanger.

2.1. Single effect VAR system

Single effect VAR system has an evaporator, condenser, generator, absorber and solution heat exchanger. The operating sequence of the

VAR system shown in Fig. 1 is as follows; the water-LiBr/LiCl solution which is referred to as weak solution because of the consistence of relatively high refrigerant quantity is pumped from the absorber to the solution heat exchanger, where the solution temperature rises. After solution heat exchanger, it enters to the generator at state 7. Some refrigerant (water) is removed from the weak solution by supplying heat to the generator. The remain water-LiBr/LiCl solution which is referred to as rich solution because of the consistence of relatively low refrigerant moves to the solution heat exchanger to give heat to the rich solution. So, its temperature decreases. As exiting from the solution heat exchanger, it enters to the absorber at state 10. At the state 1, the superheated vapor moves to the condenser where it releases heat to the ambient. At the condenser exit, the refrigerant becomes saturated vapor. After condenser, the refrigerant passing through expansion valve enters the evaporator at state 3. It evaporates here by rejecting heat from the medium and becomes saturated vapor. After that, it moves to the absorber and encounters the rich solution coming from the solution heat exchanger. Here, the rich solution absorbs the refrigerant and becomes weak solution. The heat existing at the absorber is released to the ambient.

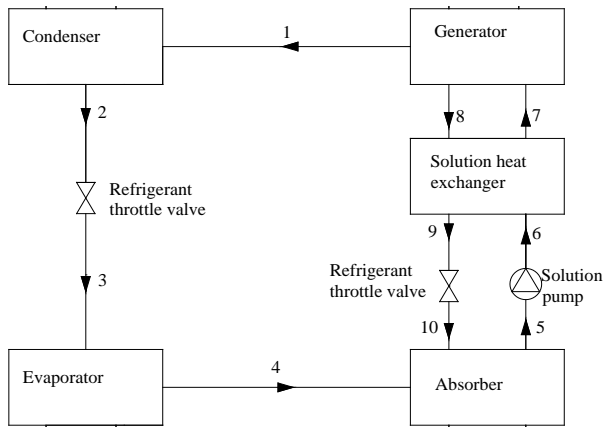


Fig. 1. The single effect Vapor Absorption System

Mass balance and general energy equations for single effect VAR system are as follows;

$$\sum m_i = \sum m_o \quad (1)$$

$$q_{CO} = \frac{Q_{CO}}{\dot{m}_1} = h_2 - h_1 \quad (2)$$

$$q_{EV} = \frac{Q_{EV}}{\dot{m}_1} = h_4 - h_3 \quad (3)$$

$$q_{GE} = \frac{Q_{GE}}{\dot{m}_1} = h_1 + fh_8 - (f+1)h_7 \quad (4)$$

$$q_{AB} = \frac{Q_{AB}}{\dot{m}_1} = (f+1)h_5 - h_4 - fh_{10} \quad (5)$$

$$q_{she} = \frac{Q_{she}}{\dot{m}_1} = f(h_8 - h_9) \quad (6)$$

$$q_{she} = (f+1)(h_7 - h_6) \quad (7)$$

$$f = \frac{X_7}{X_8 - X_7} \quad (8)$$

2.2. Double effect VAR system

The main purpose of the double effect VAR system is to utilize the heat released from the high pressure condenser to drive an additional low pressure generator to get higher COP. As it seen in Figure 2, the double effect VAR has two generator and two condenser.

In this system, the water-LiBr/LiCl solution has three concentrations level. The weak concentration of solution at the absorber outlet is pumped to the HPG and at the outlet of HPG, the solution becomes strong. Then, it enters to LPG and at the outlet of LPG, the solution becomes stronger. As it is seen from Figure 2, the double effect VAR system has three pressure level.

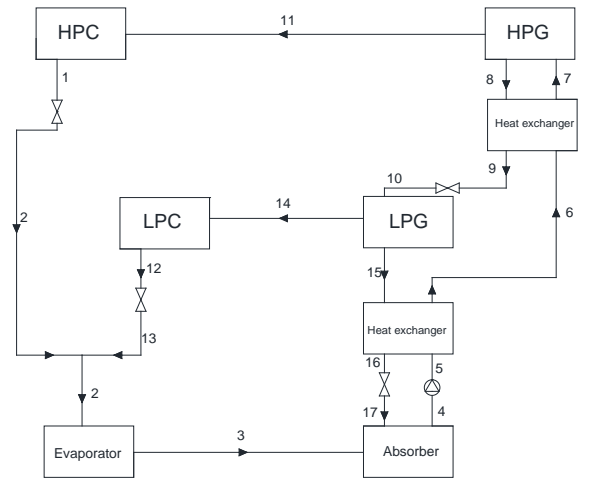


Fig. 2. The double effect Vapor Absorption System

$$q_{EV} = \frac{Q_{EV}}{\dot{m}_3} = h_3 - h_2 \quad (9)$$

$$q_{HPC} = \frac{Q_{HPC}}{\dot{m}_3} = \frac{\dot{m}_{11}}{\dot{m}_3} (h_1 - h_{11}) \quad (10)$$

$$q_{LPC} = \frac{Q_{LPC}}{\dot{m}_3} = \frac{\dot{m}_{14}}{\dot{m}_3} (h_{12} - h_{14}) \quad (11)$$

$$q_{AB} = \frac{Q_{AB}}{\dot{m}_3} = h_4 - h_3 + \frac{\dot{m}_{17}}{\dot{m}_3} (h_4 - h_{17}) \quad (12)$$

$$q_{HPG} = \frac{Q_{HPG}}{\dot{m}_3} = \frac{\dot{m}_{11}}{\dot{m}_3} h_{11} + \frac{\dot{m}_8}{\dot{m}_3} h_8 - \frac{\dot{m}_7}{\dot{m}_3} h_7 \quad (13)$$

$$q_{LPG} = \frac{Q_{LPG}}{\dot{m}_3} = \frac{\dot{m}_{14}}{\dot{m}_3} h_{14} + \frac{\dot{m}_{15}}{\dot{m}_3} h_{15} - \frac{\dot{m}_{10}}{\dot{m}_3} h_{10} \quad (14)$$

COP of both system is calculated as follows;
COP=Heat taken in evaporator/net heat supplied

2.3. Evacuated tube solar collector

The surface area, A of solar collector receives solar radiation, Q_{sol} from the sun and supplies Q_{sol-ge} to the generator. It is assumed that the evacuated solar collector surface is placed with tilted angle of 23° .

The thermal efficiency, η of evacuated tube solar collector is defined the ratio of supplied heat Q_{sol-ge} to the solar radiation production Q_{sol-ge} . [11]

$$\eta = \frac{Q_{sol-ge}}{Q_{sol}} \quad (14)$$

$$Q_{sol} = IA \quad (15)$$

$$Q_{sol-ge} = FA[(\tau\alpha)I - U(T_w - T_A)] \quad (16)$$

Where τ is the transmission coefficient, α is the absorption coefficient, U is the overall heat transfer coefficient, F is the solar-collector efficiency factor, T_w is the mean water temperature and T_A is the ambient air temperature. The values of $\tau\alpha$ and U are taken as 0.84-0.86 and $0.8 \text{ W/m}^2\text{K}$ [11].

3. RESULTS AND DISCUSSION

3.1. Air-conditioned space

The air conditioned space with 30 m^2 area is suited on the fourth floor of seven story building. It is located south and west exposure. To calculate the hourly comfort cooling load the data of solar radiation, I and ambient temperature is taken from the weather

station of Turkish State Meteorological Service situated in the province of Mersin. The hourly comfort cooling load (evaporator heat capacity) of the air conditioned space is shown in Figure 3.

As it is seen in Figure 3, the maximum evaporator heat capacity is obtained as 10.237 kW at 15.00 in September. The minimum heat capacity of evaporator is obtained as 2.55 kW at 04.00 in May.

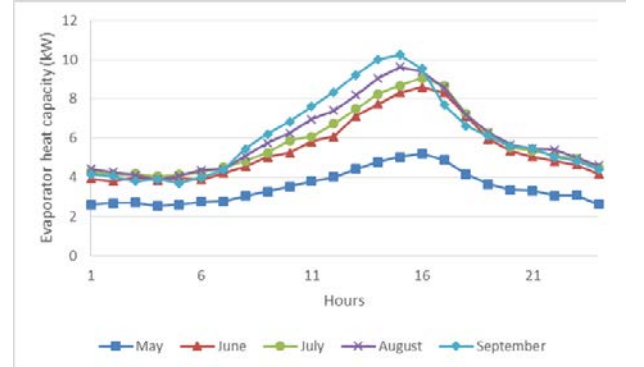


Fig. 3. Hourly evaporator heat capacity variation [12]

3.2. VAR System results

In this section, the system performances of single effect VAR and double effect VAR is evaluated. Two different working fluid is chosen to make comparisons. These are water-LiBr and water-LiCl for single effect VAR. Water-LiCl is considered for double effect VAR system. To avoid crystallisation problem for water-LiCl solution and to make comparison with the other working fluid evaporator temperature is taken to be 15°C . The absorber, the condenser and the HPC releases heat to the same heat sink which is ambient air. So their temperature is to be as T_a+10 . The HPG temperature for double effect VAR and the generator temperature single effect VAR with water-LiCl working fluid is taken as 100°C . The generator temperature single effect VAR with water-LiBr working fluid is taken as 70°C . The LPG temperature for double effect VAR is taken as 68°C .

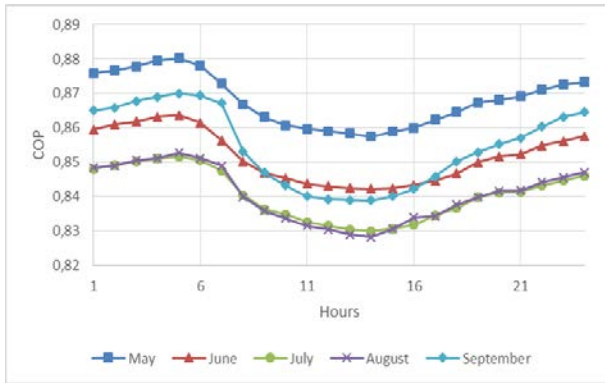


Fig. 4a. Hourly COP variation for single effect VAR system with water-LiBr working fluid

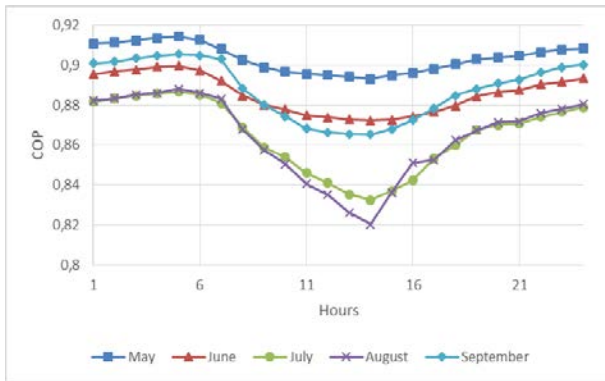


Fig. 4b. Hourly COP variation for single effect VAR system with water-LiCl working fluid

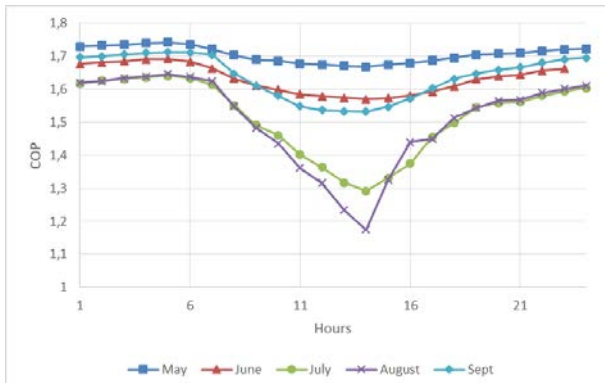


Fig. 4c. Hourly COP variation for double effect VAR system with water-LiCl working fluid

Figure 4a,b,c shows the COP variation of the systems investigated. As it seen, the maximum COP is obtained as 1.74 at 05.00in May for double effect VAR system with water-LiCl working fluid. Double effect VAR system with water-LiCl working fluid is better than the other systems in respect of performance. The minimum COP is obtained as 0.8204 at 14.00 in August for single effect VAR system with water-LiCl working fluid. But, generally the performance of water-LiCl

working fluid is better than water-LiBr working. For instance, the performance of water-LiCl working fluid reaches 0.9143 value but the performance of water-LiCl working fluid reaches 0.88.

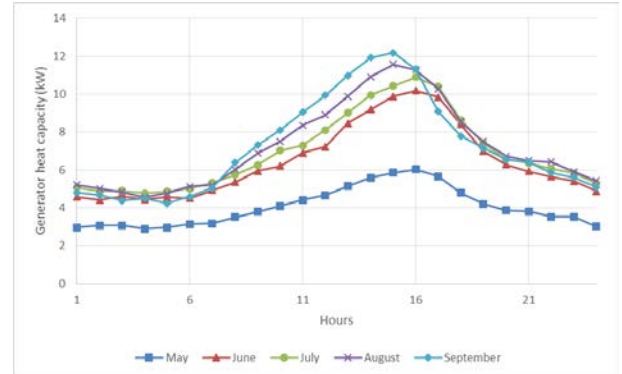


Fig. 5a. Hourly generator heat capacity variation for single effect VAR system with water-LiBr working fluid

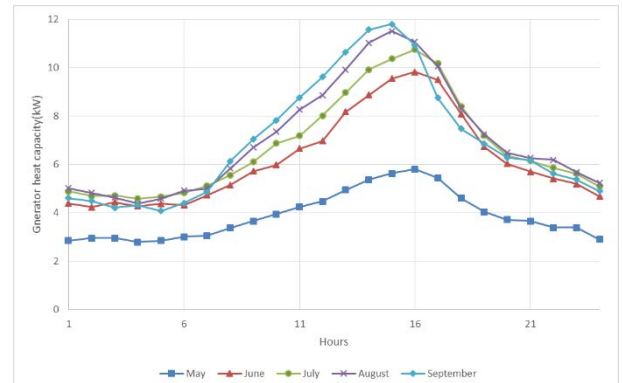


Fig. 5b. Hourly generator heat capacity variation for single effect VAR system with water-LiCl working fluid

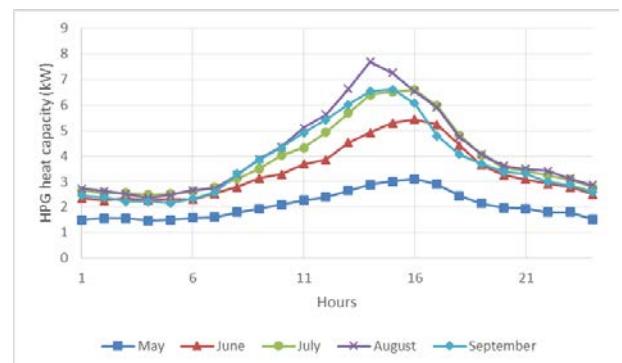


Fig. 5c. Hourly HPG heat capacity variation for double effect VAR system with water-LiCl working fluid

Figure 5 a,b,c shows the supplied heat capacity variation of the systems investigated. As, it is seen double effect VAR System requires less supplied heat than the other

systems. The max required heat is obtained as 7.7 kW at 14.00 in August for double effect VAR System. The max required heat is obtained as 12.185 kW at 15.00 in September for single effect VAR System with water-LiBr working fluid.

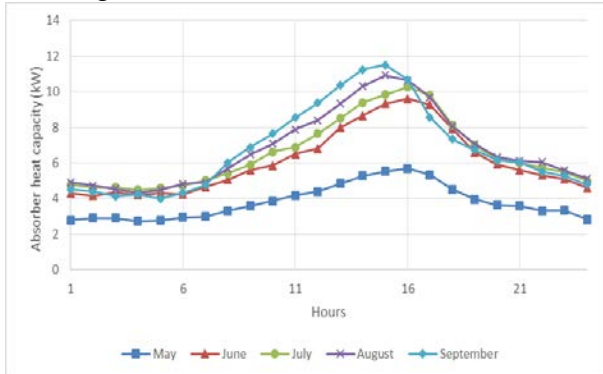


Fig. 6a. Hourly absorber heat capacity variation for single effect VAR system with water-LiBr working fluid

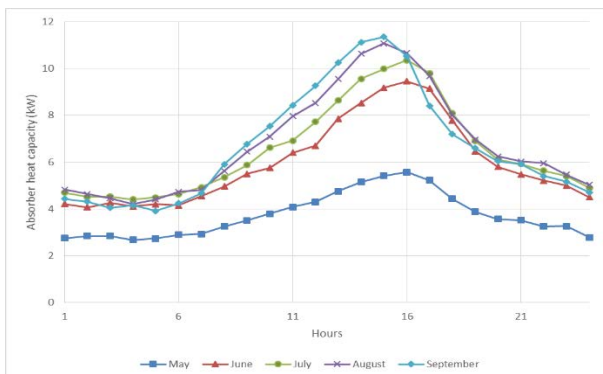


Fig. 6b. Hourly absorber heat capacity variation for single effect VAR system with water-LiCl working fluid

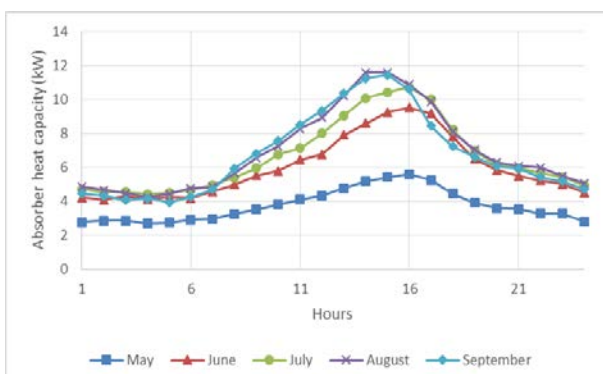


Fig. 6c. Hourly absorber heat capacity variation for double effect VAR system with water-LiCl working fluid

Figure 6 a, b, c shows the absorber heat capacities of the systems investigated. The

absorber capacities are close to each other in every system.

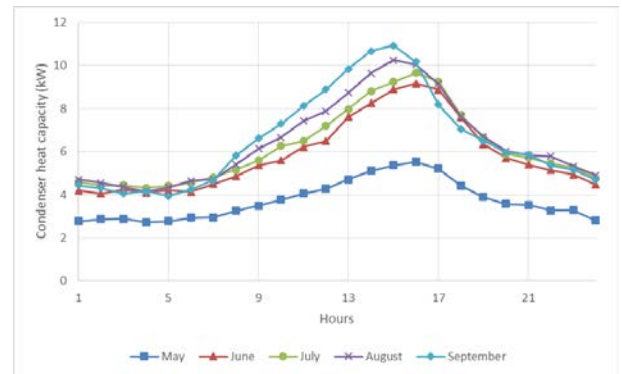


Fig. 7a. Hourly condenser heat capacity variation for single effect VAR system with water-LiBr working fluid

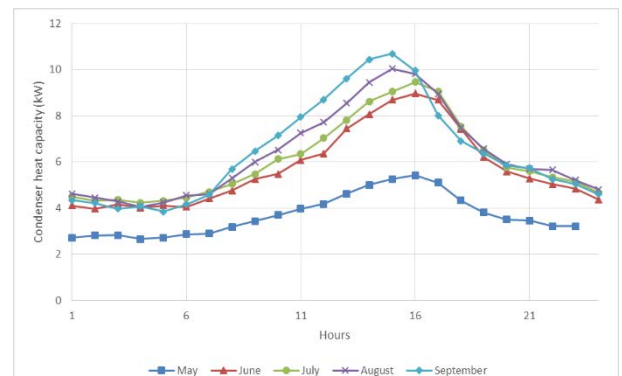


Fig. 7b. Hourly condenser heat capacity variation for single effect VAR system with water-LiCl working fluid

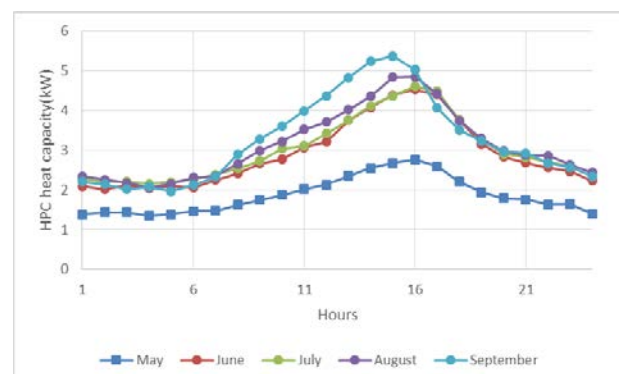


Fig. 7c. Hourly HPC heat capacity variation for double effect VAR system with water-LiCl working fluid

Figure 7 a,b,c,d shows the condenser heat capacities of the systems investigated. In figure 7c shows the variation of HPC heat capacity which is supplied to the LPG.

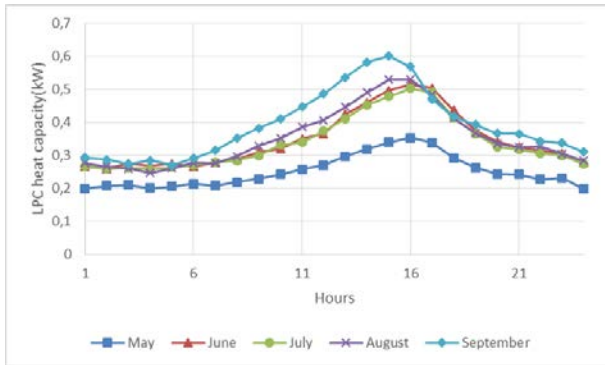


Fig. 7d. Hourly LPC heat capacity variation for double effect VAR system with water-LiCl working fluid

Figure 7 a, b, d shows the heat required to be removed from the system. The highest heat value required to be removed from the system is obtained as 10.93 kW at 15.00 in September for VAR system with water-LiBr working fluid.

3.3. Evacuated Tube Solar Collector Performance

The solar collector area is an important parameter for total cost of the system. The COP of the system, solar collector efficiency effects the required solar collector area for demanding cooling. The calculated solar collector area are shown in Figure 8a, b, c for hour between 08.00 and 17.00 for investigated systems.

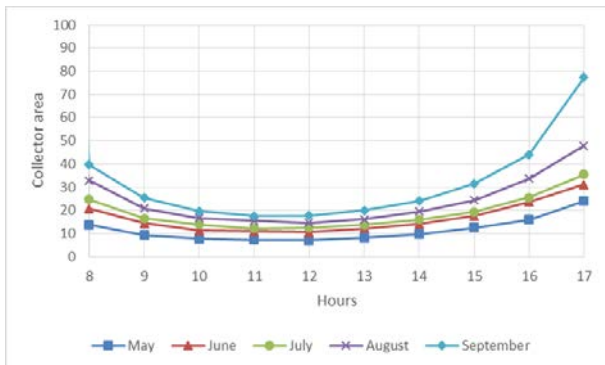


Fig. 8a. Hourly evacuated tube solar collector area for single effect VAR system with water-LiBr working fluid

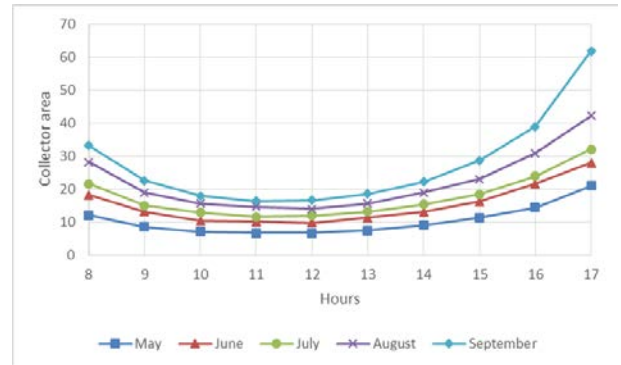


Fig. 8b. Hourly evacuated tube solar collector area for single effect VAR system with water-LiCl working fluid

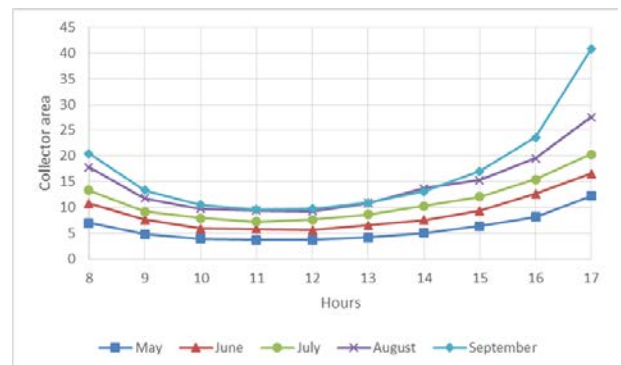


Fig. 8c. Hourly evacuated tube solar collector area for double effect VAR system with water-LiCl working fluid

As it is seen in Figure 8a,b,c, the required solar collector area is lowest for double effect VAR system with water-LiCl working fluid and highest for single effect VAR system with water-LiBr working fluid.

Selected collector area is shown in Table I considering cost and area capacity. Selected collector area supplies energy demanding by VAR systems up to the time of 16.00 in September.

Table 1. Required solar collector area

VAR system with water-LiBr	Single effect VAR system with water-LiCl	Double effect VAR system
44.10 m ²	38.93m ²	23.63m ²

4. CONCLUSIONS

In this study, performance analysis of solar powered VAR is evaluated using hourly

atmospheric air temperature and solar radiation data in Mersin, Turkey. The results showed that double effect solar powered VAR using water-LiCl as working fluid is better, relatively. Minimum required solar collector is obtained as 23.63m², 38.93 m² and 44.1 m² for double effect solar powered VAR using water-LiCl, single effect solar powered VAR using water-LiCl and single effect solar powered VAR using water-LiBr, respectively.

Nomenclature

A	Area
AB	Absorber
CO	Condenser
COP	Coefficient of performance
EV	Evaporator
F	Solar collector efficiency
GE	Generator
h	Enthalpy
HPC	High Pressure Condenser
HPG	High Pressure Generator
I	solar radiation (W/m ²)
LiBr	Lithium Bromide
LiCl	Lithium Chloride
LPC	Low Pressure Condenser
LPG	Low Pressure Generator
SHE	Solution Heat Exchanger
m	mass flow rate
T	Temperature
U	overall heat transfer coefficient (W/m ² K)
X	mass fraction of solution
VAR	Vapor Absorption Refrigeration
Q	Heat capacity (kW)

Greek Letters

τ	Transmission coefficient
α	Absorption coefficient

Subscripts

A	ambient
i	inlet
o	outlet
sol	Solar
w	water

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