Penerbit Akademia Baru

Journal of Advanced Research in Fluid Mechanics and Thermal Sciences

Journal homepage: www.akademiabaru.com/arfmts.html
ISSN: 2289-7879

Thermal Analysis of a Solar Absorption Cooling System with Hot and Cold Storage Tanks

Open Access

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ARTICLE INFO

Article history:
Received 21 June 2018
Received in revised form 15 July 2018
Accepted 20 September 2018
Available online 26 September 2018

ABSTRACT

In the present work, a numerical model is carried out to investigate the performance of a cooling system for a building. The system is operated by a solar powered absorption cooling system using a solar parabolic concentric trough collector integrated with the thermal storage tanks. The system selected a single-effect absorption cooling system has a cooling capacity of 7 kW and working with both hot storage tank in the solar collection loop and cold storage tank in the load loop. A transient system modular computer program has been developed for the cooling system to predict optimal operating conditions of the cooling system working in combination with thermal storage tanks. In this cycle, the hot storage tank is used to store the extra energy and thus to be used to operate the absorption system after the sunset. A cold storage tank is used to store the extra coolant water to cool the building at night time. The building area is 40 m2 and its load is calculated hourly in the steady region of Tikrit University, north of Iraq. The thermal storage tanks volume (cold and hot) ranges (0.2 to 1.25 m3) and this is done for an area range (10 to 30 m2) of parabolic concentric trough collector. The results showed that the typical volume of hot storage tank equals to 0.23 m3 and the cold storage tank volume is 0.57 m3 at a minimum solar collector area 16.6 m2 to condition the building during most of the day hours from 11 AM to 6 PM. The maximum value of the (COP) is 0.65 based on the optimum parameters.

Keywords:
Building cooling load, solar parabolic concentric collector, absorption system, water-lithium bromide solution, ambient temperature.

1. Introduction

The growth of industries, population, modern transportation and many other applications in the world cause an increase in the demand for conventional energy. Producing toxic gases affect the environment and increase thermal retention. Energy source environment-friendly reduces the thermal retention and pollution [1]. Solar energy was found to be the most important renewable energy source. The increase of the electric power consumption in the summer season, especially in the peak time using conventional air conditioners which consume high energy [2]. This solution is

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used to make an impact on the ozone layer to avoid this problem. The researchers deviated from the exploitation of solar energy producing thermal energy that used in the absorption cooling system. This way has been the diversified way to exploit solar power using the types of solar collectors including flat solar, parabolic concentric collectors, and focusing on achieving the best possible energy thermal efficiency [3,4]. These solar collectors were used directly with the absorption cooling system in the generator, which is the first essential part of the system that receives the thermal energy from the solar collectors. It was noted that the use of solar collector directly in the absorption cooling system is appropriated to the circumstances of the air conditioning for long periods [5]. The solar collectors depend on the intensity of the solar radiation during the day, for the processing of the thermal energy of the system. The use of these collectors is not appropriate at the nights as well as during the days where there are many clouds and dust [6]. In order to overcome this problem, a thermal storage tank has been added to continue thermal storage length of the daytime and start preparing - born highly desired temperature of the thermal that stored after the weakness of the intensity of solar radiation [7,8]. There are many studies and designs offered by the researchers regarding using solar energy to generate thermal energy with absorption cooling system works in air conditioning space.

An experimental work reported [9] on the parabolic solar collector with an area of 7.5 m^2 for the hot tank capacity (0.23 m^3). The experiments were done using therminol 55 in the solar detector as a working fluid. The maximum fluid temperature during the sunny day is 116 °C and another sunny day is 212 °C in the mid-day (12 PM) period. The researchers concluded that the thermal losses are more than the acquired heat after (2 PM). The author's recommended not to use the system after 2 PM.

A simulated and experimental studied is done by Al-Alili *et al.*, [10] for the optimal size of the hot storage tank for absorption cooling system working by the solar collector. The experimental solar collector area is 50 m^2 with the cooling capacity of 35 kW and hot tank capacity of $2m^3$. The theoretical solar collector area is less than 65 m^2 with an absorption cooling capacity system of (35 kW) and a tank capacity range of (0.1 to 0.5 m^3). The result showed an optimal hot tank volume capacity of $0.1m^3$. The solar-power absorption air conditioning system which operating by lithium bromide and water solution has been studied by using many types of solar collectors, including flat and focused solar collectors to get the best possible thermal energy [11]. The solar collectors are used directly with the absorption cooling system through the generator, which is the first essential part of the system received the thermal energy from the solar collector. They found that using of solar collector directly in the absorption cooling system is not appropriate because it depends on the intensity of solar radiation during the daytime; also it is not applicable in the nights as well as in the days where there are many clouds and dust. The thermal storage was used to overcome this problem to continue operating all the day and to prepare highly desired temperature of the thermal energy stored, after the weakness of the intensity of solar radiation.

The effect of the cold storage tank in addition to the hot storage tank is investigated [12] to evaluate the possibility of increasing the number of working hours in the system. The cold tank added to store excess coolant to cover the load required in the desired period adapted. An experimental and numerical study is done by Marc [13] to evaluate the absorption cooling system for four classrooms in the university building. The solar collector area is 90 m^2, absorption cooling system capacity is 30 kW, cooling tower capacity is 70 kW, hot tank capacity is 1.5 m^3, and cold tank capacity is 1 m^3. They used 13 units of the coil and fan distribute inside the classrooms. The operation system temperature is 80 °C. The results show that the mean absolute error is 0.5% to evaluate the consumed energy at the full time of a daytime selected in May month. The mean absolute error of production energy was about 2.7% in the evaporator on the same day.

The effect of thermal storage on the solar absorption cooling system for a medium-sized building is studied theoretically [14] when the system designed to provide 50% of the total building cooling load. The solar area collator of 200 m^2 and the cooling energy absorption system is 120 kW. The results show that the change of cold tank volume from 2 to 22 m^3 causing the solar fracture changes from 51% to 57%. In addition, the results show that the solar coolant system is most sensitive to the solar collector area, chillers capacity, hot storage tank volume, and cold storage tank volume.

Complementary to the previous results, the present work offers a thermodynamic analysis of a Solar Absorption Cooling System with Hot and Cold Storage Tanks. The time-dependent cooling load is considered for a selected building. The results of the simulation are in terms of temperature and coefficient of performance.

2. The Working Principle of the System

The solar absorption cooling system consists mainly of a parabolic concentric collector, (lithium bromide and water) absorption chillers, and a cold storage tank. A schematic diagram of the system is shown in Figure 1. The solar energy is gained through the parabolic concentric collector and is accumulated in the hot storage tank [15,16]. The hot water in the hot storage tank is supplied to the absorption chillers then to the generator, which used to boil the water vapor from the solution of (lithium bromide and water). In the condenser, the water vapor cooled down at high pressure then passes across an expansion valve to the evaporator where it evaporated at low pressure. In the evaporator, the working fluid providing cooling to the conditioned space to cover the building cooling load and the reminder used to cool the water in the cold storage tank. The saturated vapor absorbed into the absorber container by lithium bromide solution-water (high percentage lithium bromide) then supplied to the generator through the heat exchanger [17].

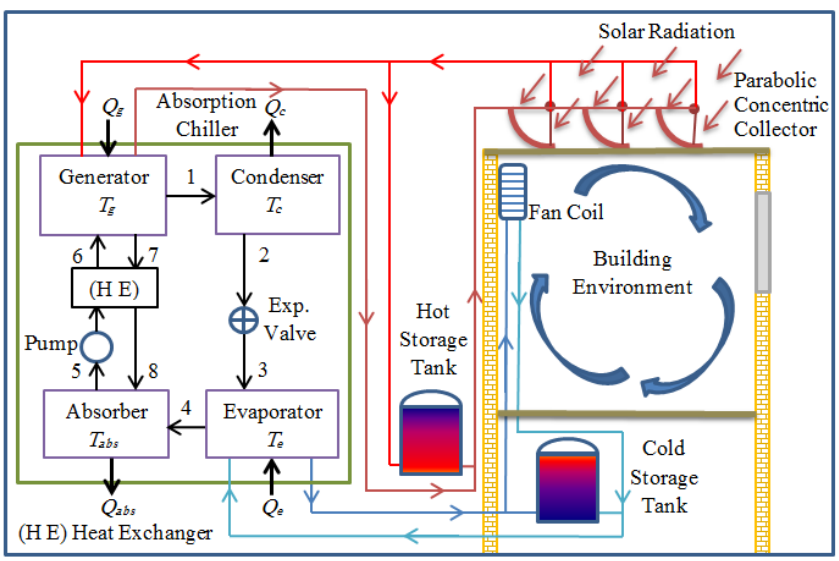


Fig. 1. Schematic diagram of the overall cooling system

The chilled water provides the cooling energy to cover the air conditioning load in the conditioned space. The extra coolant is stored in the cold storage tank, then provided to the conditioned space when the solar energy is not adequate or is not available [18,19]. Figure 2(a) shows the supply processes of the cold storage tank, and Figure 2(b) shows the discharge of this energy to cool the conditioned space.

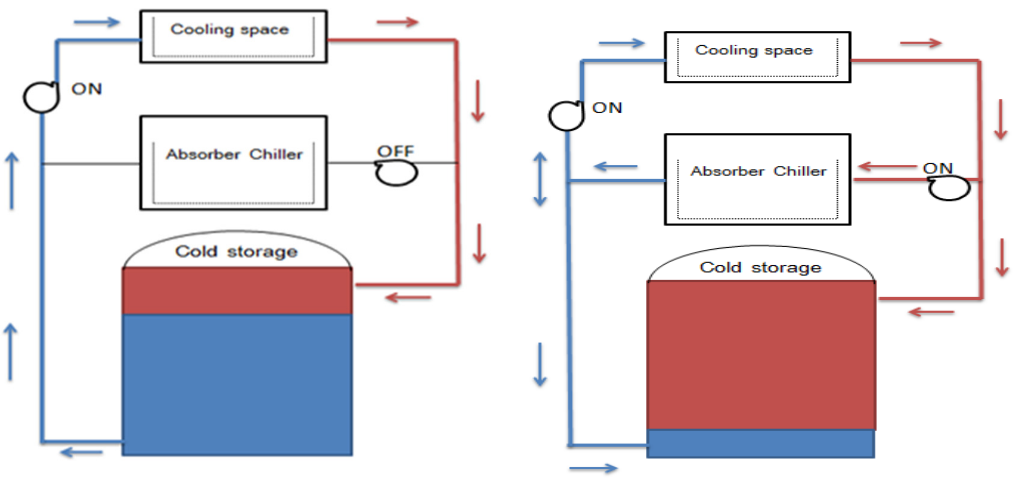


Fig. 2. (a) Cold tank charging (b) Cold tank energy exchange

3. Assumption of the System

The following assumptions were used in the simulation of the various components.

3.1 Solar Thermal Collector

- The fluid properties are constant.
- The flow in the tube of the absorber is steady state.
- The internal pressure of the absorber tube is constant.
- The heat transfer by conduction along absorber tube is neglected.
- The effect of the surrounding temperature change on the absorber tube is negligible.

3.2 Hot Storage Tank

- All sides of the tank are insulated.
- The heat losses in pipes are insignificant.
- The temperature of the tank is homogeneous and less than the liquid evaporation temperature when the heat transferred by convection.
- The primary tank temperature is the primary air temperature.

3.3 Absorption Cooling System

- The pressure is the same in the generator and the condenser.
- The evaporator and the absorber pressures are equal.
- The exit liquid from the generator is saturated.
- The exit liquid from the condenser is saturated.
- The exit steam from the evaporator is saturated.
- The exit liquid solution from the generator in boiling state.
- The exit solution from the absorption in the saturated state.
- The expansion valve is an adiabatic case.

- The pressure drop in the pipeline is negligible

3.4 Cold Storage Tank

- The primary tank temperature is the same as the room temperature which is equal to or less than 28 °C.
- The thermal conduction pipe losses were ignored.
- The temperature of the tank is higher than the freezing temperature of the water.

4. Mathematical Modelling of the Thermal Solar Collector

The solar detector efficiency can be calculated by Garg [20].

$$\eta = \frac{Q_{usf}}{I_{b_{Aape}}} \tag{1}$$

where $I_{b_{Aape}}$is the solar Intensity on the collector area. The actual useful heat Q_{usf} is defined as [20].

$$Q_{\mathrm{usf}} = \mathrm{A_{ape}F_R}\left[H_{rtra} - \frac{A_{rto}}{A_{ape}}U_L\left(T_{w,i} - T_{amp}\right)\right] \tag{2}$$

where A_{ape}, is the absorber area, A_{rto} is the external area of the absorber tube, $T_{w,i}$ is the inlet water temperature, and T_{amp} is the ambient temperature. Solar energy in the absorber tube can be described as follows [20].

$$H_{rtra} = I_b \tau\alpha\rho\gamma \times \cos\theta \tag{3}$$

where I_b is the solar intensity, τ is the transmittance of the absorber tube, α is the absorbance, ρ is the reflectiveness and γ is the objection coefficient of the absorber tube. The overall heat loss coefficient U_L in the glass tube can be calculated as follow [21].

$$U_{\mathrm{L}} = h_{win} + h_{ra,rt-sky} \tag{4}$$

Heat transfer coefficient by the wind speed is defined as follow [21].

$$h_{\mathrm{win}} = 5.7 + 3.8\,V \tag{5}$$

where V is the free stream wind velocity. The heat transfer coefficient by radiation between the absorber tube and surrounding is written as [22].

$$h_{\mathrm{ra,rt-sky}} = \varepsilon_{\mathrm{rt}}\sigma\left[T_{rt} + T_{sky}\right]\left[T_{rt}^2 + T_{sky}^2\right] \tag{6}$$

where σ is the Stefan-Boltzmann constant (5.6697×10^{8}), $\varepsilon_{\mathrm{rt}}$ is the emissivity of the absorber tube. The sky temperature can be calculated from the following [22].

$$T_{sky} = 0.055\,T_{amp}^{1.5} \tag{7}$$

The absorber tube average temperature can be described as follow [23]

$$T_{rt} = T_{w,m} + \frac{\dot{m}c_p\, Q_{usf}(T_{w,o} - T_{w,i})}{h_{c,i}\, \pi\, D_{rt,o}\, L} \tag{8}$$

where ṁ is the water flow rate, c_p is the Specific heat transfer of water, $T_{w,o}$ is the outlet water temperature, $T_{w,i}$ is the inlet water temperature, $h_{c,i}$ is the heat transfer coefficient of water inside the absorber tube, $D_{rt,o}$ is the external diameter of the absorber tube and L is the length of the absorber tube. The water mean temperature is written as [24].

$$T_{w,m} = \frac{T_{w,o} - T_{w,i}}{2} \tag{9}$$

The water temperature at the exit absorber tube can be written as follow [24]

$$T_{w,o} = T_{w,i} + \frac{Q_{usf}}{m \cdot c_p} \tag{10}$$

The heat transfer coefficient inside the absorber tube is written as [25].

$$h_{c,i} = \frac{K_w}{D_{rt,i}}\left[3.6 + \frac{0.0668\left(\frac{D_{rt,i}}{L}\right) Re_w\, Pr_w}{1+0.04\left[\left(\frac{D_{rt,i}}{L}\right) Re_w\, Pr_w\right]^{3/2}}\right] \tag{11}$$

where K_w is the water thermal conductivity, $D_{rt,i}$ is the internal diameter of the absorber tube and Pr_w is the Prandtl number. In this case, Re_w Reynolds number is defined as

$$Re_w = \frac{4\,\dot{m}}{\pi\, \rho_w V_w\, D_{rt,i}} \tag{12}$$

where ρ_w is the water density, and V_w is the water velocity. The efficiency coefficient (F^-) can be described as follows [20].

$$F^- = \frac{{}^{1}/_{\mathrm{U_L}}}{\frac{1}{\mathrm{U_L}} + \frac{D_{rt,o}}{h_{c,i} D_{rt,i}} + \frac{D_{rt,o}\, \ln\left({}^{\boldsymbol{D_{rt,o}}}/_{\boldsymbol{D_{rt,i}}}\right)}{2\, k_{rt}}} \tag{13}$$

where k_{rt} is the thermal conductivity of the absorber tube. The heat removal factor (F_R) is calculated as follow [21].

$$F_{\mathrm{R}} = \frac{\dot{m}c_p}{\mathrm{A_{rto} U_L}}\left[1 - \exp\left(-\frac{\mathrm{A_{rt,i}\, U_L\, F^-}}{m \cdot c_p}\right)\right] \tag{14}$$

The new hot tank temperature $T_{sh,new}$ after one hour is defined as follow [20].

$$T_{sh,new} = T_{sh,old} + \frac{\Delta T_{sh}}{m_{sh} C_{pw}}\left[Q_{usf} - Q_{l1} - (UA_{sh})(T_{sh,old} - T_{amp})\right] \tag{15}$$

where $T_{sh,old}$ is the hot storage tank temperature, UA_{sh} is the thermal losses coefficient of the hot storage tank and Q_{l1} is the building cooling load. The new cold storage tank temperature $T_{sc,new}$ after one hour can be calculated as follow.

$$T_{sc,new} = T_{sc,old} - \frac{\Delta T_{sc}}{m_{sc}C_p}\left[(Q_{acum} - Q_{l,ngh}) + (UA_{sc})\left(T_{sc,old} - T_{amp}\right)\right] \tag{16}$$

where $\boldsymbol{T_{sc,old}}$ is the cold storage tank temperature, $\boldsymbol{UA_{sc}}$ is the thermal losses coefficient of the cold storage tank and $\boldsymbol{Q_{acum}}$ is the accumulated load and $\boldsymbol{Q_{l,ngh}}$is the building load after the sun set.

5. Mathematical Modelling of Thermal Cooling System Absorbance

The energy balance of the thermal cooling system absorbance is shown in Figure 2.

5.1. Generator

The energy balance of the generator can be described as follow [26].

$$Q_g = \dot{m}_{1,w}h_{1,w} + \dot{m}_{8,ss}h_{8,ss} - \dot{m}_{7,ws}h_{7.ws}) \tag{17}$$

where $\dot{m}_{1,w}$ is the mass flow rate of working fluid (water), $\dot{m}_{8,ss}$ is the mass flow rate of water at saturation condition and $h_{1,w}$ is the water enthalpy, and $h_{7.ws}$ is the enthalpy at saturation condition.

5.2. Condensor

The energy balance of the condenser can be described as follows [26].

$$Q_c = \dot{m}_{1,w}\, h_{1,w} - \dot{m}_{2,w}\, h_{2,w} \tag{18}$$

5.3. Expansion Valve

Energy balance of the expansion valve can be described as follow [26]

$$h_{2,w} = h_{3,w} \tag{19}$$

5.4. Evaporator

The energy balance for the evaporator can be described as follows

$$Q_e = \dot{m}_{4,w}\, h_{4,w} - \dot{m}_{3,w}\, h_{3,w} \tag{20}$$

5.5 Pump Solution

$$w_p = \dot{m}_{6,ws}\, h_{6,ws} - \dot{m}_{5,ws}\, h_{5,ws} \tag{21}$$

5.6 Absorber

The energy balance for the absorber can be described as follow

$$Q_{\mathrm{abs}} = \dot{\mathrm{m}}_{4,w} h_{4,w} + \dot{\mathrm{m}}_{10,ss} h_{9,ss} - \dot{\mathrm{m}}_{5,ws} h_{6.ws} \quad (22)$$

and the total energy balance is obtained as

$$Q_g + Q_e = Q_c + Q_{abs} \quad (23)$$

5.7 Heat Exchanger

The heat balance in the heat exchanger can be written as follow

$$Q_{\mathrm{hx}} = \dot{\mathrm{m}}_{8,ss}(h_{8,ss} - h_{9,ss}) = \dot{\mathrm{m}}_{7,ws}(h_{7,ws} - h_{6,ws}) \quad (24)$$

5.8 Coefficient of Performance of the Absorption System (COP).

The coefficient of performance of the absorption system is determined as [26]

$$COP = \frac{Q_e}{Q_g} \quad (25)$$

5.9 Input Data Required

The input data required to simulate the overall the processes are absorption cycle capacity, hourly ambient temperature, hourly wind velocity, hourly solar radiation intensity, generator, condenser, evaporator, and absorber temperatures range and heat exchanger effectiveness.

6. Results and Discussion

The results show the effect of thermal storage tanks (hot and cold) on the performance of the absorption cooling system. The solar parabolic collector is also studied to find the optimum area. These areas must be enough to meet the requirements of the absorption cooling system that achieves proper cooling of the air-conditioning space for the daytime as possible. Figure 3 shows the comparison between the present study and a similar study by Dawood and Yousif [27] to validate the accuracy of the developed computer program. The comparison results give good agreements.

Figure 4 (a, b and c) shows the variation of cold tank temperature hourly during the day at tank volume range (0.65, 0.895 and 1.05 $\mathbf{m^3}$) with and without load for a different area of the solar collector. It can be seen that the decrease in the temperature of the water during the hours of the day, the absorption cooling system stopped when the hot tank temperature be less than 65°C. The cold storage tank operated when the absorption cooling system stopped to cover the load of the conditioned space. The thermal energy extracted by the effect of the cooling load, caused increasing the temperature of cold storage tank hourly during the night after the sunset.

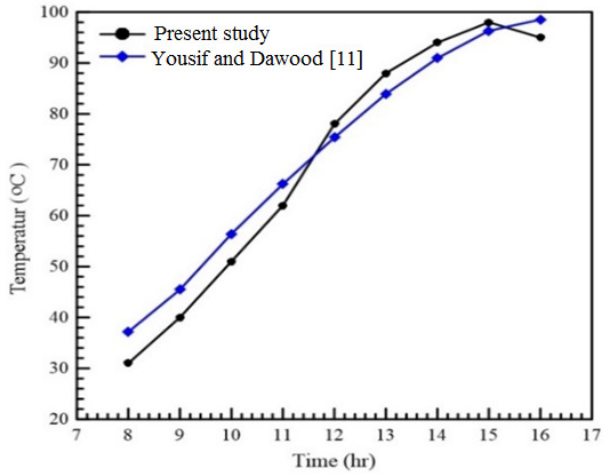
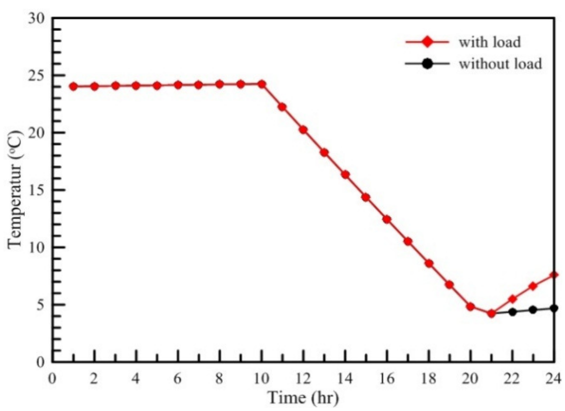
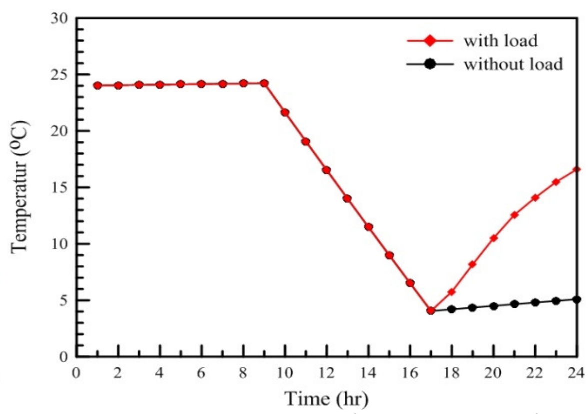


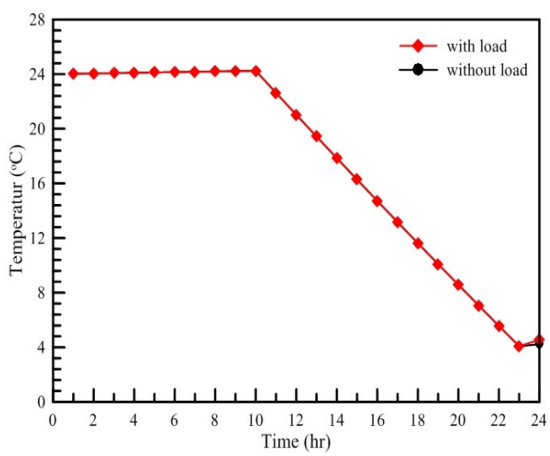
Fig. 3. Comparison of present results with [27] to predict the hot tank temperature variation with working hours for tank volume 1.5 $\mathbf{m^3}$



(a) Volume= 0.65 m^3 & Area=16.6 m^2



(b) Volume = 0.895 m^3 & Area 22.5 m^2

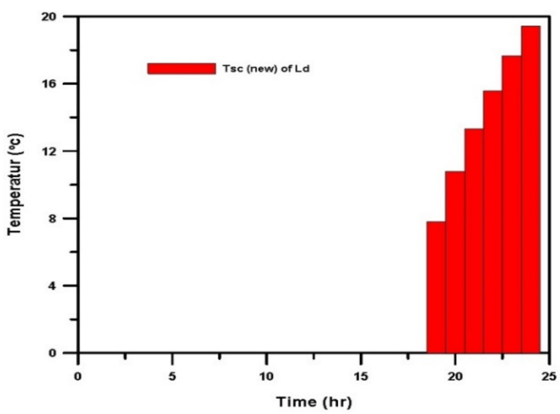


(c) Cold tank volume = 1.05 m^3 and solar collector area 27.6 m^2

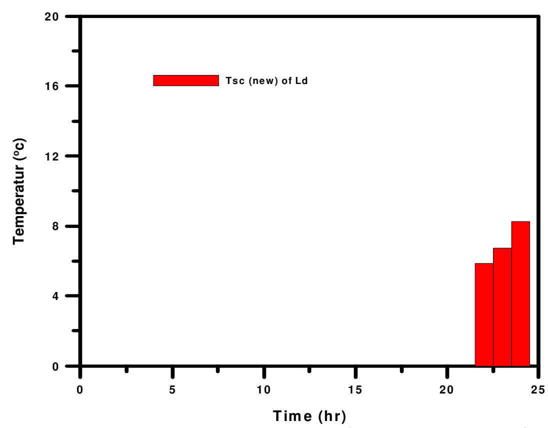
Fig. 4. (a, b and c) The cold tank temperature variation hourly along the day time at different volume with and without load

Figure 5 (a, b and c) shows the effect of temperature on working hours of the cold storage tank at volume range (0.65, 0.895 and 1.05 m^3) and area of parabolic concentric collector range (16.6,22.5 and 27.6 m^2). It can be depicted the effect of the volume of the cold storage tank and the area of a parabolic concentric collector on the number of operating hours. The number of working hours was decreasing with increasing the area of collectors and increasing the volume of the cold storage tank. This effect referred to the increase in hot storage tank energy which assisted to improve the operating time of the absorption cooling system and to still operating after the sunset.

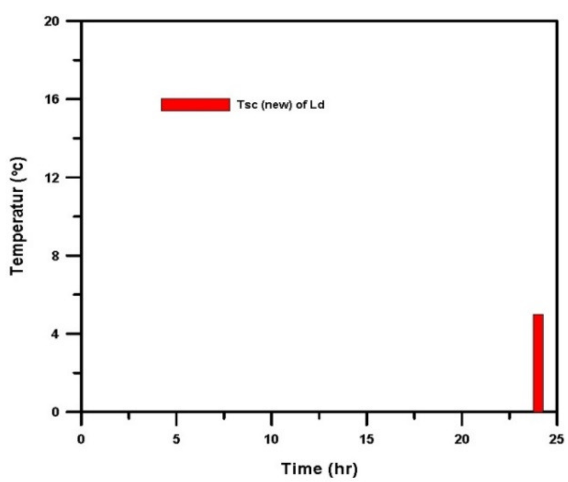
Figure 6 (a, b and c) shows the variation in the temperature of the hot storage tank hourly during the daytime. These done for the conditioned space load and without load at volume range (0.23, 0.75 and 1.25 $\boldsymbol{m^3}$) and for a different area of the solar collector. It can be seen that the temperature increased when the sunrise till 2 PM, then began to decrease gradually by the effect of heat losses to the external environment. The hot tank temperature suddenly dropped, when it started to supply energy to operate the absorption system spatially at sunset.



(a) Volume= 0.65 m^3 & Area=16.6 m^2



(b) Volume = 0.895 m^3& Area 22.5 m^2

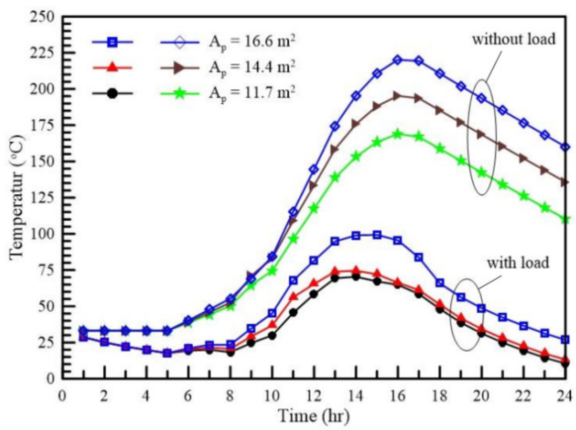


(c) Cold tank volume = 1.05 m^3 and solar collector area 27.6 m^2

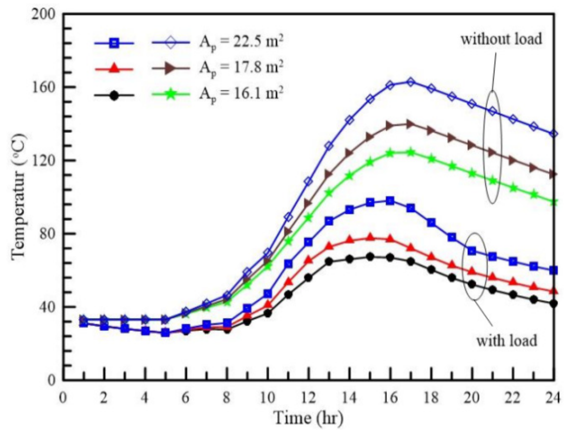
Fig. 5. (a, b and c) The effect of cold tank temperature on working hours at different volume

Figure 7 (a, b and c) shows the effect of temperature on working hours of the hot storage tank at volume range (0.23, 0.75 and 1.25 $\boldsymbol{m^3}$) and different area of the parabolic concentric collector. It can be depicted the effect of the volume of the hot storage tank and the area of the parabolic concentric collector on the number of operating hours. The number of working hours was increasing with increasing the area of the collectors at an expounded volume of the hot storage tank. This effect referred to the increase in hot storage tank energy needed to equip the absorption system. This operation must be at a temperature not less than the operating temperature of the generator of the absorption cycle which equal to (65°C), and to still operating after sunset.

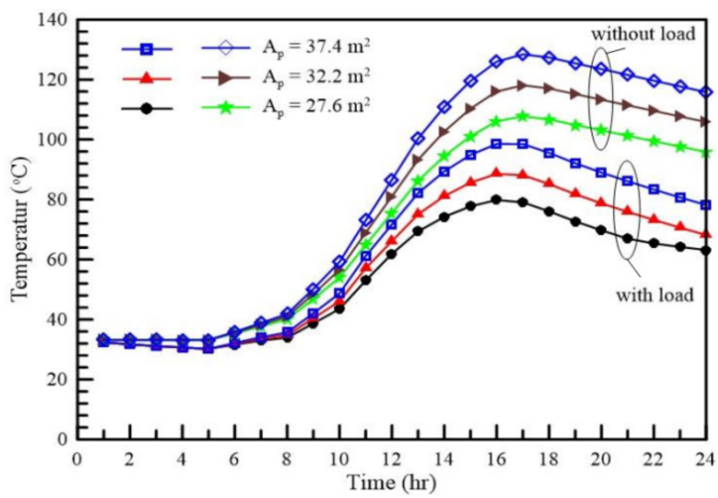
The optimization of the area of the parabolic concentric collector and the time required to operate the absorption cooling system to cover the hourly load requirements for the air-conditioned space [28,29]. These optimizations lead to finding the minimum area of the collector which covers the load. Figure 8 illustrates the effect of the temperature of the generator, condenser, evaporator and absorber on the coefficient of performance (COP). It can be seen that the (COP) increases with increasing of temperature. The optimum value of (COP) is 0.65.



(a) Hot tank volume = 0.23 m^3

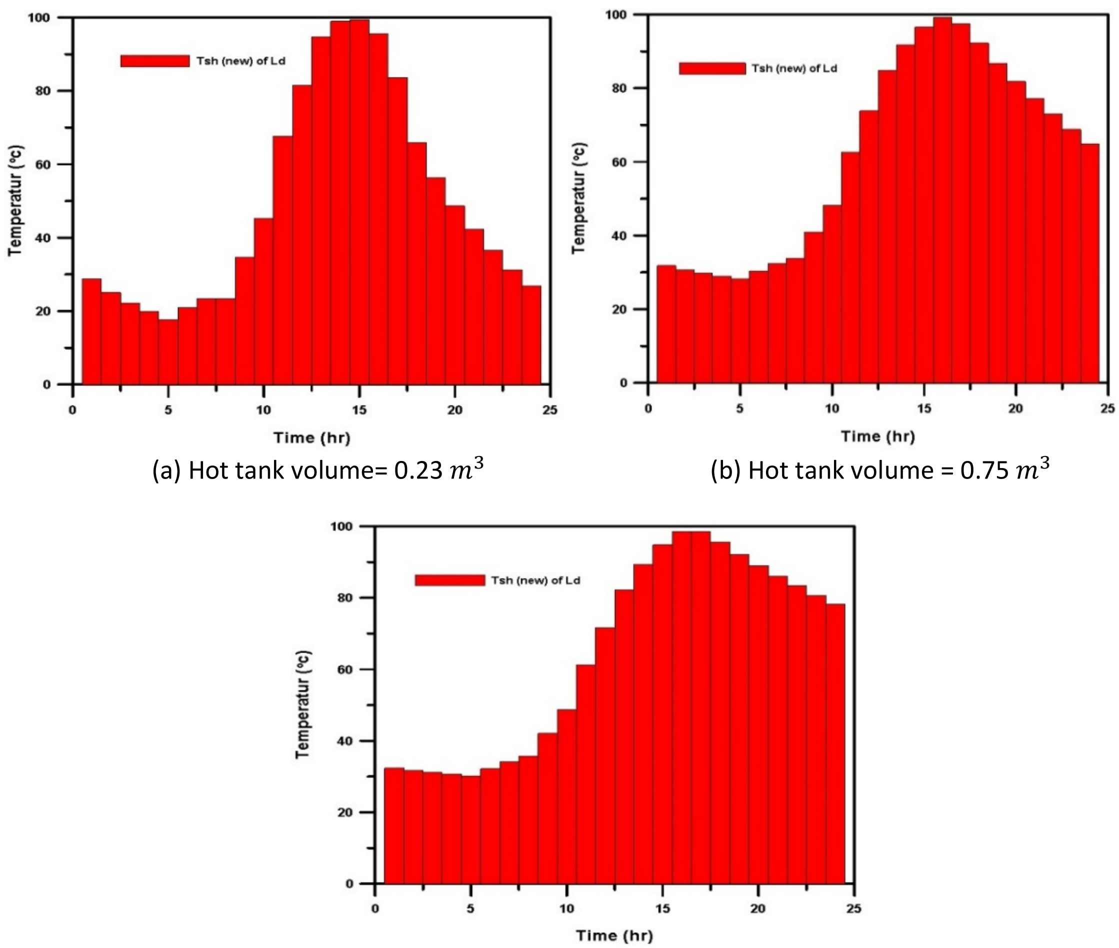


(b) Hot tank volume = 0.75 m^3



(c) Hot tank volume= 1.25 m^3

Fig. 6. (a, b and c) The hot tank temperature variation hourly along the day time at different volume with and without load



(a) Hot tank volume= 0.23 m^3

(b) Hot tank volume = 0.75 m^3

(c) Hot tank volume = 1.25 m^3

Fig. 7. (a, b and c) The effect of hot tank temperature on working hours at different volume

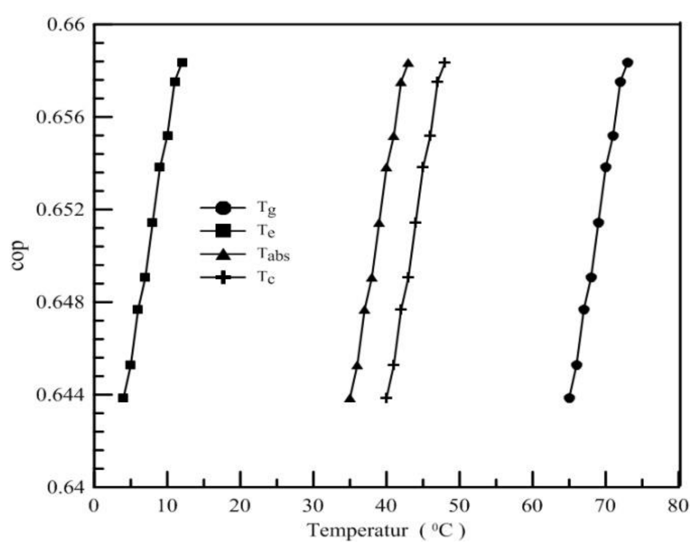


Fig. 8. The temperature for the generator, condenser and evaporator versus coefficient of performance (COP)

7. Conclusion

According to this study, the following conclusions may be deduced. The results give the maximum number of working hours at the minimum solar collector area and minimum volume of the hot and cold tanks. Also, these results match the simulations results. The best value of the hot tank volume is 0.23 m^3 with cold tank volume 0.65 m^3 for a solar parabolic collector area is 16.6 m^2 under cooling system absorber capacity of 7 kW for the summer season in Iraq.

Acknowledgements
The authors are grateful to Tikrit University for providing financial support to complete this project.

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