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Performance Evaluation of a Solar Air Conditioning System

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Abstract

The present study deals with the energy analysis of a new design of the triple generation system of cooling, heating and solar power to supply energy for a hospital complex. The proposed system includes concentrator collectors of the linear Fresnel reflector type with a thermal photovoltaic absorber, which uses solar energy to generate electrical and thermal energy, respectively, for sale to the grid and providing part of the energy for cooling and heating cycles. The proposed cooling system in this structure is a single-effect lithium bromide-water absorption chiller. The gas-burner support heating structure is intended to meet a part of the energy needs of the cooling and heating system non-satisfied by the solar cycle. The system analysis under the thermal load of a hospital complex has been performed annually. The results show that 30% of the system's annual energy is supplied by the solar cycle. In addition, the proposed structure also generates 77.7 MWH of electrical energy. The studied system provides all its energy needs from solar energy on the hottest day from 9:00 to 15:00. The proposed collector has a better performance than the thermal photovoltaic collector without concentrator and the thermal concentrator collector.

Keywords: Thermal Photovoltaic Collector, Concentrator, Triple Generation, Air Conditioning, Energy System, Lithium Bromide- Water Absorption Chiller.
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1. Introduction

The increasing use of energy in the past years and the fact that fossil fuels are the main source of this energy and the lasting effects of using this energy source have caused widespread environmental concerns in recent years [1]. One of the most important factors that cause these environmental concerns is the carbon dioxide generation resulting from the use of these energy sources, so that many developed countries have established strict policies in this field [2]. Accordingly, one of the suitable methods to replace fossil energy sources is the use of renewable energies. As a result, the amount of use of renewable energy has increased approximately 7.46 times from 1965 to 2019 [3].

Solar energy systems with the capacity to directly generate electrical and thermal energy from solar energy are considered one of the main sources of renewable energy. These systems can be used in systems with different dimensions; therefore, the development of these systems is inevitable in order to increase the performance [4]. Photovoltaic panels only have the ability to generate electrical energy and thermal collectors; but the proposed new systems that are a combination of placing photovoltaic panels on heat absorbers are promising due to simultaneous generation of electrical and thermal energy and better performance than the previous types [5].

The use of hybrid systems increases the overall performance of solar systems. Gado et al. [6] studied quantitatively the influencing parameters on the Vapor-compression refrigeration system that is fed by thermal photovoltaic collectors, and an economic analysis has been also performed for this system. Chen et al. [7] analyzed the performance and effective parameters of a night and day solar cooling system with a replaceable insulation sheet.

The use of concentrator in solar systems increases the energy received on the absorber of the collectors. According to the nature of solar cells, the efficiency of which depends on the two parameters of the cell temperature and the amount of radiation, the solar energy received on the cell and the cell temperature have a direct and inverse relationship with the electrical and thermal efficiency of thermal photovoltaic collectors, respectively. Thermal photovoltaic collectors with a concentrator have a higher thermal energy quality than that of without a concentrator [8]. Bamisil et al. [9] analyzed the energy and exergy of a renewable system including thermal photovoltaic collectors with concentrator, wind turbine and biogas for a complete system. Deimi Dasht Beyaz et al. [10] have performed a numerical modeling and validation by conducting laboratory modeling of a system including solar thermal pump and thermal photovoltaic collectors with concentrator, in which nanofluid is used as its working fluid.

The dynamic analysis of the solar system's triple generation of cooling, heating and under-load power in the annual period of a solar system is one of the very important gaps in the studies of this field. Also, the use of thermal photovoltaic collectors as a solar system used in the triple generation structure should be studied, which has not been considered in previous studies. In this research, a solar system with triple generation of cooling, heating and power is studied to meet the air conditioning needs of a hospital complex. The proposed system consists of a linear Fresnel reflector concentrator with a thermal photovoltaic absorber, the structure of the photovoltaic part of which is a three-layer photovoltaic cell type. This system uses a single-effect lithium bromide-water absorption chiller to provide the required cooling of the assembled air conditioning system. In order to prevent the lack of energy supply of the system, two support heating systems are provided in it. The energy analysis of the proposed structure has been carried out annually for the air conditioning of a hospital complex, and finally the results of using the proposed collector and thermal photovoltaic collectors without concentrator and thermal concentrator collector have been compared with each other.

2. Structure Description

Figure 1 shows the schematic of the studied structure. The proposed system is included in the category of triple solar generation systems with the capacity to generate cooling, heating and power. The main components of this structure include concentrator, thermal photovoltaic collector, tanks, support heating, absorption chiller, cooling tower, hospital complex and control system. The concentrator of this structure is a linear Fresnel type with a focusing solar factor of 15, focal length 1.5 m, 2 and 6 m of width and length, respectively [11]. Thermal photovoltaic collectors in this structure have three-layer cells with electrical efficiency of 16% in nominal conditions and temperature and radiation correction coefficients of 0.002 / C° and 0.0074 / kW [12]. The thermal absorber of thermal photovoltaic collectors consists of passing 10 parallel tubes with the same distance through a plate behind the photovoltaic collectors made of aluminum nitride. The task of this thermal absorber is to transfer solar thermal energy to the working fluid of the solar cycle. The working fluid of the solar cycle is a type of synthetic oil of Dynalin Co. with a specific heat capacity of 14.2 kJ/kg Kelvin, a density of 825 kg/m3 and a performance temperature range of 30 C° below zero to 300 C° above zero. The used solar part consists of 15 parallel cycles, each of which includes two collectors in a series circuit. This part of the structure is responsible for the generation of electricity and thermal energy.

When the temperature of the fluid output from the solar cycles is higher than the temperature of the fluid that supplies the energy of the cycle used for air conditioning according to the thermal load of the building, using heat exchangers of total heat transfer coefficient of 700 W/K, the system transfers the energy generated from the solar system to the working fluid.

The thermal load of the building is very important to determine the type and amount of energy required for air conditioning. The building under study in this paper includes two two-story hospital complexes. The total area of this complex is 4479.68 Sqm.

The structure of walls consists of plaster, brick and plaster of U-value of 1.421 W/Sqm. Kelvin, the concrete roof structure has a U-value of 1.021 W/ 0053qm. Kelvin, and the single-layer windows with a thickness of 6 mm [13]. The thermal load of the hospital complex can be calculated by the information of the buildings, and therefore the required amount and type of energy can be obtained. The use of this control system information can determine the type of system (cooling or heating) used, and at the same time, the required data of the return current from the air conditioning system is calculated.

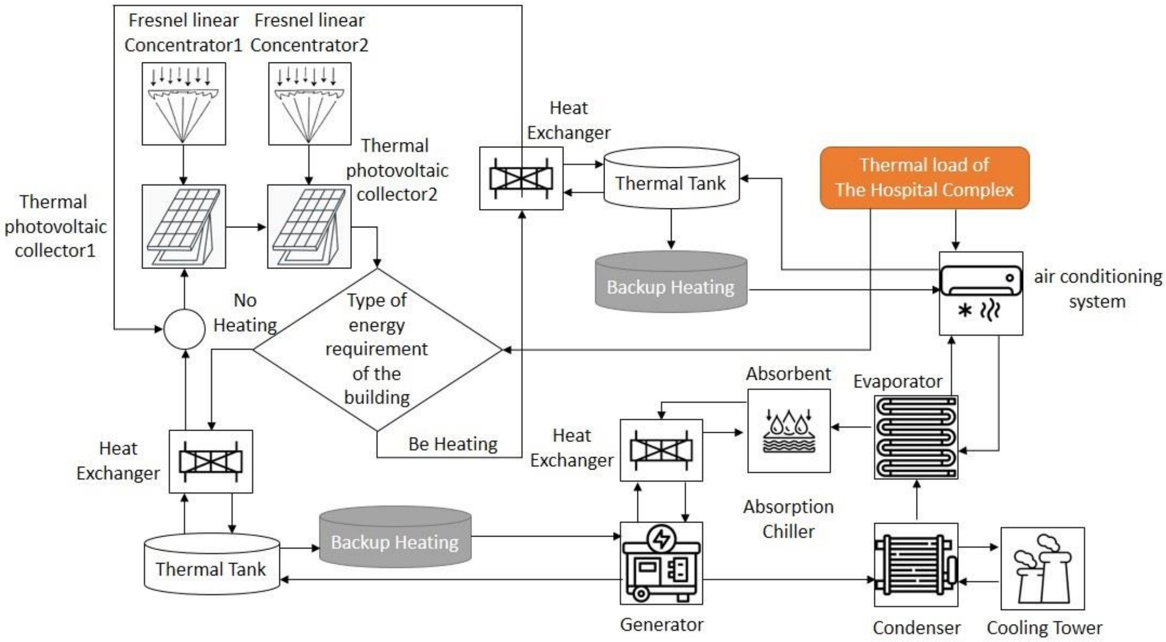

Fig 1. Schematic of the proposed cooling and heating structure of a hospital complex.

Having this information when the complex needs heating, the control system adjusts the current of the fluid of the solar cycle and the storage source by checking the output temperature from the collector and the temperature of the 3 cubic meter source used in the cooling cycle. At this stage, if the temperature of the source is greater than or equal to the temperature required for air conditioning in the heating cycle, the circuit will use the source. Also, this system adjusts the current in the solar cycle and this source in cold seasons according to the temperature of the source of 3 cubic meters feeding the chiller generator and the temperature of the collector. When the chiller generator source temperature is at or above the chiller performance temperature, the system starts using this source. In the conditions where solar energy cannot respond to all the thermal needs of the system, gas-burner support heating is used to generate this thermal energy.

When the cooling cycle is active, an absorption chiller with a capacity of 50 tons of refrigeration made by VIONA Company. With a performance factor of 0.75 is used. A cooling tower of SARAVEL company with a capacity of 75 tons of refrigeration with an electricity consumption of 7.5 hp is used to dispose of the excess energy generated from the absorption cycle. Finally, the performance of the system in the annual interval is dynamically simulated for such a complex located in the west of Tehran by EES and TRNSYS software and compared with other collector structures such as flat plate thermal photovoltaic and thermal with concentrator.

3. Modeling

The simulation of the system is performed using the models developed in TRNSYS and EAS software.

3.1. Assumptions

- The amount of energy transferred to the fluid in pumps is considered insignificant.
- According to the (parallel) structure considered for the collector, all the parallel components with the same structure act similarly.
- The effects of sunlight on the upper part of the collectors are ignored.
- The effect of wind on the collectors is ignored.
- Considering the concentrator structure, the effects of the incident angle of the optical radiation with the absorber surface are ignored.
- The pumps in the system enter the circuit quickly with full power after receiving the activation signal.
- The heat losses of the air conditioning system are ignored.
- Boiler efficiency and combustion efficiency are constant values.
- The boiler does not consume energy when it is not working.

3.2. Thermal Photovoltaic Concentrator and Collector

The concentrator used in this research is of linear Fresnel reflector type. There is no such structure in the TRNSYS model library. Accordingly, using the model provided by Bellos and Tzivanidis, according to Equations 1 to 7, the corresponding code is written in EES software [14].

$$\eta_{opt}\,(\theta_L,\theta_T) = K_L(\theta_L)\times K_T(\theta_T)\times \eta_{opt}\,(0{,}0) \quad (1)$$

Where K_L (θ_L), η_{opt} and K_T (θ_T) are the optical efficiency, longitudinal collision angle modifier and transverse collision angle modifier, respectively. The transverse collision angle is defined using the study of Bushemi et al. as Equation 2 [15].

$$\theta_T = arctan\big(|sin(\gamma_s)|\cdot tan(\theta_z)\big) \quad (2)$$

θ_z and γ_s are the zenith angle and the azimuth angle, respectively. Also, the longitudinal collision angle is defined using Equation 3.

$$\theta_L = arctan\big(cos(\gamma_s)\cdot tan(\theta_z)\big) \quad (3)$$

Having longitudinal and transverse angles, the modifier coefficients of these angles can be expressed as Equations 4 and 5.

$$K_L(\theta_L) = cos(\theta_L) - \frac{F}{L}\sqrt{1+\left(\frac{W}{4F}\right)^2}\cdot sin(\theta_L) \quad (4)$$

$$K_T(\theta_T) = \begin{cases} cos\left(\frac{\theta_T}{2}\right) - \frac{\frac{W}{4}}{F+\sqrt{F^2+\left(\frac{W}{4}\right)^2}}\,sin\left(\frac{\theta_T}{2}\right) & \theta_T < \theta_{T,\,crit} \\ \left[cos\left(\frac{\theta_T}{2}\right) - \frac{\frac{W}{4}}{F+\sqrt{F^2+\left(\frac{W}{4}\right)^2}}\,sin\left(\frac{\theta_T}{2}\right)\right]\cdot\left[\frac{D_w}{W_0}\cdot\frac{cos(\theta_T)}{cos\left(\frac{\theta_T+\varphi_m}{2}\right)}\right] & \theta_T \geq \theta_{T,\,crit} \end{cases} \quad (5)$$

In Equations 4 and 5, φ_m and $\theta_{(T,crit)}$ represent the position angle of the mirror and the solar transverse critical angle, respectively, defined as Equations 6 and 7.

$$\varphi_m = 2arctan\left(\frac{\frac{W}{4}}{F+\sqrt{F^2+\left(\frac{W}{4}\right)^2}}\right) \quad (6)$$

$$\theta_{T,crit} = 94.46 - 2.519.\lambda - 55.71.\lambda^2 - 0.48.\varphi_i + \frac{1.77\varphi_i}{1000} + 1.15.\lambda.\varphi_i \quad (7)$$

Where D_w‘L‘F‘w and W_0 are respectively the width between the center of the first and the last mirror, the focal length, the length of the collector, the distance between the reflectors and the width of the mirrors in meters. The model built based on Equations 1 to 7 in EES software is transferred to the TRNSYS simulation environment using of TRNSYS software model 66 and the necessary modifications regarding the number of variables. This model calculates the amount of radiation focused on the collector by receiving basic information including optical efficiency, zenith angle and azimuth angle. In the next step, this collector must be modeled to calculate the output variables of the thermal photovoltaic collector. In this study, the model presented in the TESS library in TRANSYS is used. Model No. 563 heat absorber, which consists of parallel tubes passed through the absorber, has been considered equivalent to the turbine problem by simplifying, and the solar panels of this model are of the uncoated type. By allocating the inherent parameters of this structure in such a way that it works in sync with the output information of the concentrator model, the suitable model is ready to receive other information such as current rate and inlet temperature, ambient temperature and sky temperature.

3.3. Weather Information

Starting the simulation of the studied model requires information such as temperature, humidity, bubble temperature, radiation amount, zenith angle, and azimuth angle and ground reflection in each time base. TRNSYS software has the capacity to put weather information into the modeling environment using model number 15-2. Metanorm software is one of the best programs for extracting this information. The west of Tehran is the place being studied for this system. The most appropriate information in this situation is the weather data of Mehrabad Airport. Accordingly, using the data of this location in metanorm, the required file of TRNSYS is created to put this data into the simulation environment.

3.4. The Hospital Complex

Simulating a system alone without considering the applied load on that system is important, but this study is the applied loads on that system which shows the actual performance of the system. By simulating the hospital complex model in this research, the loads of this structure are calculated for use in modeling the entire system. In TRNSYS, model number 56 has been developed in line with the simulation of complexes with several separate parts. In the first step, this model requires the user to define the complex and its components. The 3-D design of the buildings of the hospital complex is prepared in SketchUp software according to Figure 2, and then this structure is generated with the appropriate format of Model 56. In the following, building components such as windows, walls and their structural parameters are defined using the relevant section in TRNSYS software. At this stage, this model calculates the cooling and heating load values of the building by using the input weather information from the previous section and the governing equations of heat transfer according to the comfort temperature of 24 C° in winter and 26 C° in summer.

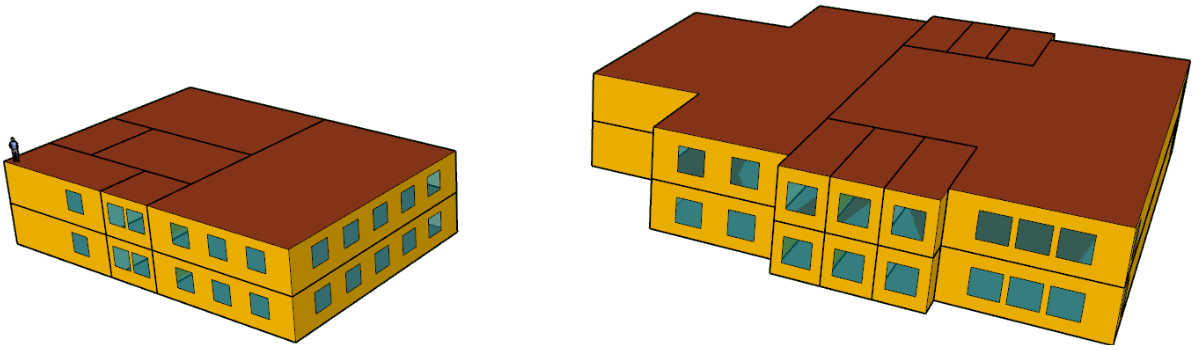

Fig 2. Hospital complex model in SketchUp software

3.5. Storage Resources

Two separate tankers, each with a capacity of 5 cubic meters, are considered as sources of energy storage for the collectors. The model developed in TRNSYS is used to simulate these tanks is model number 158. This model calculates the required parameters of this part of the system by dividing the inner environment of the tanker into smaller parts defined by a specific number and solving the equations of energy and conservation of mass at the same time. This model has two inputs and two outputs.

3.6. Support Heating

When the energy needs of the complex air conditioning system are not fully provided by the energy of the collectors, the supporting heating of the used cycle will provide the remaining energy. Model number 700 in the TRNSYS library is designed to simulate the heating of a gas-burner fluid. This model requires two fuel thermal efficiency coefficients and heat transfer efficiency to the fluid. In this study, for simplicity, these values are chosen to be 95% and 85% respectively [16, 17].

3.7. Air Conditioning System

Modeling of air conditioning is performed in TRNSYS environment using fluid energy exchanger and fluid current control based on complex heat load using model 682.

3.8. Solar Cycles Collimator

In the proposed system, two thermal photovoltaic collectors with a concentrator in series receive a current from the input of the heat exchanger of the cooling or heating cycle. In this case, assuming that 15 of these cycles are the same, to calculate the input current to the cooling or heating cycle, it is enough to multiply the current rate by 15 times.

3.9. Absorption Chiller

In this study, a lithium bromide-water hot water single-effect absorption chiller is used. Two models are used to simulate this part of the system. In the first stage, the model number 107 developed in TRNSYS, and in the next part, the model developed in the EES environment and its use in the TRNSYS environment are examined using the model No. 66 of TRNSYS. In the conditions of using model 107 to a file with dimensionless performance data of the absorption chiller, the ratio of the generated cooling capacity and the input energy according to the parameters of the amount of cooling energy consumed, the temperature set for the cooling output from the chiller, the input temperature from the cooling tower to the chiller and the input temperature to the chiller generator are required. In the second case, using the study of

Kezilkin et al., the governing equations of this section can be expressed as equations 8 to 41 [18]. Figure 3 shows the sketch of the studied absorption chiller.

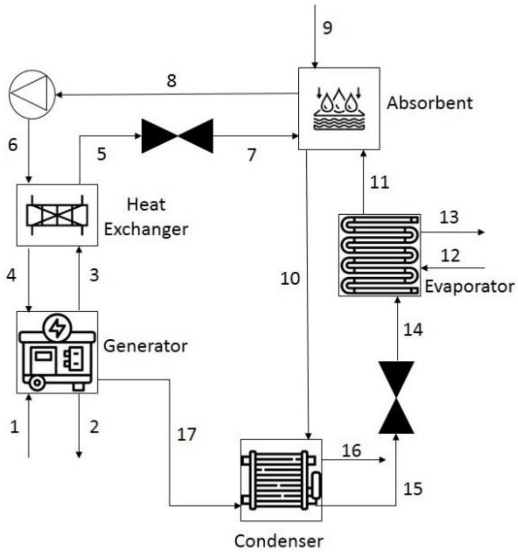

Fig 3. Single effect absorption chiller sketch

In generator, the lithium dilute solution of bromide - water turns into a concentrated solution and vapor by receiving energy. The mass and energy conservation equations of this member are as follows.

$$\dot{m}_4 = \dot{m}_{17} + \dot{m}_3 \quad (8)$$

$$\dot{m}_4 \cdot X_4 = \dot{m}_{17} \cdot X_{14} + \dot{m}_3 \cdot X_3 \quad (9)$$

$$\dot{Q}_G = \dot{m}_1 \cdot h_1 + \dot{m}_2 \cdot h_2 \quad (10)$$

Also, the efficiency of the generator can be expressed as follows.

$$EF_C = \frac{T_1 - T_2}{T_1 - T_{17}} \quad (11)$$

In the above relationships, T, $\dot{Q}_G$, X_m and EF_G represent mass current rate, solution quality, generator heat flux, and temperature and generator efficiency, respectively. By using the logarithmic average temperature difference, the relationship between the heat transfer capacity of the generator and the heat flux of the generator can be expressed as:

$$LMTD_C = \frac{(T_1 - T_3) - (T_2 - T_{17})}{Ln\left(\frac{(T_1 - T_3)}{T_2 - T_{17}}\right)} \quad (12)$$

$$\dot{Q}_G = LMTD_G \cdot UA_G \quad (13)$$

where LMTD and UA are logarithmic mean temperature difference and total heat transfer capacity, respectively.

In the next step, the output concentrated fluid from the generator enters the heat exchanger and gives part of its energy to the output narrow fluid from the absorber. The energy and mass conservation equations of this section are as follows.

$$\dot{m}_3 = \dot{m}_5 X_3 = X_5 \quad (14)$$

$$\dot{m}_6 = \dot{m}_4 X_4 = X_6 \quad (15)$$

$$\dot{m}_6 \cdot h_6 - \dot{m}_4 \cdot h_4 = \dot{m}_3 \cdot h_3 - \dot{m}_5 \cdot h_5 \quad (16)$$

$$\dot{Q}_{HEX} = \dot{m}_3 \cdot h_3 - \dot{m}_5 \cdot h_5 \quad (17)$$

By using the logarithmic mean temperature difference, a relationship between the heat transfer coefficient of the heat exchanger and the heat flux of the generator can be expressed as:

$$LMTD_{HEX} = \frac{(T_3 - T_4) - (T_6 - T_5)}{Ln\left(\frac{(T_3 - T_4)}{T_6 - T_5}\right)} \quad (18)$$

$$\dot{Q}_{HEX} = LMTD_{HEX} \cdot UA_{HEX} \quad (19)$$

The output concentrated solution from the heat exchanger enters the pump.

$$\dot{m}_5 = \dot{m}_7 \quad X_5 = X_7 \quad (20)$$

$$\dot{w}_p = \dot{m}_7 \cdot h_7 - \dot{m}_7 \cdot h_5 \quad (21)$$

$$\dot{w}_p = \dot{m}_7 \cdot v_5 \cdot (P_H - P_L) \quad (22)$$

The output dilute solution from the absorber enters the expansion valve before entering the heat exchanger.

$$\dot{m}_8 = \dot{m}_6 \qquad X_8 = X_6 \qquad h_8 = h_6 \quad (23)$$

In the absorption chiller, the water vapor generated in the evaporator is absorbed by the concentrated solution. This process is an exothermic one. Chiller cooling fluid is used to remove this heat. The equations governing this part of the absorption chiller system are as follows:

$$\dot{m}_8 = m_7 + \dot{m}_{11} \quad (24)$$

$$\dot{m}_8 \cdot X_8 = \dot{m}_7 \cdot X_7 + \dot{m}_{11} \cdot X_{11} \quad (25)$$

$$\dot{Q}_{Ab} = \dot{m}_7 \cdot h_7 + \dot{m}_{11} \cdot h_{11} - \dot{m}_8 \cdot h_8 \quad (26)$$

$$EF_{Ab} = \frac{T_{10} - T_9}{T_8 - T_9} \quad (27)$$

$$LMTD_{Ab} = \frac{(T_8 - T_{10}) - (T_7 - T_9)}{Ln\left(\frac{(T_8 - T_{10})}{T_7 - T_9}\right)} \quad (28)$$

$$\dot{Q}_{Ab} = LMTD_{Ab} \cdot UA_{Ab} \quad (29)$$

In the evaporator, by spraying water on the cooling supply pipes of the chiller, due to the low pressure of this chamber, the water at a low temperature is turned into vapor and provides the required cooling load for the output. The governing equations in this section are as follows:

$$\dot{m}_{14} = \dot{m}_{11} X_{14} = X_{11} \dot{m}_{12} = \dot{m}_{13} \quad (30)$$

$$\dot{Q}_{EV} = \dot{m}_{11} \cdot h_{11} - \dot{m}_{14} \cdot h_{14} = \dot{m}_{12}(h_{13} - h_{12}) \quad (31)$$

$$EF_{EV} = \frac{T_{12} - T_{13}}{T_{12} - T_{11}} \quad (32)$$

$$LMTD_{EV} = \frac{(T_{12} - T_{11}) - (T_{13} - T_{11})}{Ln\left(\frac{(T_{12} - T_{11})}{T_{13} - T_{11}}\right)} \quad (33)$$

$$\dot{Q}_{EV} = LMTD_{EV} \cdot UA_{EV} \quad (34)$$

After leaving the condenser, the water entering the evaporator enters an expansion valve to reduce the pressure and reach the pressure of the absorber and evaporator. Absorption chiller condenser is responsible for condensing water vapor from the generator using absorption chiller cooling fluid.

$$\dot{m}_{15} = \dot{m}_{14} \quad X_{15} = X_{14} \quad h_{15} = h_{14} \quad (35)$$

$$\dot{m}_{15} = \dot{m}_{17} \quad X_{15} = X_{17} \quad (36)$$

$$\dot{Q}_{Co} = \dot{m}_{17} \cdot h_{17} - \dot{m}_{15} \cdot h_{15} = \dot{m}_{10}(h_{16} - h_{10}) \quad (37)$$

$$EF_{Co} = \frac{T_{10} - T_{16}}{T_{10} - T_{15}} \quad (38)$$

$$LMTD_{Co} = \frac{(T_{15} - T_{10}) - (T_{15} - T_{16})}{Ln\left(\frac{(T_{15} - T_{10})}{T_{15} - T_{16}}\right)} \quad (39)$$

$$\dot{Q}_{Co} = LMTD_{Co} \cdot UA_{Co} \quad (40)$$

The performance coefficient of the absorption chiller can be expressed as:

$$COP = \frac{\dot{m}_{14}(h_{11} - h_{14})}{\dot{m}_1(h_1 - h_2)} \quad (41)$$

3.10. Cooling Tower

The cooling tower is responsible for supplying the cooling fluid of the absorption chiller cycle. To model this member of the system, model number 168, developed in TRNSYS for simulating the cooling tower is used with nominal working conditions. Figure 4 shows the studied structure in TRNSYS environment.

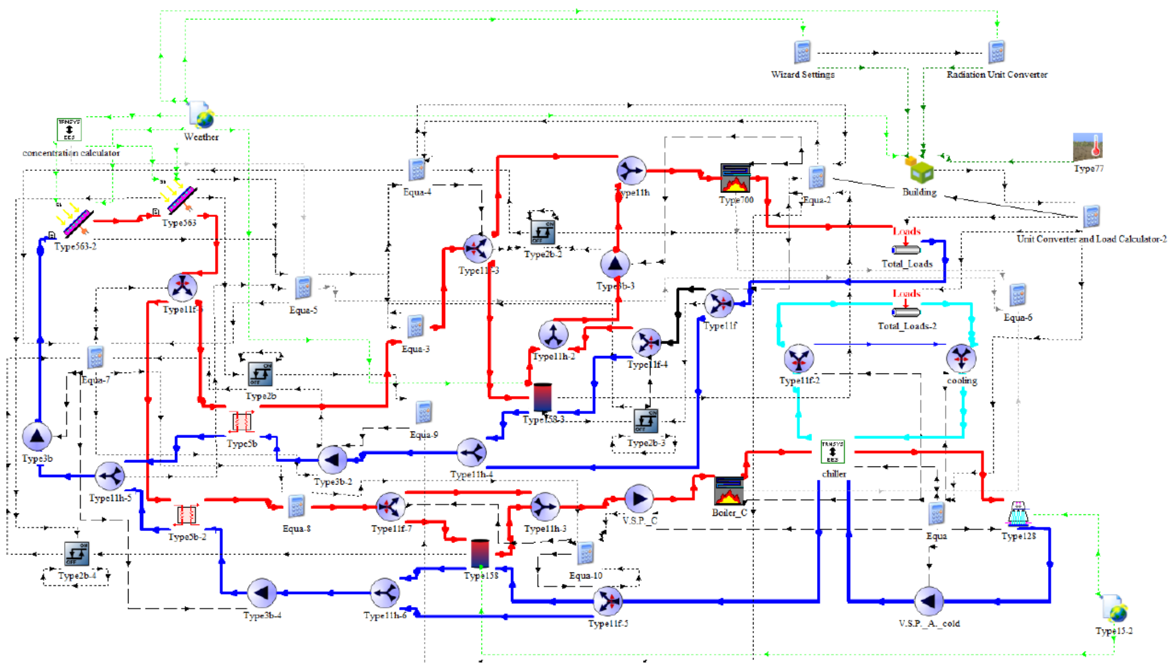

Fig 4. Schematic of solar air conditioning system in TRNSYS.

4. Results and Discussion

4.1. Validation

As far as the authors know, there is no study with a similar structure for all at once validation of the system; therefore, the models used for different parts of the system are validated by using existing studies. The main models requiring validation include the hospital complex, thermal photovoltaic collector with concentrator and absorption chiller.

In their study, Khoda Karmi et al analyzed a similar hospital complex on one floor and without considering the effects of shading of buildings on each other. Table 1 shows the annual heat loads per surface unit (13). From results, due to the lack of access to the weather data used in the source, the maximum difference of 9% between the present results and the results presented in the source is within the acceptable range.

Table 1
Comparison of thermal loads calculated by Khoda Karmi et al. [13]

	Type of study	Cooling Load Kwh/ m^2	Heating Load Kwh/ m^2
The first building	agreeable	62.38	68.68
	source	56.40	62.37
The second building	agreeable	59.72	98.99
	source	59.41	97.99
The whole complex	agreeable	60.76	87.14
	source	58.23	84.02

From the study conducted by Bushmi et al., in which the Fresnel linear collector is used, Fresnel linear collector model is used for validating. Figure 5 shows the collision angle modifier coefficient in the used model and the study model of Bushemi et al. As can be seen, these results are similar [15].

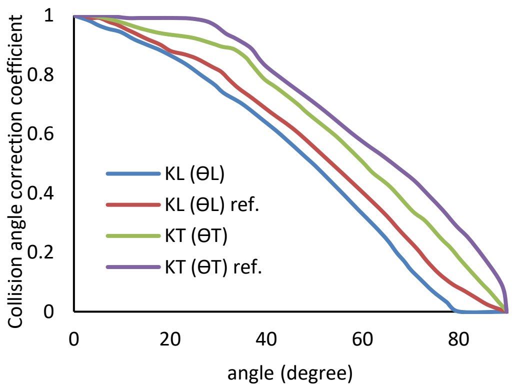

Fig 5. Comparison of the collision angle modifier coefficient of the present model and the source [15].

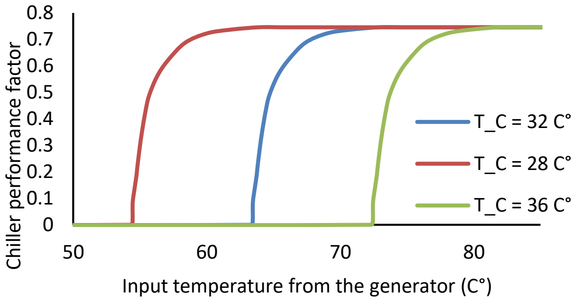

Fig 6. Performance coefficient as a function of the temperature of the absorption chiller generator and cooler or using the data of Gomari [19].

To validate the model used for the absorption chiller, it should be considered that the output data from the two mentioned models are similar. Figure 6 shows the performance coefficient calculated from the models used in this research by applying the performance conditions of the Gomari study on the solar absorption chiller [19]. Examining the given results shows the synchronization of the results of the model used in this research and the source.

4.1. Results of Modeling

Collectors with concentrators are usually known for their high working temperatures. By adding photovoltaic modules to these collectors, thermal and electrical energy are generated simultaneously by this system. Since the generation of electrical energy is expressed as a function of temperature and radiation intensity, information about the temperature of these panels is necessary to analyze the performance of the photovoltaic panels of the system. The temperature of the panels can be synchronized with the output fluid temperature from the collectors with a good approximation. Figure 7 shows the changes in the output temperature of the collectors in the annual interval. Also, by increasing the temperature of the collector, the amount of thermal energy loss with the environment increases and as a result, the thermal efficiency of the system decreases. The maximum thermal and electrical range of the system are 59.3% and 25.2%. In order to prevent excessive heating of the collectors when the system does not need heating or cooling, when the temperature of the collector exceeds the specified range, the absorption chiller enters the circuit and uses energy to reduce the input temperature and thus the output temperature. The higher temperature when using cooling is due to two reasons: the higher temperature required for chiller working and the higher intensity of radiation received on the collectors.

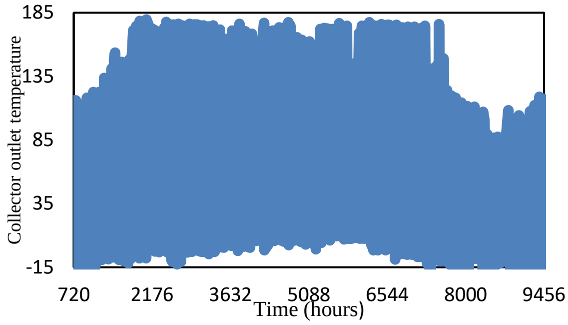

Fig 7. Output temperature from the collectors throughout the year.

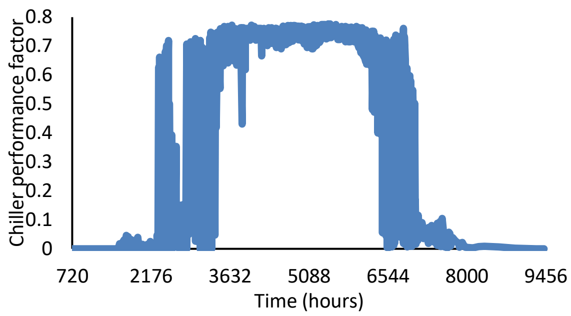

Fig 8. Chiller performance coefficient in the annual period.

The chiller performance factor is another parameter influencing the performance of the overall system in terms of cooling generation. This parameter represents a wide range of chiller performance parameters, including the input temperature from the cooling tower, the temperature and current of the input to the chiller generator, and the load consumed by the building. The nominal coefficient of performance of the used absorption chiller is 0.75 and in this research, the input temperature to the generator is set to be equal to the temperature value in the design conditions, so two other factors, the input temperature from the cooling tower to the chiller and the thermal load used by the building, are the variables affecting the performance coefficient of the chiller. At the beginning of the working of chiller, load variable has the greatest effect on this parameter, and in the middle of the working period, with the increase of the ambient temperature, this parameter is affected highly by the input temperature variable from the cooling tower. Figure 8 shows the changes of this parameter in one year. The maximum coefficient of performance, as expected, is equal to the value of the design conditions and equal to 0.75.

The most important part of the results presented in this simulation is the output information related to energy consumption, generative and thermal loads of the building. Figure 9 shows the information obtained from this modeling for the 5th day of January (a winter day). The generation energy of support heating is higher than the amount of energy consumed due to the coefficient of performance of the gas burner heater. In this research, the coefficient of performance of the heater is assumed to be constant. The energy generated in the solar cycle increases with the increase in radiation and reaches its maximum value in the middle of the day. At this time, the energy required for heating is also reduced by absorbing part of the solar energy received by the building. In the middle of the day, the need for support heating tends to be at minimum. Only 34% of the required energy in the middle of the day is provided by support heating. The maximum need for support heating is on January 13th at 7 am. In this study, it is assumed that the generated electric energy is not used for providing heating and this generated energy is sold directly to the power grid. The electrical energy needed by the system is purchased from the grid. If the electrical energy generated by the collector is used for support heating, the amount of gas-burner support heating is reduced by 68%.

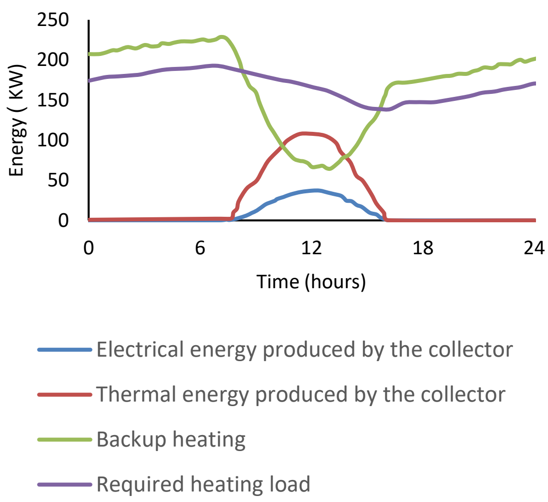

Fig 9. The required heating load of the hospital complex and the energy generated by collector and support heating in a winter day (December 25).

The analysis of the results of the simulator for the cooling cycle in a summer day (July) is presented in Figure 10. From this diagram, it is clear that the proposed system on July 21 from 9:00 to 15:00 will provide all the energy needs of the air conditioning system of the hospital complex. As expected, this system will require support heating during hours when there is little or no solar radiation. On the day under study, the electrical energy generated by the solar cycle is completely sold to the grid and the system's need for electrical energy is met by purchasing electricity from the grid. The maximum cooling load of the complex and the required support heating is on July 21 at 17:00 and 18:00 on the same day. In the current system, the capacity of the tanks is considered small compared to the system under study, therefore, it is not possible to store long-term energy by these tanks.

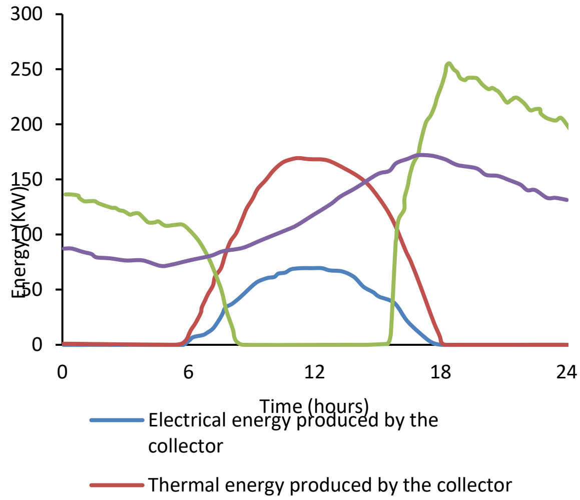

Fig 10. The required cooling load of the hospital complex and the energy generated by collector and support heating in a summer day (July 30).

Table 2 shows the required thermal loads of the complex, the energies generated by collectors and the required heating of the system during the entire annual working period.

Table 2
Thermal load consumption, required support heating energy and collector generating energy in one year

Cycle	Heating (MWh)	cooling (MWh)
Complex load	272.4	389.8
Backup heating required	237.3	367.6
The total thermal energy produced by the collector	200.9	
The total electrical energy produced by the collector	77.7	

Table 3 shows a comparison between the required support heating energy of the proposed system and the system without solar cycle. The information provided shows 20 and 42% energy savings in the heating and cooling cycle. These values are very suitable due to the working of the system all day and night; because about 6 hours of direct solar radiation during the day and night leads to this reduction in consumption. In addition to this, the proposed system will generate 77.7 MW of energy in a year.

Table 3
Comparison of the system using solar energy and without using it

Cycle	Heating (MWh)	cooling (MWh)
Solar cycle system	367.6	237.3
System without solar cycle	457.8	408.9

5. Conclusions

In the present study, the performance of a proposed system for providing the required energy of a hospital complex has been analyzed. The proposed system uses a thermal photovoltaic collector with a concentrator, back-up gas heating and a single-effect absorption chiller to meet the needs of a hospital complex with two separate two-story buildings located in the west of Tehran. A linear Fresnel reflector is the concentrator structure used in the proposed system. Three-layer photovoltaic cells are used in this collector. The absorption chiller used in this system is a single effect lithium bromide-water absorption chiller.

Gas burner support heating is installed to provide energy to the system when the solar part is not sufficient for system. The system was modeled using a combination of two programs EES and TRNSYS in a period of one year and one month and the data of the first month has been ignored to fix the errors of the boundary conditions. The results have been validated using information from research articles. The most important results of this simulation can be summarized as follows.

- The proposed system using solar energy required 80 and 56% of the energy needed for the system without using this energy per year, respectively.
- The proposed system generates 77.7 MWH of electrical energy in 2017, which is sold to the grid according to the government's incentive plan for generating electrical energy from solar energy.
- Total thermal and electrical energy generated by the system during the year is 278.6 MWH.
- In the summer days when there is a need for heating, the system is able to provide all the energy required by the system for working of the absorption chiller between 9:00 and 15:00, if there is enough direct radiation.
- The proposed system gives the energy generated from the collectors to the environment when there is no need for cooling and heating.
- The proposed system requires a total of 30% less energy for air conditioning, and

by using electric energy as heating, this value reaches 39%.

- The use of a thermal photovoltaic collector with a concentrator generates more than three times and 18% more energy compared to a thermal photovoltaic collector without a concentrator and a thermal concentrator collector with the same covering surface.

6. Future Studies

- Designing the current controller of the collectors to increase the efficiency of the system.
- Analysis of the effects of using other types of cooling towers and absorption chillers on system performance.
- Using other concentrator structures in the system.
- Economic analysis of the proposed system.
- Studying the performance of the system in different places.
- Energy analysis of the proposed system.

List of indexes

English index

Absorbing	Ab
Condenser	Co
Chiller performance coefficient (dimensionless)	COP
Distance between reflectors (m)	D
Yield (dimensionless)	EF
Evaporator	Ev
Focal length (m)	F
Generator	G
Enthalpy (J/g)	h
Heat exchanger	HEX
Longitudinal collision angle modifier (Dimensionless)	$K_L(\Theta_L)$
Transverse collision angle modifier (Dimensionless)	$K_T(\Theta_T)$

Greeks

Azimuth angle	(degrees)
Optical efficiency	(dimensionless)
Solar transverse critical angle	(degrees)
Collector length (m)	L
Logarithmic mean temperature difference (K)	LMTD
Mass current rate (kg/s)	th
Pressure in the generator (kPa)	P_H
Pressure in the absorbing (kPa)	P_L
Heat flux (KW)	Q
Temperature (C°)	T
Total heat transfer capacity (KW/K)	UA
Width between the center of the first and last mirror (m)	W
Width of mirrors (m)	W_a
Pump work (KW)	$\dot{w}_p$
Solution quality (dimensionless)	X
Zenith angle (degrees)	Θ_z
Solution specific volume (m3/kg)	v
Mirror position angle (degrees)	φ_m

References

[1] Martins T, Barreto AC, Souza FM, Souza AM. Fossil fuels consumption and carbon dioxide emissions in G7 countries: Empirical evidence from ARDL bounds testing approach. Environmental Pollution. 2021; 291:118093.

[2] Yirong Q. Does environmental policy stringency reduce CO2 emissions? Evidence from highpolluted economies. Journal of Cleaner Production. 2022; 341:130648.

[3]. Ritchie H, Roser M. Renewable energy. Our World in Data. 2020.

[4] Akrofi MM, Okitasari M. Integrating solar energy considerations into urban planning for low carbon cities: A systematic review of the state-of-the-art. Urban Governance. 2022.

[5] Jidhesh P, Arjunan TV, Gunasekar N. Thermal modeling and experimental validation of semitransparent photovoltaic- thermal hybrid collector using CuO nanofluid. Journal of Cleaner Production. 2021; 316:128360.

[6] Gado MG, Megahed TF, Ookawara S, Nada S, El-Sharkawy II. Potential application of cascade adsorption-vapor compression refrigeration system powered by photovoltaic/thermal collectors. Applied Thermal Engineering. 2022; 207:118075.

[7] Chen H, Li Z, Sun B. Performance evaluation and parametric analysis of an integrated diurnal and nocturnal cooling system driven by photovoltaic-thermal collectors with switchable film insulation. Energy Conversion and Management. 2022; 254:115197.

[8] Afzali Gorouh H, Salmanzadeh M, Nasseriyan P, Hayati A, Cabral D, Gomes J, et al. Thermal modelling and experimental evaluation of a novel concentrating photovoltaic thermal collector (CPVT) with parabolic concentrator. Renewable Energy. 2022; 181:535-53.

[9] Bamisile O, Huang Q, Li J, Dagbasi M, Desire Kemena A, Abid M, et al. Modelling and performance analysis of an innovative CPVT, wind and biogas integrated comprehensive energy system: An energy and exergy approach. Energy Conversion and Management. 2020; 209:112611.

[10] Deymi-Dashtebayaz M, Rezapour M, Farahnak M. Modeling of a novel nanofluid-based concentrated photovoltaic thermal system coupled with a heat pump cycle (CPVT-HP). Applied Thermal Engineering. 2022; 201:117765.

[11] Moaleman A, Kasaeian A, Aramesh M, Mahian O, Sahota L, Nath Tiwari G. Simulation of the performance of a solar concentrating photovoltaic-thermal collector, applied in a combined cooling heating and power generation system. Energy Conversion and Management. 2018; 160:191-208.

[12] Helmers H, Schachtner M, Bett AW. Influence of temperature and irradiance on triple-junction solar subcells. Solar Energy Materials and Solar Cells. 2013; 116:144-52.

[13] Khodakarami J, Knight I, Nasrollahi N. Reducing the demands of heating and cooling in Iranian hospitals. Renewable Energy. 2009; 34(4):1162-8.

[14] Bellos E, Tzivanidis C. Development of analytical expressions for the incident angle modifiers of a linear Fresnel reflector. Solar Energy. 2018; 173:769-79.

[15] Buscemi A, Panno D, Ciulla G, Beccali M, Lo Brano V. Concrete thermal energy storage for linear Fresnel collectors: Exploiting the South Mediterranean's solar potential for agri-food processes. Energy Conversion and Management. 2018; 166:719-34.

[16] Bonaros V, Gelegenis J, Harris D, Giannakidis G, Zervas K. ANALYSIS OF THE ENERGY AND COST SAVINGS CAUSED BY USING CONDENSING BOILERS FOR HEATING DWELLINGS IN GREECE2013.

[17] Jayamaha L. Energy-Efficient Building Systems: Green Strategies for Operation and Maintenance: Green Strategies for Operation and Maintenance: McGraw Hill Professional; 2006.

[18] Kızılkan Ö, Şencan A, Kalogirou SA. Thermoeconomic optimization of a LiBr absorption refrigeration system. Chemical Engineering and Processing: Process Intensification. 2007; 46(12):1376- 84.

[19] Gomri R. Simulation study on the performance of solar/natural gas absorption cooling chillers. Energy Conversion and Management. 2013; 65:675-81.