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Simulation and optimization of a LiBr solar absorption cooling system with evacuated tube collectors

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Abstract

Solar radiation is a clean form of energy, which is required for almost all natural processes on earth. Solar-powered air-conditioning has many advantages when compared to a conventional electrical system. This paper presents a solar cooling system that has been designed for Malaysia and similar tropical regions using evacuated tube solar collectors and LiBr absorption unit. The modeling and simulation of the absorption solar cooling system is carried out with TRNSYS program. The typical meteorological year file containing the weather parameters for Malaysia is used to simulate the system. The results presented show that the system is in phase with the weather, i.e. the cooling demand is large during periods that the solar radiation is high. In order to achieve continuous operation and increase the reliability of the system, a 0.8 m^3 hot water storage tank is essential. The optimum system for Malaysia's climate for a 3.5 kW (1 refrigeration ton) system consists of 35 m^2 evacuated tubes solar collector sloped at 20°.

Keywords: Air-conditioning; Solar energy; TRNSYS

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1. Introduction

Absorption cooling is one of the first and oldest forms of air-conditioning and refrigeration systems used. The system uses thermal energy to produce cooling and thus solar energy, waste heat and other forms of low grate heat can be employed. As no CFCs are used, absorption systems are friendlier to the environment. Absorption is the process of attracting and holding moisture by substances called desiccants. Desiccants are sorbent materials that have an ability to attract and hold other gases or liquids and have a particular affinity for water. During absorption the desiccant undergoes a chemical change as it takes in the moisture, for example, table salt, which changes from a solid to a liquid as it absorbs the moisture. The characteristics of the binding of desiccants to moisture make the desiccants very useful in chemical separation processes [1]. Absorption air-conditioning systems are similar to vapor compression air-conditioning systems, but differ in the pressurization stages. In general an absorbent in the low-pressure side absorbs an evaporating refrigerant (H_2O). The most usual combinations of chemical fluids used include lithium bromide–water ($LiBr–H_2O$), where water vapor is the refrigerant, and ammonia–water ($NH_3–H_2O$) system where ammonia is the refrigerant.

Computer modeling of thermal systems presents many advantages. The most important are the elimination of the expense of building prototypes, the optimization of the system components, estimation of the amount of energy delivered from the system, and prediction of temperature variations of the system.

Various researchers presented recently modeling and simulation studies of solar cooling and air-conditioning systems.

Ghaddar et al. [2] presented the modeling and simulation of a solar absorption system for Beirut. The results showed that for each ton of refrigeration it is required to have a minimum collector area of 23.3 m^2 with an optimum water storage capacity ranging from 1000 to 1500 l when the system operates solely on solar energy for about 7 h per day. The monthly solar fraction of total energy used for cooling is determined as a function of solar collector area and storage tank capacity. The economic analysis performed showed that the solar cooling system is marginally competitive only when it is combined with domestic water heating.

Hammad and Zurigat described the performance of a 1.5 ton solar cooling unit. The unit comprise a 14 m^2 flat-plate solar collector system and five shell and tube heat exchangers [3]. The unit was tested in April and May in Jordan. The maximum value obtained for actual coefficient of performance was 0.55.

Florides et al. [4] modeled a complete system, comprised of a solar collector, a storage tank, a boiler and a LiBr–water absorption refrigerator, which can cover a typical house load for the whole year. The TRNSYS program was used to model the system, together with the weather values of a typical meteorological year (TMY) file for Nicosia, Cyprus. Using this approach, a system optimization was performed in order to select the right equipment, i.e. the collector type, the storage tank volume, the collector slope angle and area and the optimum setting of the auxiliary boiler thermostat. The collector area was decided by performing an economic analysis of the system. Also the long-term integrated system performance and the dynamic system's behavior was evaluated.

The objective of this work is to study the use evacuated tube solar collectors to collect solar radiation and use it as source of energy of a LiBr–H_2O absorption air-conditioning system. The analysis and optimization of the system components is carried out with the TRNSYS software, which is also used to simulate the best system with respect to the optimum coefficient of performance (COP). The coefficient of performance (COP) is defined as the ratio of the cooling effect to the heat input.

2. System description

The basic energy flows of a solar-powered cooling system are shown schematically in Fig. 1. The collector receives energy from sunlight. The energy is then transferred through high temperature energy storage reservoir to the refrigeration system. In the absorption system, heat is taken from the evaporator or cold storage reservoir (from the water that flows in the evaporator and comes out as chilled water), which evaporate the refrigerant as water vapor. The water vapor then passes into the absorber and is being absorbed by the lithium bromide in the absorber to form a weak solution. The weak solution is then pumped into the generator. In the generator, the water from the weak solution is separated to form water vapor and strong lithium bromide solution. The generator requires heat from the solar collector system to separate the water vapor from the solution. The water vapor thus generated is at high temperature and pressure. It is then passed to the condenser where heat is removed and the vapor cools down to form a liquid. The liquid water at high pressure is passed through the expansion valve to the low-pressure area in the evaporator where the water is turned into vapor again by drawing heat from the entering water in the tube heat exchanger. The vapor then passes to the absorber again and the process is repeated. The strong solution from the generator is pumped through a heat exchanger to

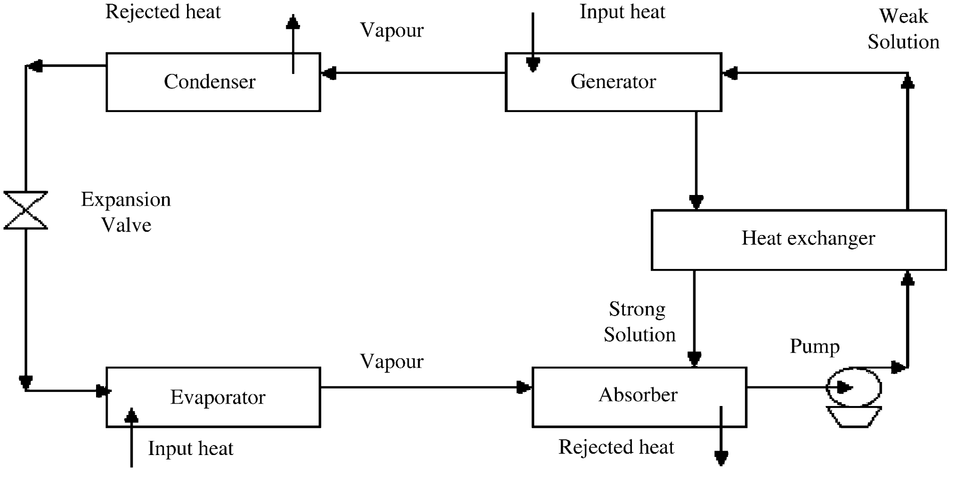

Fig. 1. The basic principle of the absorption air-conditioning system.

the absorber and the weak solution from the absorber is passed through the same heat exchanger to the generator. The heat is removed from the system by cooling water, which passes through the condenser and the absorber to a cooling tower where the heat is dissipated to the environment. In the case that the sun is not shining, the generator heating energy may be supplied from an auxiliary heat source such as electricity or conventional boiler to run the system.

There are many types of solar collectors, which are used in air-conditioning applications. These can be flat-plate collectors, evacuated tube collectors or compound parabolic collectors. In the present study, advanced evacuated tube collector with selective surface, which can be effective collectors for cooling applications, are considered.

Evacuated tube collectors are highly efficient as they are made of an absorber pipe enclosed within a larger glass tube and the space between the glass and the absorber is evacuated. The absorber pipe may also be attached to a black copper fin that fills the tube (absorber plate). The performance equation of the collector considered is given by

$$\eta = 0.82 - 7.884\frac{T_\mathrm{i} - T_\mathrm{a}}{I_\mathrm{T}} \tag{1}$$

where

T_i inlet temperature of fluid to collector (°C)
T_a ambient temperature (°C)
I_T total incident radiation on a flat surface per unit area (kJ/h m^2)

Air collectors are not cost-effective for solar cooling applications because the heat exchange surface areas required are very large. The complete schematic solar cooling diagram is presented in Fig. 2. This schematic diagram represents the proposed conceptual design for the system under investigation. The system consists of two important parts: (a) the solar collector system and (b) the absorption cooling system. In the present application, evacuated tube solar collectors are used to produce the vapor in the generator. An auxiliary heater is used when the solar heating is not sufficient. A storage tank is also needed to store hot water in order to increase the efficiency of system and allow the system to operate when there is no sunshine but heat is available in the storage tank.

3. Program description and assumptions

TRNSYS program is employed for the modeling and simulation of both the solar collector system and the absorption cooling system based on Malaysia's climatic data. In Fig. 3, the steps for modeling and simulation of the system are described. Initially, the metrological weather data for the proposed site are created, then the suitable components are set-up in the deck file. This is followed by the determination of the parameters, inputs and variables and the execution of the program. Subsequently, the results are analyzed based on charts and diagrams produced by the program, thus the optimum components may be selected.

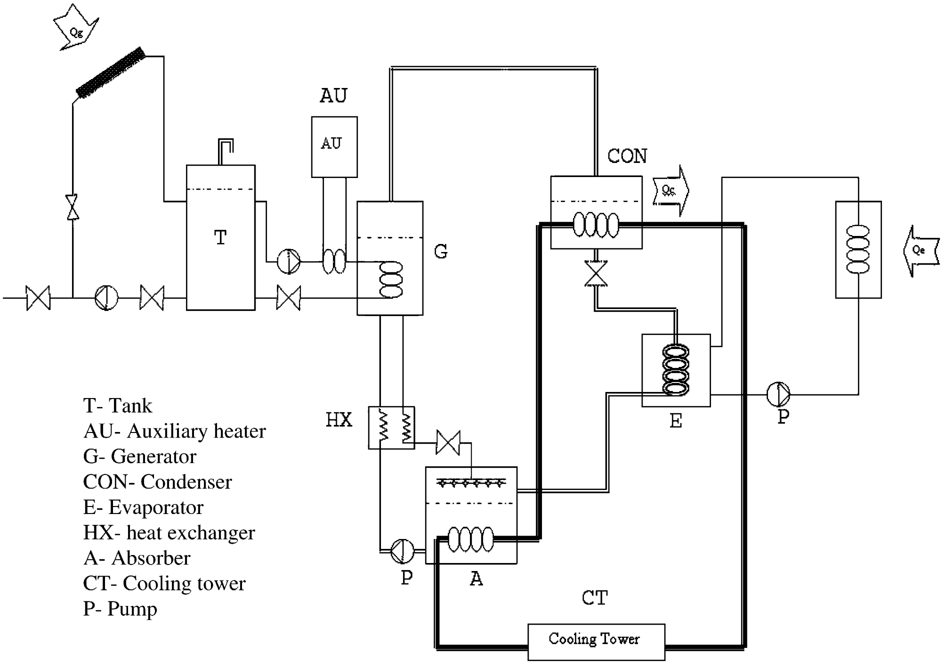

Fig. 2. Schematic diagram of solar air-conditioning system.

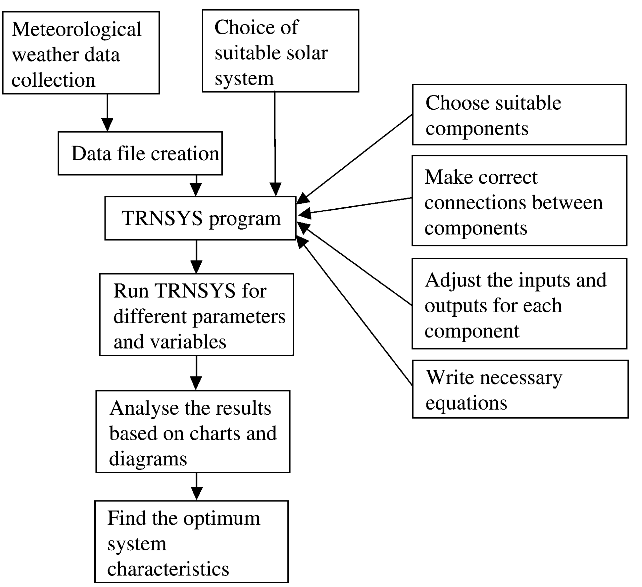

Fig. 3. Flowchart of solar air-conditioning project.

To model the system a number of assumptions are required. The assumptions are based on empirical and experimental results taken from published works on solar cooling and are necessary to create an integrated system for computer simulation. These are:

1. The solar fraction is taken to be the part of the generator load that can be covered by the solar system. Power consumption by other equipment (circulating pump and controllers) is excluded.
2. Since the daily average ambient temperature is higher than the indoor temperature, the storage tank is kept outdoors, thus the energy loss from the storage tank is minimized.
3. There is no need to use antifreeze solution or a heat exchanger between the collector water loop and the storage tank.
4. The circulation pump in the collector water loop operates when the temperature difference between the collector outlet water and the top layer temperature of the storage tank exceeds 3 °C, and stops when this difference becomes lower than 0.5 °C.

3.1. Weather data

In many applications, such as in solar energy technologies an accurate climatic database is needed. In these applications the accuracy of solar radiation and ambient air temperature are crucial. The climatic data based on the typical year concept for the Kuala Lumpur have been used in this work.

The variation of the average dry bulb temperature during 1 year is presented in Fig. 4. As can be seen the maximum dry bulb temperature occurs in March and the temperature is between 26 and 28 °C year round.

The variation of the total radiation on the surface of the collector, the direct normal solar radiation and the global horizontal solar radiation are presented in Fig. 5. As can be seen the average solar radiation on a horizontal surface is about 700 W/m^2 year round.

4. System optimization

A number of runs are carried out in order to optimize the various factors affecting the performance of the system. All runs consider the weather data for Kuala Lumpur, Malaysia. The parameters considered are as follows:

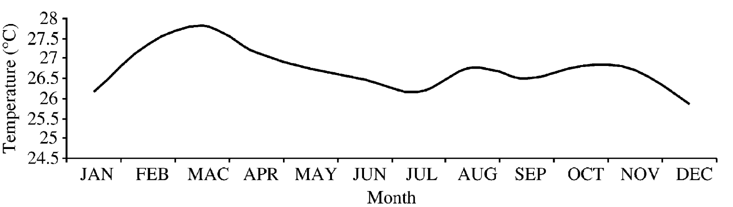

Fig. 4. Monthly average variation of dry bulb temperature during 1 year.

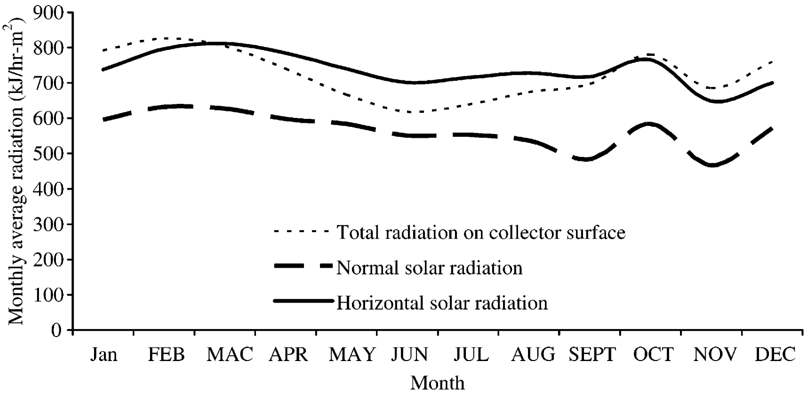

Fig. 5. Monthly variation of solar radiation.

(a) *The collector slope angle*. The solar heat gain from the system for various collector slope angles is shown in Fig. 6. The optimum angle in the Malaysia environment is around 20° for the evacuated tube solar collector.

(b) *Pump flow rate*. Variation of solar fraction is shown in Fig. 7. As observed the solar fraction increases dramatically by increasing the pump flow rate and decreases if its value is more than 0.25 kg/s. So this parameter has the best effect on solar fraction when it set on 0.25 kg/s.

(c) *Boiler thermostat setting*. The boiler thermostat is used in order to control the operation of the boiler, allowing the boiler to operate only when the temperature of the fluid delivered to the load is below an optimum value, which minimizes the required boiler input. The variation of COP with the thermostat setting is shown in Fig. 8. The optimum value is around 91 °C.

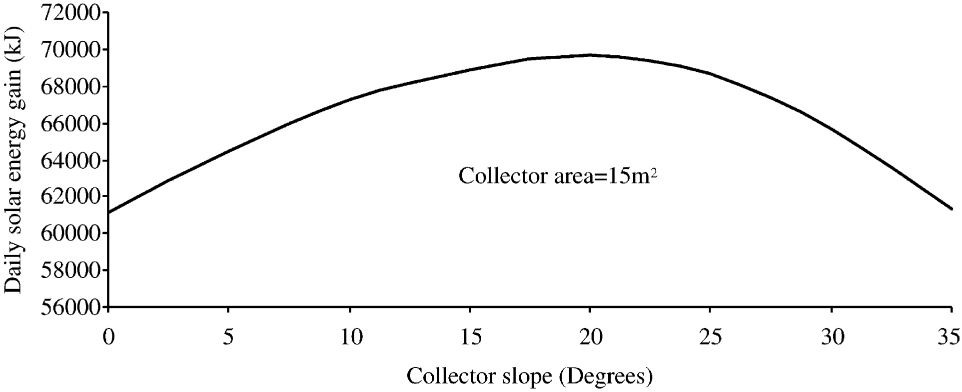

Fig. 6. Effect of collector slope angle on solar energy gain.

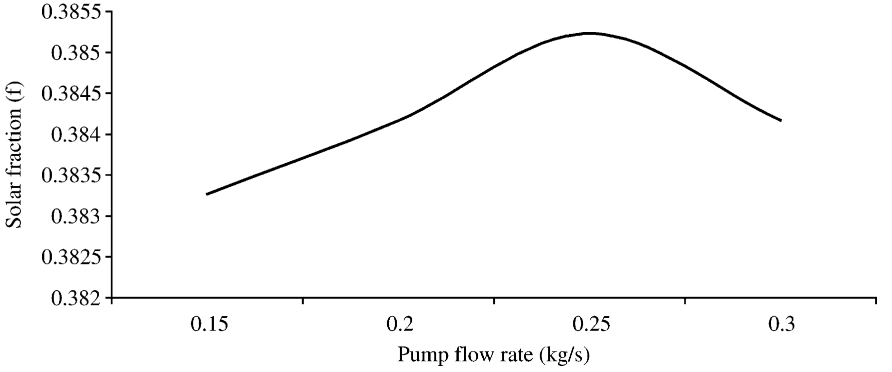

Fig. 7. Effect of pump flow rate on solar fraction.

(d) *Storage tank size*. This factor also plays a role in the optimization of the system. The boiler heat required by the system for different storage volumes is shown in Fig. 9. The figure shows variation of the auxiliary heat required based on different tank sizes. As observed, the optimum size for storage tank is 0.8 m^3.

(e) *Collector area*. The effect of the collector area is evaluated against the boiler heat required. As expected, the greater the collector area the less the boiler heat needed as indicated in Fig. 10 and the more the collected heat as indicated in Fig. 11. Therefore, the optimum value needs to be decided by following an economic analysis, which is presented in Section 5.

5. Economic analysis

It is widely recognized that discounted cash flow analysis is the most appropriate for applications such as in sizing an energy system. This analysis take into account both

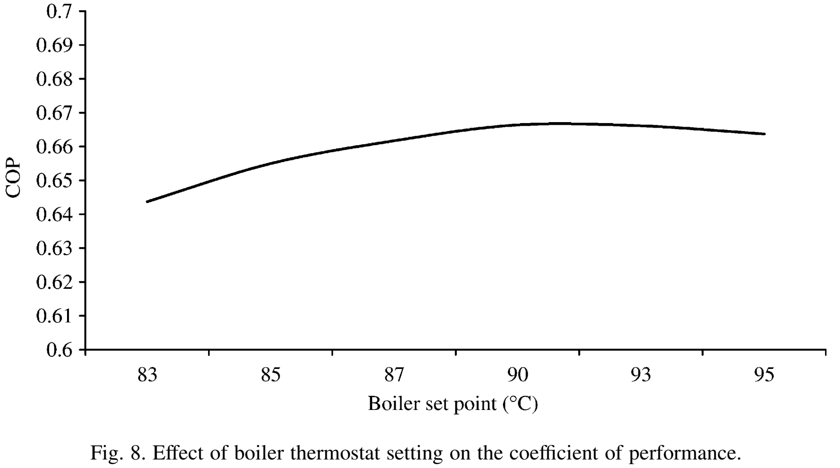

Fig. 8. Effect of boiler thermostat setting on the coefficient of performance.

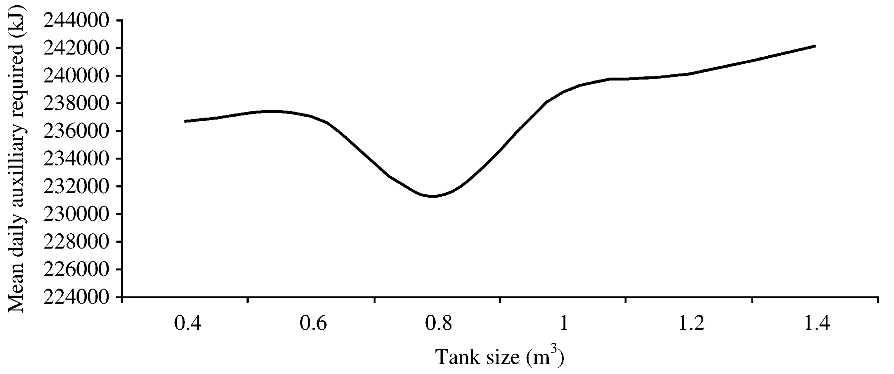

Fig. 9. Effect of storage tank size on the boiler heat.

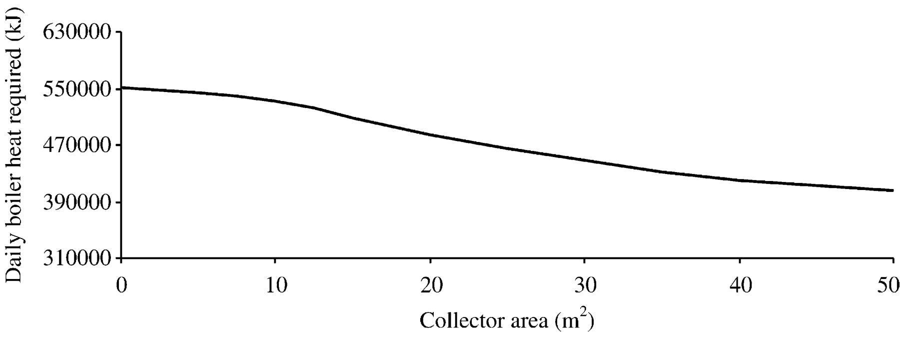

Fig. 10. Effect of collector area on boiler heat required.

the initial cost incurred during the installation of the system and the annual running cost over its entire life span.

The economic objective function for optimal system selection can be expressed in terms of either the energy cost incurred or the energy saving. These two approaches are

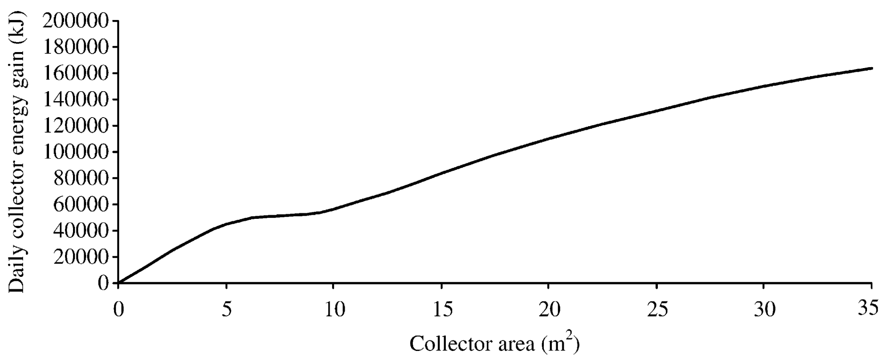

Fig. 11. Effect of collector area on collector energy gain.

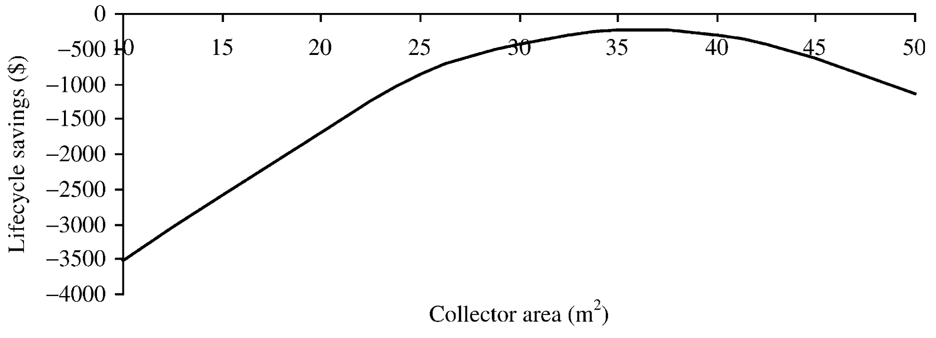

Fig. 12. Collector area against life cycle savings in dollars.

basically identical and differ in the sense that the former has to be minimized while the latter has to be maximized. In our analysis, the latter approach is considered, which can be further subdivided into the following two methods.

(a) Present worth or life cycle savings, wherein all running costs are discounted to the beginning of the first year of operation of the system.
(b) Annualized life cycle savings, wherein the initial expenditure incurred at the start as well as the running costs over the life of the system are expressed as yearly mean values.

To determine the optimum collector area, the electricity savings resulting from the use of the solar system is compared to the cost of the solar system. The results of economic optimization are shown graphically in Fig. 12.

As observed, based on the current rate of electricity for Malaysia which is equal to 0.058$/kW h, the most economic solution is to use the evacuated tube collector with an optimum area of 35 m^2. As can be seen from Fig. 12 even the optimum solution gives negative life cycle savings which is due to the high cost of the solar collectors and the low price of electricity.

The specifications of the final system obtained from the optimization study are shown in Table 1.

Table 1
The final system specification

Collector type	Evacuated tube
Unit capacity	1 ton (3.5 kW)
Collector area	35 m^2
Collector slope	20°
Storage tank size	0.8 m^3
Boiler thermostat setting	91 °C

6. System long-term performance

In this section, the results of various simulations of the optimised system are presented. All graphs in this study are presented for the first day of July which is a typical day with good sunshine and no clouds.

6.1. Collector system performance

The hourly variation of the solar collector outlet temperature and the collector energy gain is presented in Figs. 13 and 14, respectively. At 7 o' clock in the morning when the sun rises the water temperature inside the collector start to increase. As can be seen from Fig. 13, between 13:00 and 15:00, the temperature coming out from the collector is at its maximum and is about 130 °C, i.e. the system needs to be pressurized to prevent evaporation of the water. This is because, as can be seen from Fig. 14, the collector energy gain is also maximum at the same time interval.

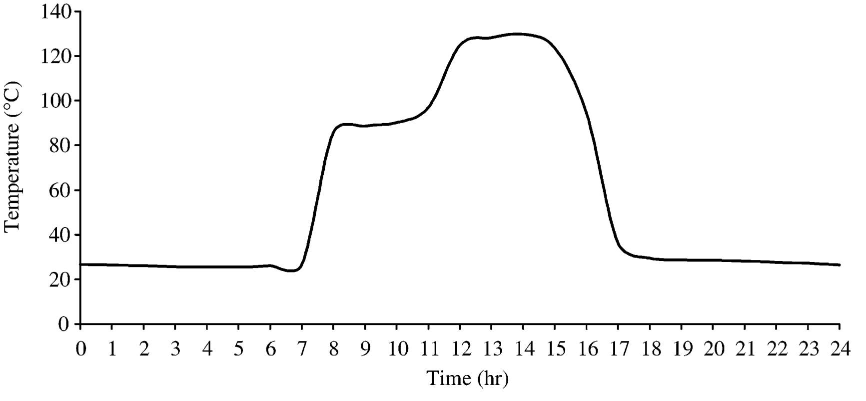

Fig. 13. Hourly variation of solar collector temperature (July 1st).

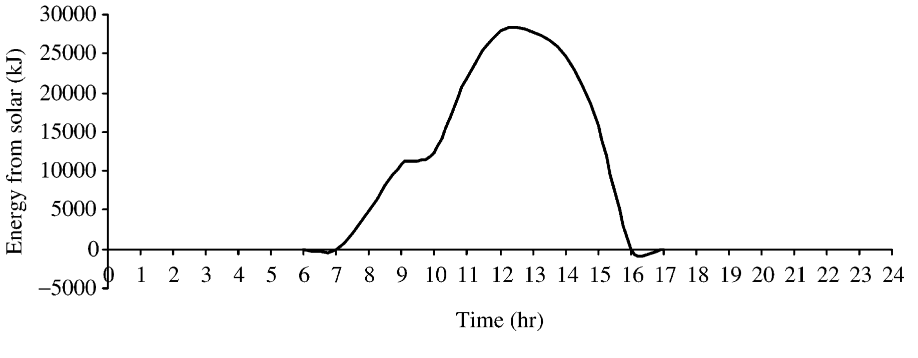

Fig. 14. Hourly variation of collector energy gain from solar collector (July 1st).

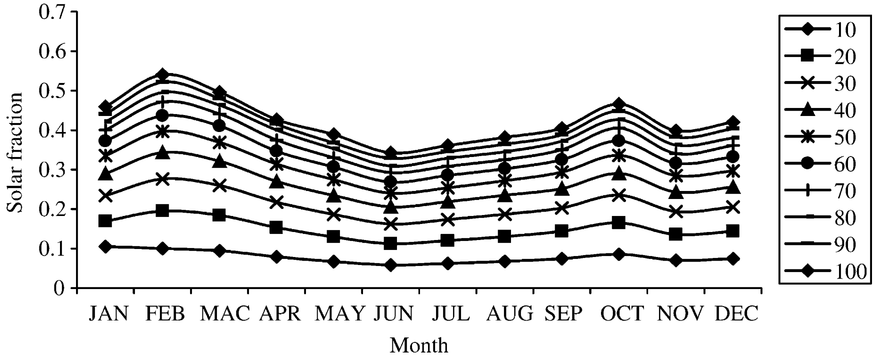

Fig. 15. Monthly variation of solar fraction for different solar collector areas.

6.2. Solar fraction

The purpose of solar energy systems is to displace part of the conventional fuel consumption of the auxiliary heater or boiler. The solar fraction is defined as

$$f = \frac{Q_u - Q_{aux}}{Q_u} \tag{2}$$

where

Q_u rate of useful energy gain from solar collector system (kJ)
Q_{aux} rate of auxiliary energy consumption (kJ)

Since solar fraction is one of the most important parameters to solar collectors, here we tried to investigate this parameter for the solar collector considered based on Malaysia's climate.

Fig. 15 shows the monthly variation of solar fraction for different solar collector areas and Fig. 16 shows the variation of this property for different collector area for the month of February. For collector areas bigger than about 50 m^2 there is no sensible change in solar fraction.

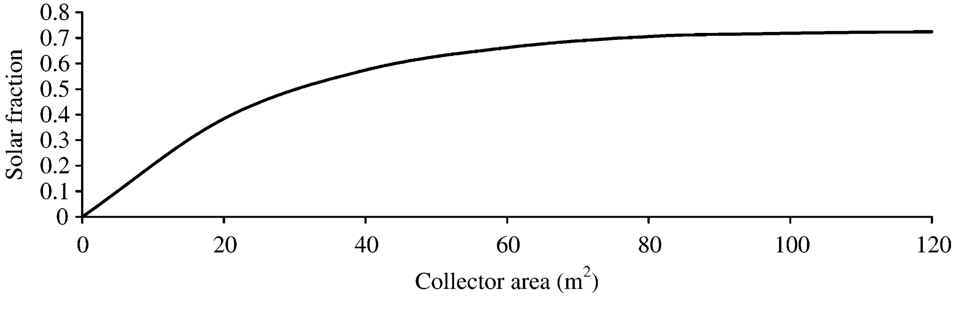

Fig. 16. Variation of solar fraction for different collector areas in February.

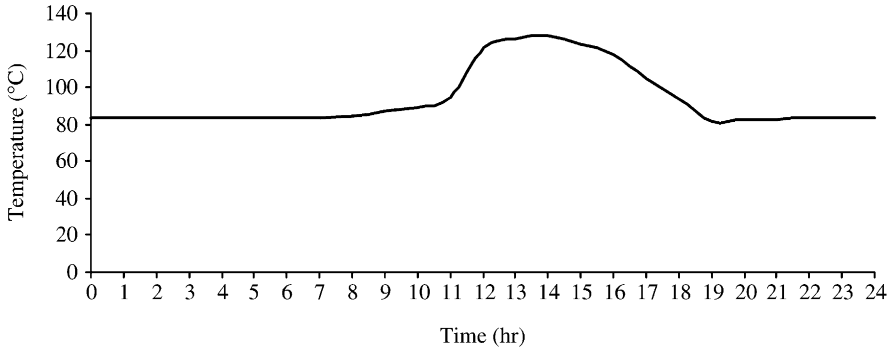

Fig. 17. Hourly variation of storage tank temperature (July 1st).

6.3. Storage tank

This component models the behavior of a temperature stratified storage tank with variable size segments of fluid. The size of segments is governed by the simulation time step, the magnitude of collector and load flow rates, heat losses and auxiliary input. The main advantage over fixed node simulation techniques is that temperature stratification can be modeled with small segments in the temperature gradient zone without the need to use small simulation time steps to obtain a good solution. This model is most appropriate for tanks that exhibit a large degree of stratification.

Fig. 17 shows the variation of the storage tank temperature. Load temperature from tank never falls below 80 °C because when the temperature from the solar collector is not enough and the auxiliary heater is automatically switched ON. Based on the results presented in Fig. 17, during noon when the system is supplied with solar energy the tank load temperature is increased to about 120 °C; therefore, the tank needs to be pressurized to avoid steam generation.

The energy rate from solar to storage tank and energy rate from storage tank to load can be seen in Figs. 14 and 18, respectively. During the operation of the absorption unit these

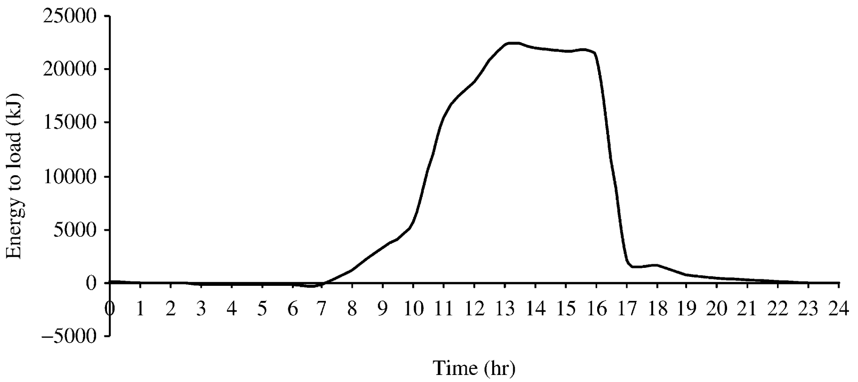

Fig. 18. Hourly energy rate to load from system storage tank (July 1st).

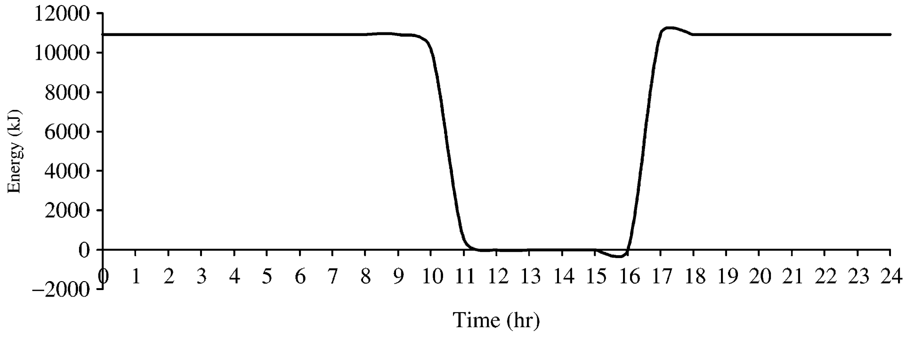

Fig. 19. Hourly auxiliary energy rate of heater operation (July 1st).

two diagrams can be used to find the maximum energy rate from solar collector that result in the maximum energy rate from storage tank to load. This occurs at noon when solar energy is maximum. A comparison of energy from solar and from auxiliary, shown in Fig. 19, proves that the system is in phase with the weather.

6.4. Auxiliary heater

An auxiliary heater is employed to elevate the temperature of a flow stream using internal control, external control or a combination of both types of control. The heater is designed to add heat to the flow stream at a user-designated rate (Q_{max}) whenever the external control input signal is equal to one and the heater outlet temperature is less than a user-specified maximum (T_{set}). By specifying a constant value of the control function of one and specifying a sufficiently large value of Q_{max}, this routine will perform like a domestic hot water auxiliary with internal control to maintain an outlet temperature of T_{set}. By providing a control function of zero or one from a thermostat or controller, this routine will perform like a furnace adding heat at a rate of Q_{max} but not exceeding an outlet temperature of T_{set}. In this application, a constant outlet temperature is not required and T_{set} may be thought of as an arbitrary safety limit.

By considering that the maximum possible heat addition to fluid stream is equal to Q_{max} times conversion efficiency (n_c)

$$Q_{loss} = UA(T - T_{env}) + (1 - n_c)Q_{max} \tag{3}$$

where

Q_{loss} losses from heater (kJ)
T_{env} temperature of heater surroundings for loss calculations (°C)
UA overall loss coefficient between the heater and its surrounding (kJ/h °C)

As it was explained above, for the continuous operation of absorption unit, the auxiliary heater automatically operates whenever tank load temperature is below 80 °C. Fig. 19 shows the auxiliary heater work time during 1 day (1st of July). During the period from

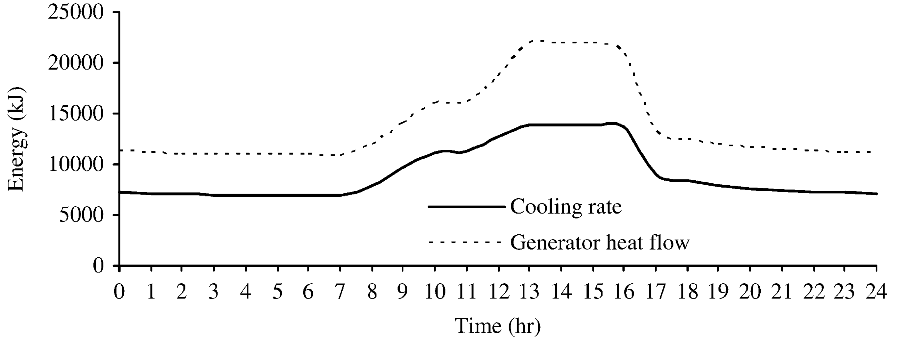

Fig. 20. Hourly cooling rate and generator heat flow of absorption system (July 1st).

11:00 to 16:00, tank temperature could satisfy the system (with energy supplied from the solar system) and there is no need to use the auxiliary heater.

7. Absorption cooling system simulation

The LiBr–water absorption air conditioner employed is a single-effect unit, based on Arkla model WF-36. Its nominal capacity is taken as 12,660 kJ/h, assuming no auxiliary heater.

The rate of cooling provided by the absorption chiller and the rate of energy transferred from the hot water to the generator of the absorption chiller is shown in Fig. 20. A comparison of these two heat rates gives an estimate of coefficient of performance for the absorption unit. The system energy flows during a typical day are presented in Fig. 21. From this graph, a comparison of operation time of each component of the system can easily be seen.

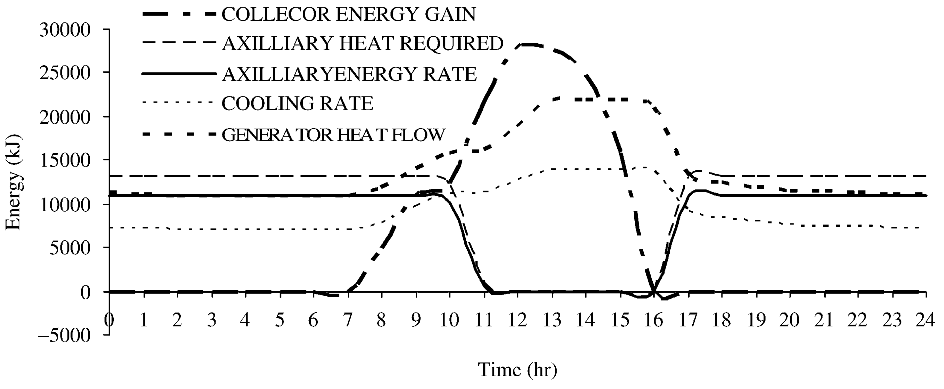

Fig. 21. Energy graph of the system (July 1st).

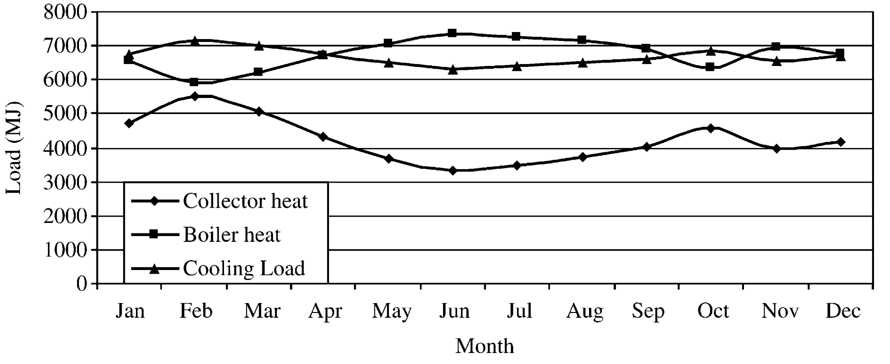

Fig. 22. System energy flows.

Finally the annual energy flows of the system are shown in Fig. 22. The cooling load of the building reaches a maximum monthly value of 7150 MJ (in February). The heat required from the conventional boiler is also shown in Fig. 22. The maximum value is 7432 MJ and occurs during the month of June where the energy supplied from the solar collector is minimum (3315 MJ). The maximum monthly heat energy supplied by the solar system is about 5500 MJ and occurs in February.

8. Conclusions

The greatest advantage of solar-powered air-conditioning when compared to other power sources is that the system is in phase with the daily solar radiation, i.e. the greater the sunshine and thus the cooling load, the larger the cooling effect achieved by the solar refrigerating system. In order to achieve continuous operation of the generator and increase the reliability of the system, a hot water storage tank is essential for high quality performance. As it was proved in this work due to high efficiency at high temperature, evacuated tube solar collector type has been chosen as it can provide good performance at the high temperature required by the absorption system.

Although all the above findings refer to a particular application in Malaysia, the authors believe that similar results can be obtained in countries with high solar availability. However, before any decision is taken, on which type of system to install, the system needs to be optimized with the procedure suggested in this paper. Finally by considering the problem of pollution of the planet due to the burning of fossil fuels the adoption of solar energy to power absorption chillers, even with marginal economic benefits, should not be underestimated.

Acknowledgements

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