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Design and Construction of an Absorption Cooling System Driven by Solar Energy

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Abstract: The objective of this work is to design and construct a lithium bromide–water (LiBr-H_2O) absorption cooling system with a nominal capacity of approximately 1 TOR driven by solar energy which uses Lithium Bromide as absorbent and Water as refrigerant. The proposed absorption cooling system comprises a roof-mounted vacuum tubes solar collector, a single-effect LiBr–H_2O absorption chiller (generator, a solution heat exchanger, an evaporator, a condenser, and an absorber),fan coil unit, a cooling tower, pumps, flow throttling and controlling valves. A thermodynamic analysis of the absorption cooling cycle has been performed to study the effect of various operating conditions on the thermal performance. Results from experimental work carried out in Khartoum City during October 2015show that the COP of the absorption cooling system ranged from 0.57 to 0.64 while the Chilled water temperatures mostly ranged between 17°C to 19.5°C. Condensation and absorption temperatures were under 45°C while the maximum temperature of the driving water is 83°C. The results also show that the generator temperature had a great effect on the performance of absorption and solar collector systems.

Keywords: Absorption system; Air conditioning; Solar energy; Water- Lithium bromide solution

1. INTRODUCTION

In general, a comfortable and healthy environment is now considered as a necessity and many modern processes and products would not exist without precise control of environmental conditions. This explains the fact why at present there are many computer applications in the field of air conditioning to determine the optimum building design that satisfies thermal human comfort with minimum energy requirements.

The conventional vapor compression systems use non-natural working fluids and refrigerants like the chlorofluorocarbon (CFC), hydro chlorofluorocarbon (HCFC). These refrigerants have high global warming as well as ozone layer depletion potentials. Moreover, the European Commission Regulation 2037/2000, which has been implemented on 1 October 2000, treats the whole spectrum of control and phase-out schedule of all the ozone depleting substances. It is indicated that by now all HCFCs will be banned for servicing and maintaining existing systems [1].

Solar cooling is an attractive alternative since it has the advantage of removing the majority of harmful effects of traditional refrigeration machines and that the peak requirements of cooling demand coincide most of the time with the availability of the solar radiation. Solar absorption cooling systems have become more attractive because the maximum cooling load occurs when the high solar radiations are available. It was counted that about 59% of the solar cooling systems in Europe were solar absorption cooling systems [2, 3]. In China, almost all the large-scale solar cooling demonstration projects during the last twenty years were based upon absorption systems

As for tropical areas, the solar intensity is very high and thus solar energy can be used as power sources. Jaruwong Wittaya et al. [4] pointed out that the absorption cooling technology using lithium bromide/water was the most appropriate for the solar applications in Thailand. Fong et al. [5] compared five types of solar cooling systems for Hong Kong, which is commonly featured with long hot and humid summer. The main advantages of solar cooling systems are concerned with the reduction of peak loads for electricity utilities, the use of zero ozone depletion impact refrigerants, the decrease of primary energy consumption and global warming impact [6,7]. Moreover, the freedom from noise and vibrations, long lasting, cheap maintenance, and most importantly the possibility of using any type of heat source, including solar radiation and geothermal or waste heat, to energize the system and provide reliable cooling. The applications of these cooling systems are wide and include freezing, cooling, and air- conditioning. However, these systems are heavy in weight and have a relatively high initial cost. Commercially available absorption chillers for air conditioning applications usually operate with solution of lithium bromide in water and use steam or hot water as the heat source [8]. It has been testified that single-effect LiBr/H2O absorption units using fossil-fuels are not competitive from the energy, economic and environmental points of view. They are only competitive when using waste or renewable heat as part of the driving energy [9].Besides, according to the operating temperature range of driving thermal source, single-effect LiBr/H2O absorption chillers have the advantage of being powered by ordinary flat-plate or evacuated tubular solar collectors available in the market. Consequently, the majority of solar cooling systems are based on single-effect LiBr- H2O absorption chillers. Under normal operation conditions, such machines need typically temperatures of the driving heat of 85 to 100 ◦C and achieve a COP of about 0.75. Figure (1) illustrates a general scheme of a solar-powered single-effect absorption cooling system. The system employs a solar collector array, an absorption chiller, a cooling tower, a heat storage water tank, and an auxiliary heater. The hot water storage tank is used in the system as a heat reservoir. When there is no cooling demand to satisfy, the solar energy is accumulated in the storage tank. When solar energy is insufficient to heat the water to the required generator inlet temperature level, the auxiliary heat source is provided to supply the generator.

2. ABSORPTION SYSTEM DESCRIPTION

The major components of the absorption system as shown in figure (2) are a roof-mounted vacuum tubes solar collector, a single-effect absorption chiller, a cooling tower, a solution heat exchanger, pumps, flow throttling and controlling valves. In addition, it is comprised of four main flow circuits which are the solar circuit, the hot water circuit, the chilled water circuit, and the cooling water circuit.
The system primary energy source is the solar energy, which is converted into thermal energy by the vacuum tubes solar collector where the water is the heat transfer medium in the circuit. With reference to the numbering system shown in Figure (2), at point (4) the solution is rich in refrigerant and a pump forces the liquid through a heat exchanger to the generator (6). The temperature of the solution in the heat exchanger is increased. In the generator thermal energy is added and refrigerant boils off the solution. The refrigerant vapour (7) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (8) flows through a flow restrictor to the evaporator (9). In the evaporator, the heat from the room load evaporates the refrigerant, which flows back to the absorber (10). At the generator exit (1), the steam consists of absorbent-refrigerant solution, which is cooled in the heat exchanger, returns back to the absorber. From points (10) to (4), the solution absorbs refrigerant vapour from the evaporator and rejects heat to the circulating cooling water from the cooling tower.

3. MODELING OF ABSORPTION SYSTEM

In the present study, the space to be air conditioned by solar cooling has a net floor area of 8.5 m^2 and a volume of 25.5 m^3. It is supposed to be occupied by two persons. For an outdoor temperature of 43^o C, the thermal load, for a comfortable indoor temperature of 24 ^{o} C and 50% relative humidity, is about 4 kW. The chilled water circuit consists of a pump and a fan coil which cools the space. Piping is made of copper and is properly isolated to minimize thermal loses.
All components of the absorption chiller (generator, condenser, evaporator, absorber, solution heat exchanger) are merely heat exchangers. A type of shell and tube heat exchanger (STHE) is selected for the design and experimental work since it is the most common type and heat transfer efficiency is better[10,11] . In the present study a flooded type STHE is adopted where the tubes carrying the hot or cold fluid are totally immersed in the cycle working solution
(LiBr-H_2O). Counter flow is selected for the design of the STHE since it is the most efficient heat transfer unit [12].
At the beginning of the study a number of simplifying assumptions are made to lay the foundations without obscuring the basic physical situation.

These assumptions are as follows [13,14]:

1. Generator and condenser as well as evaporator and absorber are under the same pressure.
2. There are no pressure changes except through the flow restrictors.
3. Pressure drops through tubes are negligible.
4. Refrigerant leaving the evaporator is saturated pure water.
5. Liquid refrigerant leaving the condenser is saturated.
6. Temperature change across the regulating valves is negligible.
7. Heat transfer between equipments and surrounding is negligible.
8. Pumping process is isentropic.

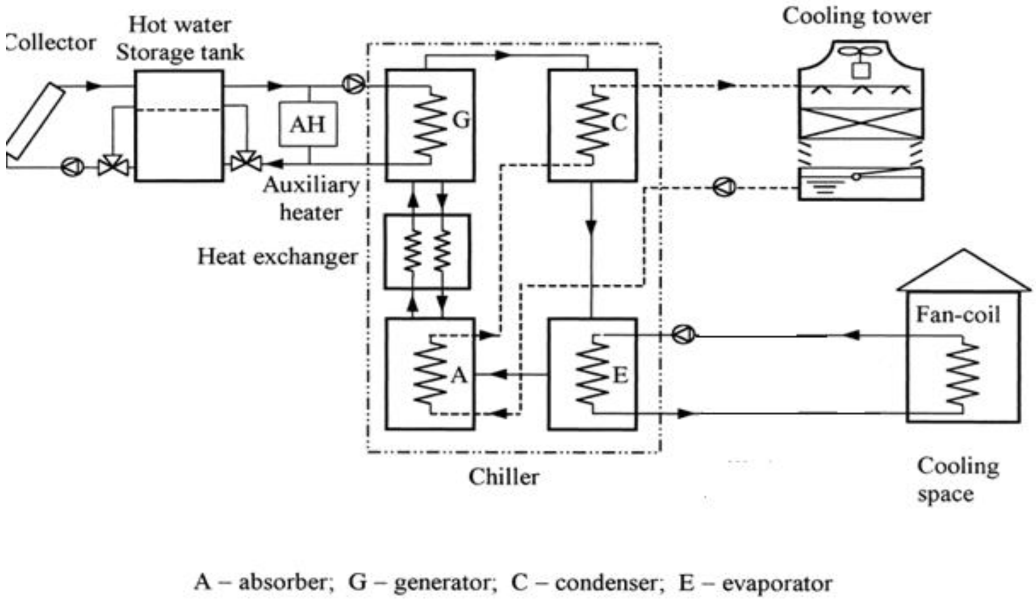

Fig. 1: The schematic diagram of the solar-powered single-effect absorption cooling system.

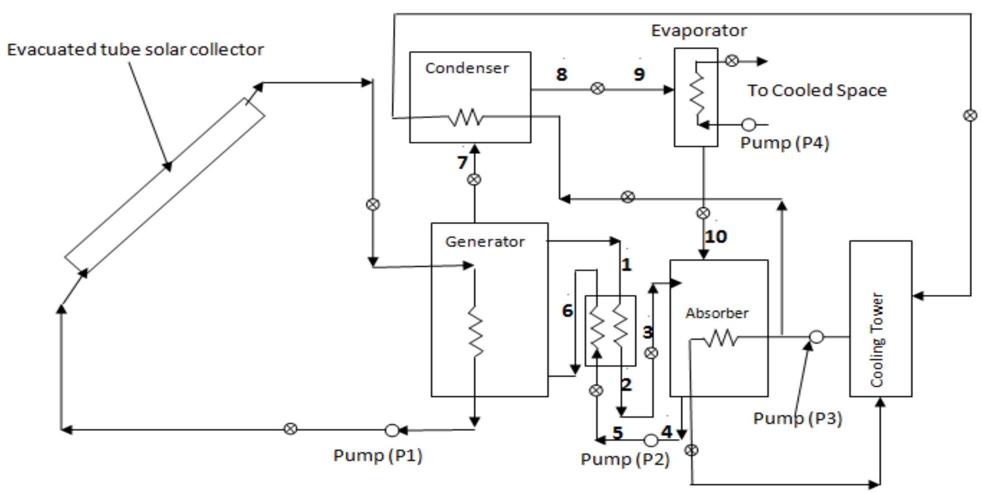

Fig. 2: Schematic of an experimental solar operated absorption air conditioner

4. THERMODYNAMICS PROPERTIES

With reference to figure (2) flow of refrigerant is assumed to be unity [15, 16].

$$m_7 = m_8 = m_9 = m_{10} = 1 \quad (1)$$

Weight fraction of absorbent:

$$x_7 = x_8 = x_9 = x_{10} = 0\% \quad (2)$$

Evaporator condition:

$$T_9 = T_{10} \quad (3)$$

$$P_3 = P_4 = P_9 = P_{10} \quad (4)$$

Absorber condition:

$$T_4 = T_5$$

Enthalpy

$$h_4 = h_5 \quad (5)$$

$$x_4 = x_5 = x_6 \quad (6)$$

Condenser condition:

Enthalpy

$$h_9 = h_8 \quad (7)$$

$$T_9 = T_8 \quad (8)$$

Generator condition:

$$T_1 = T_7 \quad (9)$$

$$h_2 = h_3 \quad (10)$$

$$x_1 = x_2 = x_3 \quad (11)$$

5. FLOW AND ENERGY REQUIREMENTS

Generator Material Balance:

$$m_1 + m_7 = m_6 \quad (12)$$

$$m_1 x_1 = m_6 x_6 \quad (13)$$

$$m_1 = m_7 \left(\frac{x_6}{x_1 - x_6}\right) \quad (14)$$

$$m_1 = m_2 = m_3 \quad (15)$$

$$m_4 = m_5 = m_6 \quad (16)$$

Heat Exchanger Energy Balance:

$$m_5 h_5 + m_1 h_1 = m_2 h_2 + m_6 h_6 \quad (17)$$

$$h_6 = h_5 + \frac{m_1}{m_6}(h_1 - h_2) \quad (18)$$

Generator Energy Requirements:

$$Q_g = \dot{m}_7 h_7 + \dot{m}_1 h_1 - \dot{m}_6 h_6 \quad (19)$$

Condenser Energy Requirements:

$$Q_c = \dot{m}_7 (h_8 - h_7) \quad (20)$$

Absorber Energy Requirements:

$$Q_a = Q_c - Q_g - Q_e \quad (21)$$

Solution Heat Exchanger Energy Requirements:

$$Q_{he} = \dot{m}_1 (h_1 - h_2) \quad (22)$$

Coefficient of Performance

$$COP = \frac{Q_e}{Q_g} \quad (23)$$

Evaporator Design

Water circulation requirements:

$$\dot{m} = \frac{Q_e}{C_p \Delta T} \quad (24)$$

$$\Delta T = LMTD$$

$$A = \frac{\dot{m}v}{V} \quad (25)$$

Tube flow area [73]:

$$A_t = \frac{\pi}{4} {d_i}^2 \quad (26)$$

Design will be based on a circular tube of stainless steel with an inner diameter 0.5"(1.27 cm) and an outer diameter of 0.62"(1.57cm)

Number of tubes required:

$$N = \frac{A}{A_t} \quad (27)$$

Mass velocity C_t [73]:

$$C_t = \frac{\dot{m}}{A} \quad (28)$$

Reynolds number:

$$R_e = \frac{d_i C_t}{\mu} \quad (29)$$

Prandtl number [73]:

$$Pr = \left(\frac{C\mu}{k}\right)^{\frac{1}{3}} \quad (30)$$

Coefficient of heat transfer[73]:

$$h_i = J_h \left(\frac{k}{d_i}\right) (Pr) \quad (31)$$

Inside film coefficient [73]:

$$h_{io} = h_i \frac{d_i}{d_o} \qquad (32)$$

Over all heat transfer coefficient [73]:

$$U = \frac{1}{\frac{1}{h_o}+\frac{1}{h_{io}}} \qquad (33)$$

Heat transfer area :

$$A_{ht} = \frac{Q}{U\Delta T} \qquad (34)$$

$$\text{tube area avilable/m} = \pi d_o \qquad (35)$$

Absorber, Condenser, Generator, and Solution Heat Exchanger Design

Design procedures for Absorber, Condenser, Generator, and Solution Heat Exchanger are similar to that of evaporator.

Configuration of Heat Exchangers

Table (1) shows the specifications of Generator, Absorber, Condenser, Generator, Solution Heat Exchanger, and Evaporator which obtained from the above design procedures.

Calculation of System COP

The experimental measurements have been used to calculate system COP, using the following equations:
Heat supplied to the generator

$$Q_g = \dot{m}_{gw} C_{pgw} (T_{igw} - T_{ogw}) \qquad (36)$$

Heat removed by evaporator

$$Q_e = \dot{m}_{ew} C_{pew} (T_{oew} - T_{iew}) \qquad (37)$$

The results obtained are shown and discussed below.

Table 1: Specifications of Evaporator, Absorber, Condenser, Generator, and Heat Exchanger

Item	Evaporator	Absorber	Condenser	Heat Exchanger	Generator
Number of tubes	19	19	21	19	23
Outside diameter	15.7 mm	15.7 mm	15.7 mm	15.7 mm	15.7 mm
Inside diameter	12.7 mm	12.7 mm	12.7 mm	12.7 mm	12.7 mm
Thickness	1.5 mm	1.5 mm	1.5 mm	1.5 mm	1.5 mm
Outlet cooling water temperature	17 ºC	34 ºC	43 ºC	/	85 ºC (heating water to solar collector)
Inlet cooling water temperature	12 ºC	30 ºC	30 ºC	/	92 ºC (heating water from solar collector)
Length of tubes	1.2 m	1.4 m	1.3 m	0.7 m	1.3 m

6. RESULTS AND DISCUSSION

The solar absorption cooling system was tested in a single-effect operation mode during the period 5th October to 15th October 2015 in Khartoum City. In this section, experimental results carried out during 3 selected days will be detailed:
Figures (1), (2) and (3) show the variation of solar collector outlet temperatures, solar collector inlet temperatures, fan coil outlet temperatures, fan coil inlet temperatures, room temperatures, and ambient temperatures with time, for the three selected days.

Figure (4) shows the system COP with time. The temperatures corresponding to both sides of the solar collector, in the first run (5th of October, 2015), are represented in Figure (1). At the beginning of the experiment, at 11:00 hour, hot water from the solar collector entered the generator at 79 ^{o}C, reaching the maximum of 82.6 ^{o}C at 12:00 hour .When the pump (P2),as shown in Figure (2), circulating the working solution (LiBr-H_2O) is switched on at 12:00 hour the outlet temperature from solar collector entered the generator slightly dropped down reaching a minimum of 70.6 ^{o}C at 18:00 hour. At this hour the pump (P2) is switched off, since the amount of heat in the generator is not sufficient enough to release the refrigerant from the absorber. This can be seen clearly from the sharp fluctuation of the inlet room air temperature in figure (1). The water temperatures at inlet and outlet of the fan coil unit are shown also in Fig. (1) as well as the room air temperatures corresponding to the air-conditioned space. One can see that the solar absorption system was able to keep the indoor temperature between 27.4^{o}C and 26 ^{o}C by cooling down the water up to 17.1 ^{o}C. Evaporator temperatures were under 19 ^{o}C for most of the time. At the beginning of theexperiment the indoor room temperature was 36.3 °C, after an hour it was 28 °C reaching a minimum of 26 °C at 15:00 hour. Likewise, the chilled water temperature dropped from 35 °C to 23.5 °C reaching a minimum of 17.1 °C at 15:00 hour. Moreover, it is interesting to note that the maximum outdoor temperature reached 44 °C. The slight downward slope observed in the chilled water temperature curve is due to the slightly low increase in ambient temperatures. When no more hot water was supplied to the generator, the chilled water temperature sharply increased till the end of the experiment.

Figure 2 shows the variation of solar collector outlet temperatures, solar collector inlet temperatures, fan coil outlet temperatures, fan coil inlet temperatures, room temperatures, and ambient temperatures with time. These results were obtained from the experimental results on the second run (7 October 2015), this day can be considered as intermittent clouds day in which the maximum ambient temperature was 42 °C at 16:00 hour while the minimum reached 26 °C at night hours. It was observed that the solar collector temperature entering the generator reached a maximum of 80.8 °C at 12:00 hour. At this hour the absorption system cycle was started up when the room temperature recorded was 36.6 °C reaching a minimum of 26.5 °C at 15:00 hour. After approximately three hours and a half intermittent clouds appeared in the sky, the weather became partly cloudy at 16:00 hour, a sharp decrease in the solar collector temperature occurred during this period which reached a drop of 10 °C from 15:00 to 17:00 hour, the solar collector temperatures dropped from 79.5 °C to 68 °C during this period, while the indoor temperature increased from 26.5 °C to 33 °C during the same period. At this time the pump circulating the working solution is turned off.

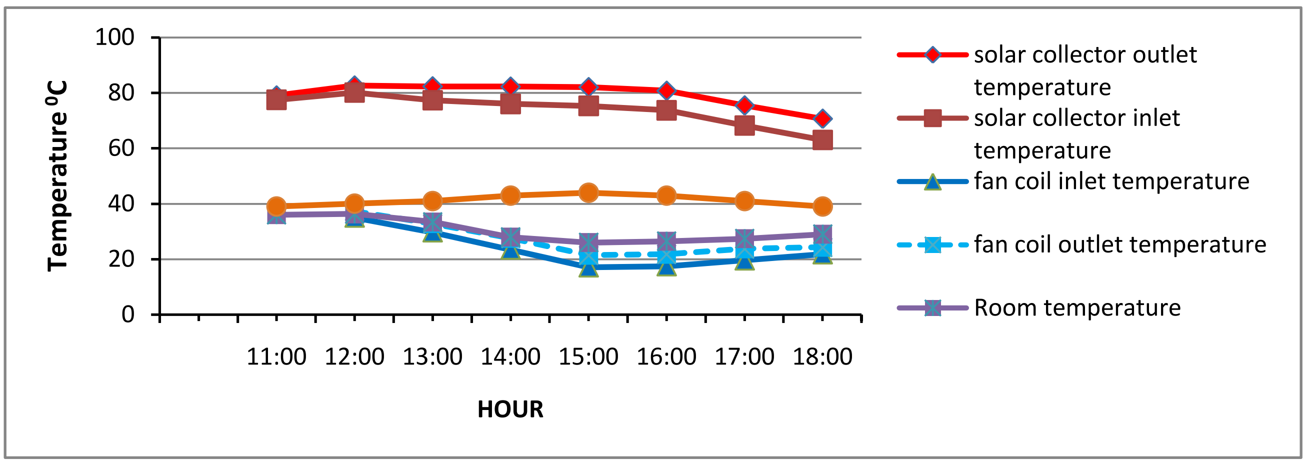

Fig. 1: The variation of solar collector, fan coil, room, and ambient temperatures corresponding to 5/10/2015

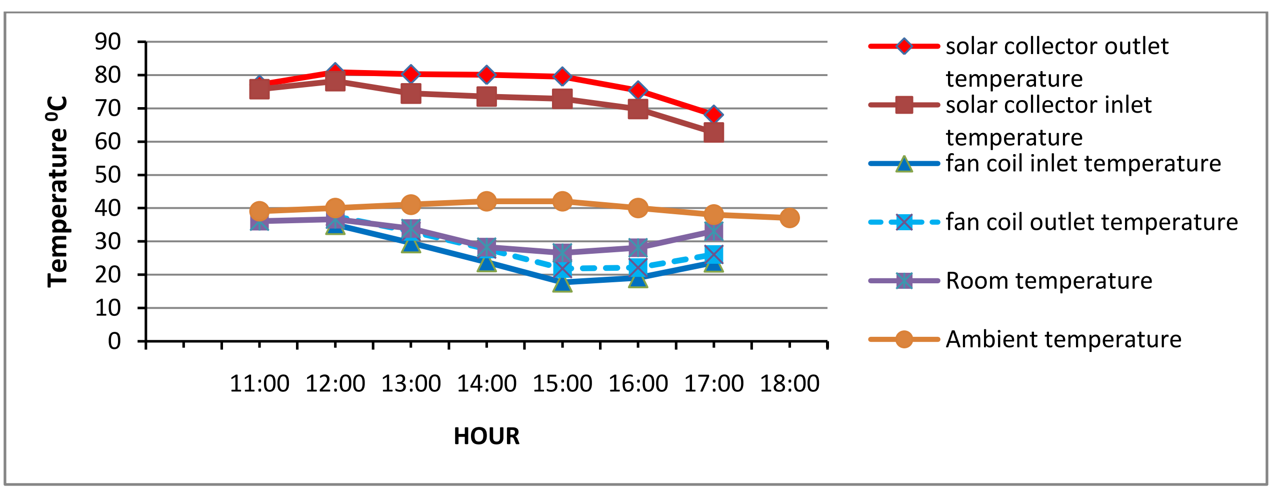

Fig. 2: The variation of solar collector, fan coil, room, and ambient temperatures corresponding to 7/10/2015

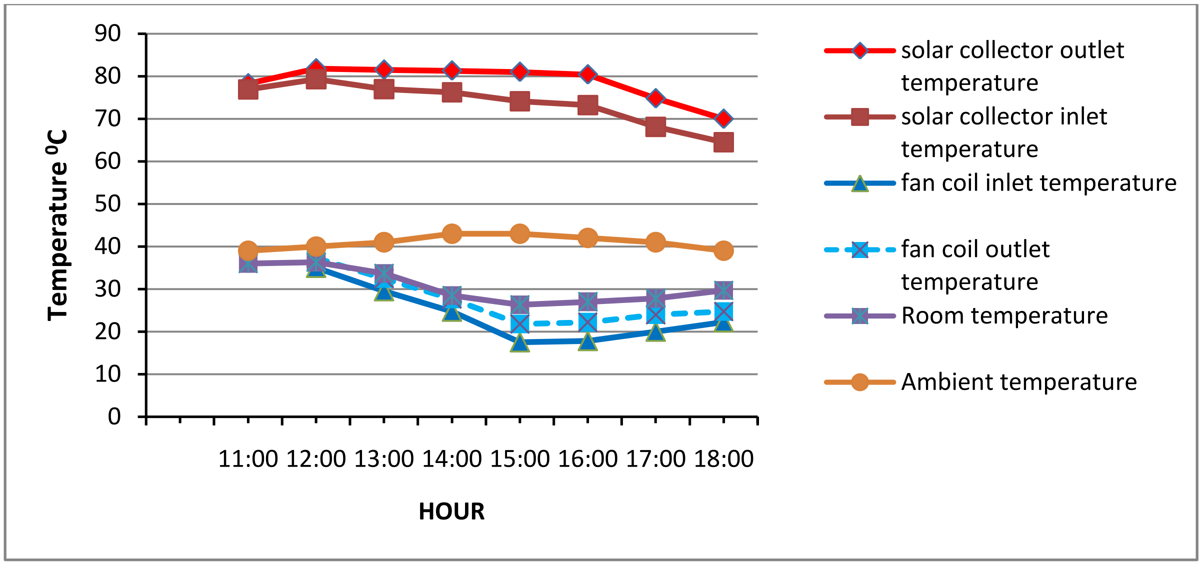

Fig. 3: The variation of solar collector, fan coil, room, and ambient temperatures corresponding to 10/10/2015

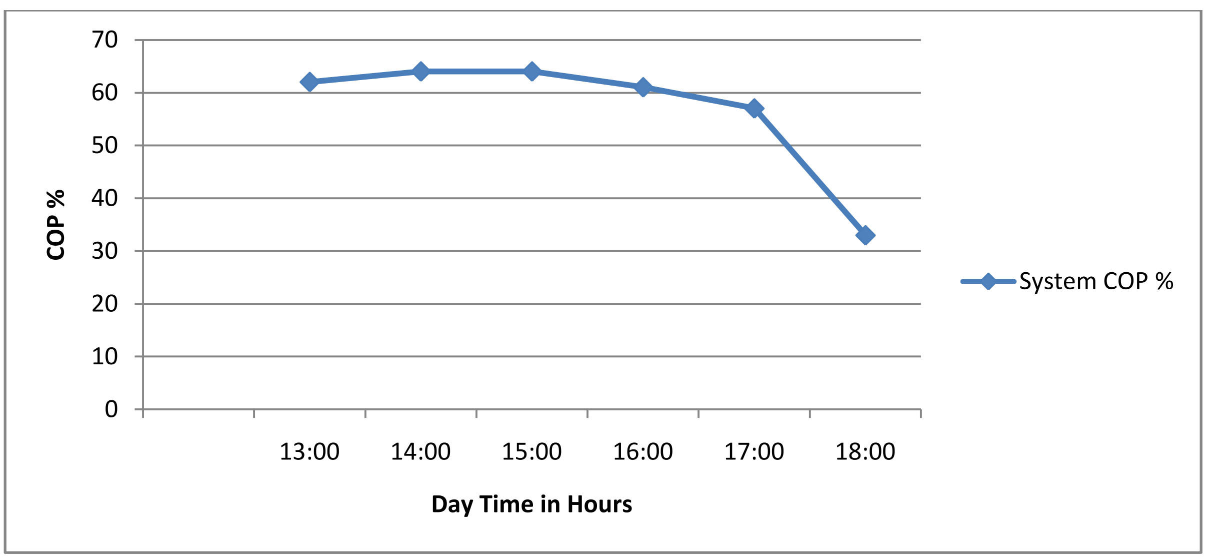

Fig. 4. System COP with Time Corresponding to day 5/10/2015

The experimental results obtained during the third run (10 October 2015) are presented in figure (3). Discussion will be focused on comparing these results with those from the previous experiment, trying to avoid aspects that could be repetitive. This day can be considered as similar to the first run (5 October 2015), with a maximum ambient temperature of 43 ^{o}C. In Figure (3) one can see that this test was started up at the same time of the first run, around 11:00 hour. The higher temperature of the solar collector entered generator was 81.8 ^{o}C at the same hour of the first run (12:00) which was recorded as 82.6 ^{o}C, while the minimum is 70 ^{o}C compared with 70.6 ^{o}C at the same hour of the first run(18:00). Only after approximately 1 hour, the indoor temperature in the cooled space began to decrease. It dropped from 36.3 ^{o}C, at the beginning of the experiment, to 26.3 ^{o}C, which is the minimum, at 15:00 hour. The indoor temperature started to increase from 16:00 hour coinciding with the decrease in temperature of solar collector entering the generator. This can be seen from the sharp fluctuation of the indoor temperature shown in figure (3).The input power was cut off at 18:00 hour since the indoor temperature increased to 29.7. From table (4) the maximum COP calculated from the experimental data were found to be 0.64 compared with 0.73 which is the maximum obtained from the theoretical results.

7. CONCLUSIONS

An absorption refrigeration cycle employing a lithium bromide-water solution as the working fluid has been investigated in this work. Solar thermal systems providing a low temperature heat source for absorption cooling units have been also examined. At first, theoretical absorption refrigeration systems were described and fully analyzed. An aqueous Li-Br absorption cooling cycle, with hot water from an evacuated tube solar collector as a heat source and cooling water from a cooling tower as a heat sink, was then modeled and constructed.The maximum generator temperature was found to be 83°C, while the range of minimum evaporator temperature was within the range (17°C – 19.5°C).The results of the thermodynamic analysis indicated that the value of COP for the solar absorption system was (0.73) compared with (0.64) which was obtained from the experimental results. The results also showed that the generator temperature had a great effect on the absorption and solar collector systems performance.

NOMENCLATURE

Symb	Description	Unit
A_t	Flow area of single tube	m^2
A_{ht}	Heat transfer area	m^2
COP	Coefficient of performance	/
C_p	Specific heat	kJ/kg K
C_{pgw}	Specific heat of generator water	kJ/kg K
C_{pew}	Specific heat of evaporator water	kJ/kg K
d_i	Inside tube diameter	m
d_o	0utside tube diameter	m
h	Enthalpy	kJ/kg
h_i	Inside film coefficient	W/m^2K
h_{io}	Inside film coefficient referred to outside surface	W/m^2K
J_h	Heat transfer factor	/
K	Thermal conductivity	W/mK
$LMTD$	Logarithmic Mean Temperature Difference	^{o} C
m	Mass flow ratio	Kg/kg of H_2O
$\dot{m}$	Mass flow rate	kg/s
$\dot{m}_{gw}$	Water Mass flow in generator	kg/s
$\dot{m}_{ew}$	Water Mass flow in evaporator	kg/s
N	Number of tubes	/
P	Pressure	bar
Q_g	Heat supplied to the generator	kW
Q_c	Heat removed by condenser	kW
Q_a	Heat removed by absorber	kW
Q_e	Heat removed by evaporator	kW
Q_{he}	Heat flux in heat exchanger	kW
T	Temperature	^{o} C
T_{igw}	Inlet generator water temperature	^{o} C
T_{ogw}	Outlet generator water temperature	^{o} C
T_{iew}	Inlet evaporator water temperature	^{o} C
T_{oew}	Outlet evaporator water temperature	^{o} C
U	Overall heat transfer coefficient	W/m^2K
V	Velocity	m/sec
x	Lithium bromide Weight fraction	/
v	Specific volume	m^3/kg
μ	Dynamic viscosity	kg/ms

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