

Feasibility Study on a Solar Powered Absorption Cooling System in Bangladesh

Sanwarul Haque, Himangshu Ranjan Ghosh* and Neem Chandra Bhowmik

Department. of Applied Physics, Electronics and Communication Engineering, Dhaka University, Dhaka-1000 Bangladesh

*Renewable Energy Research Centre, Dhaka University, Dhaka-1000 Bangladesh

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Abstract

Now-a-days absorption chillers have become popular in commercial complexes for air conditioning in Bangladesh. Grid electricity or generator is commonly used for heating of the absorbent in the chiller. In this research work a feasibility study has been carried out on solar thermal energy based absorption chiller in Bangladesh. For measured solar radiation data (at Lat 23.73°, Long 90.40° and Period 2003-2005) and black-crome selective coating based flat plate solar water heating systems, the modeling and simulation of the absorption solar cooling system has been carried out with VBA, C++ program and an economic evaluation is predicted using RETScreen simulator. At Dhaka for a typical 1 ton AC, 3.5kW rated power a flat plate solar water heating system (SWHS) having 30 m² collector areas with fixed tilt angle of 23.73°, azimuth angle of 0° is sufficient to deliver required energy to the chiller. It has been found that average Coefficient of Performance (COP) of the solar power based absorption chiller is 0.58. Moreover, the system mitigates 13.485 tCO₂/year. Though for a low efficient flat plate collector based SWHS the overall system's maximum efficiency becomes 27%, it has a payback period of 12.6 years (lifetime of 20 years with low maintenance) in comparison with the grid electricity, which predicts that the solar thermal powered air cooling system is feasible for Bangladesh.

I. 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 grade heat can be employed. Absorption is the process of attracting and holding moisture by sorbent materials that have an ability to attract and hold other gases or liquids and have a particular affinity to water. During absorption the sorbent material 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 moisture. At present LiBr-Water absorption chiller¹ is widely used for absorption cooling systems since it uses the environmental friendly LiBr-Water solution to overcome the environmental problems associated with the use of traditional cooling systems.

Various researchers presented modeling and simulation studies on solar cooling and air-conditioning systems recently. In Ispra, Italy, an experimental study of a solar-powered LiBr-H₂O absorption cooling was conducted by Van Hattem and Dato². The system has a collector area of 36 m² and the overall cooling efficiency was found to be around 11%. Florides et al modeled a complete system³, comprised of a solar collector, a storage tank, a boiler and a LiBr-water absorption chiller, 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 file for Nicosia, Cyprus. In Spain a solar powered absorption air cooling system was constructed in Madrid⁴ and it was found that the absorption chiller had an average COP of 0.37.

The purpose of the present paper is to study the feasibility of the system on the basis of the system's operational performance under the climatic conditions of Bangladesh.

II. Solar Resource Assessment at Dhaka

Renewable Energy Research Centre, University of Dhaka has been measuring global radiation since September 2002

under Solar and Wind Energy Resource Assessment program at Dhaka. The organization has been collecting data at one minute intervals using an Eppley Precision Pyranometer and ICP-CON data acquisition card with a microcomputer.

The performance of a solar radiation conversion system is affected by its orientation and its tilt angle with the horizontal plane. This is because both of these parameters change the amount of solar energy received by the surface of solar system. The best way to collect maximum daily energy is to use tracking systems. The trackers are often expensive and are not always applicable. The optimum output over the years the collectors are placed at latitude angle. A recent study shows the output would be more if the solar energy collectors are placed at 10° for Mar-Sep and 40° for Oct-Feb⁵. The measured values of global and diffuse radiation for January 2003 to December 2005 are given in table 1 along with the estimated values of global radiation for tilted surfaces. The variation of solar radiation over hours is shown in Fig 1.

Table. 1. Availability of global radiation (kWh/m²/day)

Month	Tilt angles			
	0°	0°	24°	10° and 40°
Jan	1.72	3.16	3.83	4.00
Feb	1.99	4.46	5.18	3.90
Mar	2.34	4.88	5.19	5.08
Apr	2.90	5.28	5.20	5.32
May	3.12	5.46	5.09	5.38
Jun	2.83	4.22	3.90	5.09
Jul	2.76	4.42	4.10	4.40
Aug	2.79	4.18	4.03	4.11
Sep	2.54	3.74	3.80	3.86
Oct	1.91	3.53	3.85	3.87
Nov	1.60	3.92	4.81	5.04
Dec	1.39	3.17	3.96	4.28
Avg	2.32	4.20	4.41	4.53

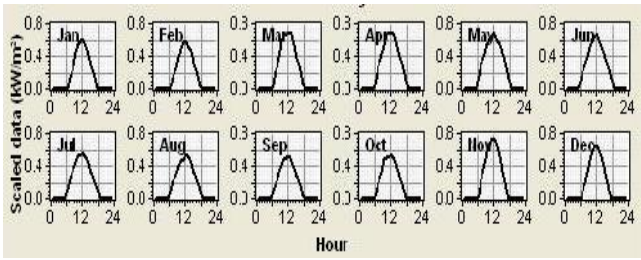


Fig 1. Variation of global radiation with hour

III. Energy Balance Model for an Absorption Chiller

It is possible to write the full set of energy balances, material balances, the rate equations and equilibrium relationships, considering each component in the cooler, and simultaneously solve the set to determine the operating conditions and energy rates. There are four basic heat exchanger surfaces: the evaporator, the absorber, the generator and the condenser, in addition to a liquid-liquid heat exchanger. The cycle of operation can be started from the evaporator. The refrigerant (water) is evaporated while it is taking heat from the fluid being chilled (air for instance). The water vapor is then sucked up by lithium bromide spray injected into the absorber, the name absorption system. Due to the exothermic reaction taking place in the absorption process, heat has to be removed, and the mixture of lithium bromide and refrigerant vapor at this stage is called strong solution. ‘Strong’ and ‘weak’ refer to the amount of refrigerant present⁶. The strong solution is then pumped through a liquid-liquid heat exchanger to the generator. This heat exchanger will improve the cycle performance. In the generator (sometimes called the concentrator) the strong solution is heated and boiled by an external heat source to release the refrigerant vapor, leaving behind the concentrated lithium-bromide-water solution. The refrigerant vapor leaving the generator is condensed in the condenser and is directed to the evaporator through an expansion valve. The weak solution flows back to the absorber through the liquid –liquid heat exchanger as a spray to complete the cycle. Fig.2 shows the flow diagram of an absorption chiller.

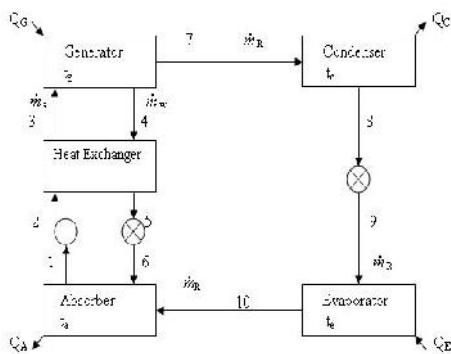


Fig.2. Flow diagram for a lithium bromide- water absorption chiller.

Enthalpies of water (refrigerant) and LiBr (absorbent) solutions were calculated with reference temperature at 25°C. The following expressions⁷⁻⁹ were found essential to the calculation of the absorption cycle performance:

(1) The enthalpy of pure water liquid at temperature $t^{\circ}\text{C}$
 $= (t - 25) \text{ kcal/kg}$ (1)

(2) The enthalpy of saturated water vapor at temperature $t^{\circ}\text{C}$
 $= (572.8 + 0.147t) \text{ kcal/kg}$ (2)
 (latent heat = $597.8 - 0.583t \pm 0.4 \text{ kcal/kg}$)

(3) The enthalpy of saturated steam at temperature $t^{\circ}\text{C}$ and at a pressure equal to the saturation pressure of steam at temperature $t^{\circ}\text{C}$ = $(572.8 + 0.417 t_8) + 0.46(t - t_8) \text{ kcal/kg}$ (3)

by taking specific heat of water vapor = $0.46 \text{ kcal/kg}^{\circ}\text{C}$ at very low pressures (0.01 to 0.1 atm)

(4) The specific heat of lithium bromide –water solution of concentration X is given by $C_x = 1.01 - 1.23X + 0.48X^2 \text{ kcal/kg}$ (4)

(5) The enthalpy of lithium bromide –water solution of concentration X kg LiBr/kg solution at 25°C is
 $H_{x,25} = 68.06 - 456.67X + 416.6 X^2 \text{ kcal/kg}$ (5)

where, H = specific enthalpy kcal/kg

(6) The enthalpy of LiBr-water solution of concentration X kg LiBr/kg solution at $t^{\circ}\text{C}$ is given by

$$H_{x,25} + C_x(t - 25) = (42.81 - 425.92 X + 404.67 X^2) + (1.01 - 1.23 X + 0.48X^2) (t) \quad (6)$$

(7) In the range of concentration from 0.50-0.65 kg LiBr/kg solution, it is possible to fit the refrigerant temperature t_R , the saturated solution temperature t_m and the concentration X by the relation $(t_R^{\circ}) = (49.04 - 134.65X) + (1.125 - 0.47X) (t_m^{\circ}\text{C})$

with standard deviation of $\pm 0.2\%$. This may be written as

$$X = (49.04 + 1.125t_m - t_R) / (134.65 + 0.47t_m) \quad (7)$$

(8) The saturated vapor pressure P in mm Hg corresponding to saturation temperature $T^{\circ}\text{K}$ for pure water is given by

$$\log_{10} P \text{ mm Hg} = 7.8553 - 1555/T - 11.2414X \cdot 10^4 / T^2 \quad (8)$$

Together with the assumptions of neglecting the pump work and neglecting the pressure drop in components and assigning saturation conditions to all the states numbers, the properties are determined as follows:

Applying the first law of thermodynamics to the evaporator will give

$$\dot{m}_R = Q_E / (H_{10} - H_8) \quad (9)$$

where \dot{m}_R , H_{10} and H_8 are the refrigerant flow rate, enthalpies of the states 10 and 8 respectively.

On the other hand, the lithium bromide mass balance in the absorber gives

$$\dot{m}_w X_6 + \dot{m}_R X_{10} = \dot{m}_s X_1 = (\dot{m}_{10} + \dot{m}_R) X_1 \text{ and by using equation (9), then}$$

$$\dot{m}_w = (Q_E / (H_{10} - H_8)) \cdot (X_1 / X_4) \quad (10)$$

$$\dot{m}_s = (Q_E / (H_{10} - H_8)) \cdot (X_4 / X_1) \quad (11)$$

Since the concentrations X_1 and X_4 are restricted not to exceed certain limits to avoid crystallization problems, and if the temperatures of the cycle are set to vary according to ambient and load conditions, the mass flow rates in the will be varied accordingly. This necessitates the existence of LiBr and water solution inventories to be used for flow compensation, especially at times when variations of load, hot water temperature, and cooling water temperature do occur.

The specific heat of the concentration X_1 and X_4 are determined from equation (4) as

$$C_{X1} = 1.01 - 1.23 X_1 + 0.48 X_1^2$$

$$\text{and } C_{X4} = 1.01 - 1.23 X_4 + 0.48 X_4^2 \quad (12)$$

The enthalpies H_1 and H_5 are then calculated using equation (6) as follows:

$$H_1 = (42.81 - 425.92 X_1 + 404.67 X_1^2) + (1.01 - 1.23 X_1 + 0.48 X_1^2) t_a \quad (13)$$

$$H_5 = (42.81 - 425.92 X_4 + 404.67 X_4^2) + (1.01 - 1.23 X_4 + 0.48 X_4^2) t_5 \quad (14)$$

The enthalpy of water vapor leaving the generator and entering the condenser (state 7) is determined by equation (3) as

$$H_7 = 572.8 + 0.46 t_g - 0.043 t_c \quad (15)$$

The heat balance of the condenser gives

$$Q_C = \{Q_E / (H_{10} - H_8)\} / (H_7 - H_8) \quad (16)$$

Heat balance for the combined generator and heat exchanger control volume gives:

$$Q_G = \dot{m}_{10} H_5 + \dot{m}_R H_7 - \dot{m}_8 H_2 \quad (17)$$

Since the pump work is negligible, then

$$H_1 \approx H_2 \quad (18)$$

Using equations (9), (10), (11) and (17), it is possible to write Q_G as

$$Q_G = \{Q_E / (H_{10} - H_8)\} \{ X_1 H_5 / (X_4 - X_1) + H_7 - X_4 H_1 / (X_4 - X_1) \} \quad (19)$$

Heat balance at the absorber gives Q_A as

$$Q_A = \dot{m}_{10} H_6 + \dot{m}_R H_{10} - \dot{m}_R H_1 \quad (20)$$

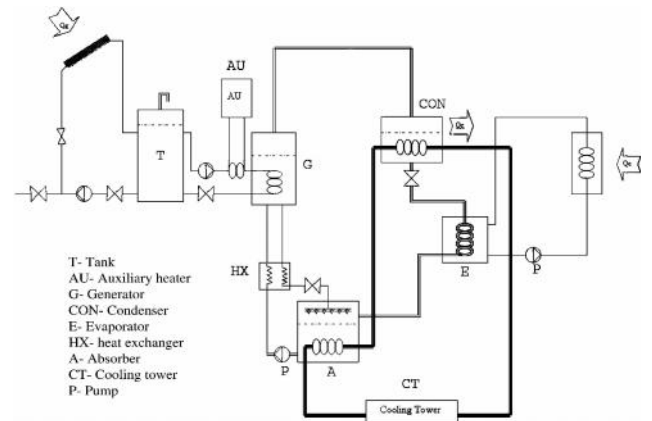
Equations (19) and (20) are governed by the first law of thermodynamics in the form

$$Q_C + Q_A = Q_G + Q_E \quad (21)$$

IV. Methodology of Absorption Cooling with Solar Energy

An absorption cooling systems with solar thermal energy is consists of incident solar energy, solar flat plate collectors, solar fluid, storage tank, cooling tower and a 3.5kW low temperature hot water driven absorption chiller. Fig.3 shows the schematic diagram of a solar powered absorption (single effect) cooling system. Here, the solar energy is gained through the collector and is accumulated in the storage tank. Then the hot solar fluid in the storage tank is supplied to the absorption chiller to start the cooling operation. An auxiliary heater is used as a backup source for the worst case.

The vital part of the solar flat plate collector is a flat absorber usually a blackened metal to which metal tubes filled with water are attached. When radiation is incident on the plate it warms up as it absorbs radiant energy and the heat then gets transferred to water increasing its temperature.



Black chrome is an important member of absorber coating. It has a number of properties¹⁰ which make it a useful functional finish. When applied to dull substrate there is a reduction of reflectivity of 94% whilst applied to bright surfaces, reduction of the order of 40 to 50% in reflectivity is obtained. Black chrome has a high absorptance to incident solar radiation with the wavelength below 3000nm and low emittance beyond this wavelength. (18)

V. Performance Analysis of the Absorption Chiller with Solar Energy

Performance of an absorption chiller is predicted from the coefficient of performance (COP). It is defined as the ratio

of the cooling effect at the evaporator to the heat input at the generator.

Therefore,

$$COP = Q_E / Q_G$$

It is simply derived from the equation (19) as

$$COP = \frac{(H_{10} - H_8) (X_4 - X_1)}{\{X_1 H_5 + (X_4 - X_1) H_7 - X_4 H_1\}} \quad (22)$$

If the loss of heat due the coupling between the outlet of the SWHS and the inlet of the generator of the absorption chiller is negligible then maximum coefficient of performance of the above absorption cycle is given by

$$(COP)_{max} = T_e(T_o - T_a) / T_e(T_c - T_e) \quad (23)$$

where, T_e , T_a , T_c and T_o are the absolute temperature of the evaporator, absorber, condenser and the outlet of the SWHS respectively.

VI. CO₂ Emission Factor for Fuel Mixture in Electricity Production in National Grid

Table. 2. GHG emission factor of the grid electricity from various sources

Fuel type	CO ₂ emission factor (kg/GJ)	CH ₄ emission factor (kg/GJ)	N ₂ O emission factor (kg/GJ)	Fuel conversion efficiency (%)	T & D losses (%)	GHG emission factor (tCO ₂ /MWh)
Natural gas	56.1	0.0030	0.0010	45.0%	0.0%	0.452
Large hydro	00.0	0.0000	0.0000	100.0%	0.0%	0.000
Diesel	74.1	0.0020	0.0020	30.0%	0.0%	0.897
Electricity mix						0.465

1 ton CH₄ = 21 tons CO₂, 1 ton N₂O = 310 tons CO₂

generation for the fiscal year 2004¹². For the fuel mixture the Green House Gas (GHG) emission per MWh energy production is shown in Table 2. emissions estimates of the non-CO₂ gases. Fig.4 presents the fuel mixture for grid electricity

VII. Results and Discussion

Based on this research, a set of mathematical models and computing programs have been developed using VBA (Visual Basic Application) and C++. Programs have been written in VBA language to study the performance of solar collectors and to calculate water outlet temperature. For the COP (Co-efficient of performance) of the single stage chiller, required programs have been coded in C++. The cost analysis and GHG emission analysis of the system have been estimated using RETScreen simulator. The program mentioned above was use to predict the performance of the system having an area of 30 m² flat-plate collector, storage

The 2006 IPCC Guidelines¹¹ estimate carbon emissions in terms of the species which are emitted. During the combustion process, most carbon is immediately emitted as CO₂. However, some carbon is released as carbon monoxide (CO), methane (CH₄) or non-methane volatile organic compounds (NMVOCs). Most of the carbon emitted as these non-CO₂ species eventually oxidises to CO₂ in the atmosphere. This amount can be estimated from the

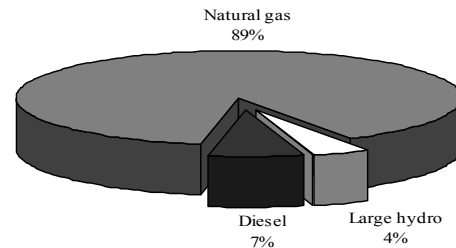


Fig. 4. Fuel mixture for grid electricity generation on the fiscal year 2004

and 3.5 kW low temperature hot source driven absorption chiller. The simulations are based on the monthly averaged hourly global horizontal irradiance (GHI) of Bangladesh. Fig 5 shows the hourly variation of the water outlet temperature in April, May and

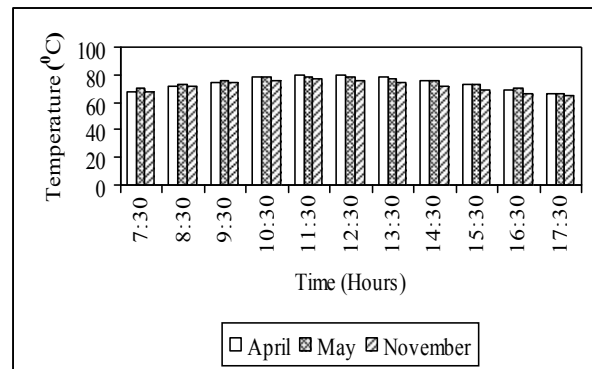


Fig. 5. Hourly variation of water outlet temperature.

November. It is evident that the water outlet temperature reached approximately 80°C in April. Fig.6 shows the hourly variation of instantaneous efficiency of the flat-plate solar collectors. It can be seen that the instantaneous efficiency of the collectors ranges from 0 to 44%. Fig.7 shows the hourly variation of the COP of the absorption chiller in the month of April, May and November. It can be seen that the COP of the chiller ranges approximately from 0.55 to 0.62. Fig.8 shows the hourly variation of the overall system efficiency, the product of the COP of the chiller and the instantaneous efficiency of the collectors. It can be seen the overall system efficiency ranges from 0 to approximately 27%. Fig.9 shows cumulative cash flow graph of the solar absorption air conditioning system that has been calculated using the RETScreen simulator.

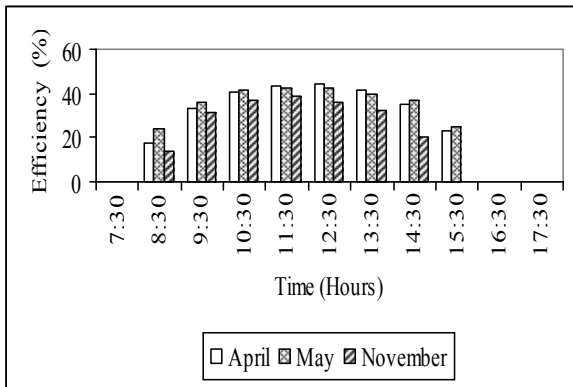


Fig. 6. Hourly variation of instantaneous efficiency

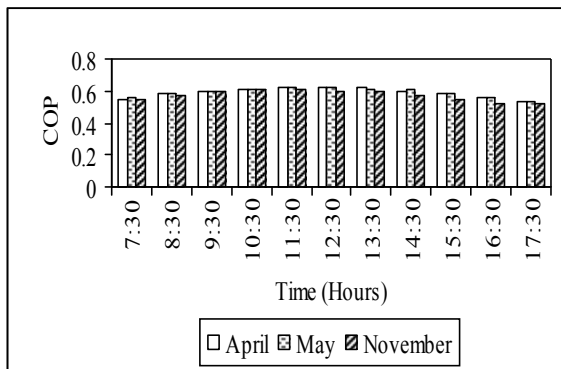


Fig. 7. Hourly variation of COP of the chiller.

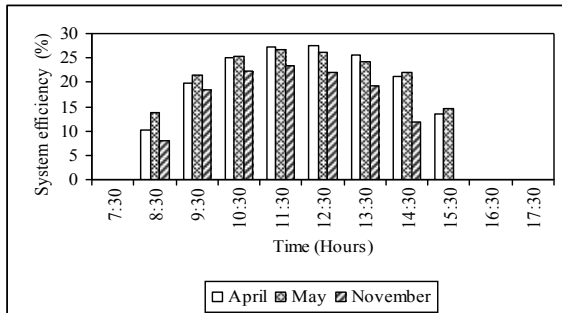


Fig. 8. Hourly variation of overall system efficiency.

Variations of the system performances with daily solar insolation have been analyzed for two months of summer and one month of winter along the typical year. The simulation of the units has been obtained by use of suitable computer codes. The local climatological data employed in the calculations were based on measurements over a number of years.

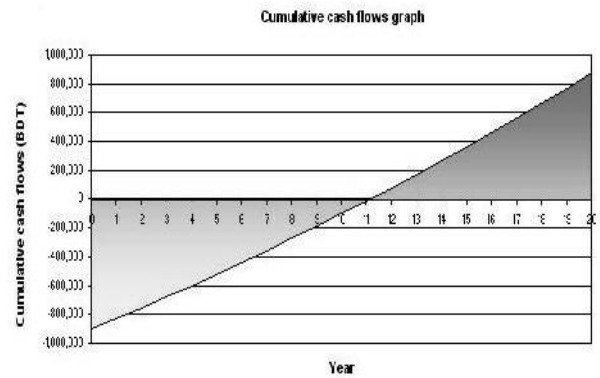


Fig. 9. Cumulative cash flow graph.

The flat plate collectors are designed by using black chrome as selective coating and these collectors have been tilt at latitude angle 23.7°. If the tilt angle is changed over the year, like 0° in summer and 40° in winter the system will be more efficient. It is seen from Fig.5 that the water outlet temperature from the collectors reaches at the range from approximately 65°C to 80°C which is sufficient to drive a low temperature hot source driven absorption chiller. The COP of the absorption chiller ranges approximately from 0.55 to 0.62. In Bangladesh about 96% grid electricity is generated from fossil fuel, which is emitting abundant CO₂ in the atmosphere. Every year 13,485 ton CO₂ emission would be mitigated by introducing solar cooling systems in stead of conventional electricity driven air-conditioning systems.

VIII. Conclusion

In spite of economical limitations, the greatest advantages of solar-powered air conditioning system when compared to other systems 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 cooling system. In fact, most of the buildings' cooling demands in summer are associated with high solar energy availability, which offers an opportunity to further exploit solar energy for cooling. It has also shown that the absorption chiller has performed reasonable satisfactory especially in May, having highest daily average GHI. The collector efficiency of maximum 44% is not excessively high but reasonable for its relatively high collector

temperature. One of the major problems is its inability to supply sufficient hot water from the storage tank to the chiller. This results in underutilization of the thermal energy collected by the collectors and a decrease in cooling capacity. In addition, substantial auxiliary heating is required in the morning. Despite the relatively large amount of energy collected by the collectors, the average cooling capacity of the chiller is below the rated capacity. Overall, the solar powered absorption air conditioning system has maximum 27% system efficiency. Its performance is quite better than the similar systems in Italy.

Nomenclature

C	specific heat of solution of concentration X
T	absolute temperature
H	specific enthalpy, kcal/kg
\dot{m}	mass flow rate, kg/hr
P	pressure, mm Hg
Q	rate of heat transfer, kcal/hr
t	temperature, °C
X	solution concentration, kg LiBr/kg solution

Subscripts

a, A	absorber
c, C	condenser
e, E	evaporator
g, G	generator
s	strong solution
w	weak solution

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