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Modelling and simulation of a solar absorption cooling system for India

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Abstract
This paper presents modelling and simulation of a solar absorption cooling system. In this paper, the modelling of a solar-powered, single stage, absorption cooling system, using a flat plate collector and water–lithium bromide solution, is done. A computer program has been developed for the absorption system to simulate various cycle configurations with the help of various weather data for the village Bahal, District Bhiwani, Haryana, India. The effects of hot water inlet temperatures on the coefficient of performance (COP) and the surface area of the absorption cooling component are studied. The hot water inlet temperature is found to affect the surface area of some of the system components. Moreover the effect of the reference temperature which is the minimum allowable hot water inlet temperature on the fraction of total load met by non-purchased energy (FNP) and coefficient of performance (COP) is studied and it is found that high reference temperature increases the system COP and decreases the surface area of system components but lower reference temperature gives better results for FNP than high reference temperatures.

Keywords: solar absorption, flat plate collector, water–lithium bromide solution, cooling, simulation

Nomenclature

A_c	collector area (m^2)
COP	coefficient of performance
FNP	the fraction of the total load met by non-purchased energy (%)
FR	flow ratio (X_A / (X_G – X_A))
h	specific enthalpy (kJ/kg)
H	enthalpy (J)
I	solar insolation (W/m^2)
k	heat conductivity (W/mK)
m	mass flow rate (kg/s)
Q	heat transfer rate (W)
Q_{aux}	auxiliary heater capacity (kW)
Q_L	extracted energy from the storage tank (kW)
Q_{load}	generator load (kW)
Q_u	useful energy (kW)
T	temperature (K)
T_{in}	collector inlet temperature (K)
T_o	the environment temperature (K)
T_{ref}	reference temperature (K)
T_s	storage tank temperature (K)
U_L	overall heat transfer coefficient (W/m^2K)
X	weight fraction of LiBr-H_2O solution
C_v	specific heat at constant volume
η_c	collector efficiency
Δt	time period (h)
τ	transmittance
α	absorptance

Subscript

f	liquid (fluid)
s	storage tank
v	vapour
w	water

1. Introduction

The energy demand for refrigeration and air-conditioning to control temperature and humidity and for the provision of fresh air has increased continuously during the last few decades especially in developing countries like India. This increase is caused amongst other reasons by increased thermal loads, occupant comfort demands, and architectural trends. This has been responsible for the escalation of electricity demand and especially for the high peak loads due to the use of electrically driven vapour compression machines. Moreover, the consumption of primary energy and the emissions of greenhouse gases associated with electricity generation from fossil fuels lead to considerable environmental consequences and monetary costs. Conventional energy will not be enough to meet the continuously increasing need for energy in the future. In this case, renewable energy sources will become important.

An alternative solution for this problem is solar energy, available in most areas and representing a good source of thermal energy. Of the various solar air conditioning alternatives, the absorption system appears to be one of the most promising methods (Wilbur & Mitchell 1975). The absorption cycle is similar in certain respect to the electrically driven vapour compression machines. Of the various solar absorption air conditioning systems, LiBr-H_2O and H_2O-NH_3 are the major working fluid pairs available in these systems. It is reported that the LiBr-H_2O pair has a higher COP than any other pair of the working fluids. The LiBr-H_2O system operates at a generator temperature in the range of 343 to 368 K with water used as a coolant in the absorber and condenser. The COP of the system is between 0.6 and 0.8 (Duffie & Beckman, 1991). The major components in the LiBr-H_2O solar absorption cooling systems are the chillers and solar collectors. Many researchers have developed solar assisted absorption refrigeration systems. Most of them have been produced as experimental units and computer codes were written to simulate the systems. Some of these designs are presented here.

Hammad and Audi (1992) described the performance of a non-storage, continuous, solar operated absorption refrigeration cycle. The maximum ideal coefficient of performance of the system was determined to be equal to 1.6, while the peak actual coefficient of performance was determined to be equal to 0.55.

Haim et al. (1995) performed a simulation and analysis of two open-cycle absorption systems. Both systems comprised a closed absorber and evaporator as in conventional single stage chillers. The regenerator, used to reconcentrate the absorber solution by means of solar energy, makes up the open part of the cycle. The analysis was performed with a computer code developed for modular simulation of absorption systems under varying cycle configurations (open- and closed- cycle systems) and with different working fluids. Based on the specified design features, the code calculates the operating parameters in each system. Results indicate a definite performance advantage of the direct-regeneration system over the indirect one.

Hawlader et al. (1993) developed a LiBr absorption cooling system employing an 11x11 m^2 collector/regenerator unit. They also developed a computer model, which they validated against actual experimental values with good agreement. The experimental results showed a regeneration efficiency varying between 38 and 67% and the corresponding cooling capacities ranged from 31 to 72 kW.

Ameel et al. (1995) gives performance predictions of alternative low-cost absorbents for open cycle absorption using a number of absorbents. The most promising of the absorbents considered was a mixture of two elements, lithium chloride and zinc chloride. The estimated capacities per unit absorber area were 50–70% less than those of lithium bromide systems.

In order to improve the system design of a solar powered absorption air conditioning system, a parametric study must be carried out to investigate the influence of key parameters on the overall system performance. If experiments were used to perform the parametric study, effects of one key parameter on the overall system performance would normally require several cooling seasons and hence years to establish a conclusion. Also, it is extremely difficult to keep the performance of the system components constant over the entire experimental period as the components deteriorate with time. Therefore, it is extremely difficult and expensive to carry out experiments to investigate the influence of all the key parameters on the overall system performance of an existing solar cooling system. The objective of this study is to model a complete solar absorption cooling system, comprised of a solar collector, storage tank and a LiBr–water absorption chiller and to develop a simulation program for parametric study of this system. The program is used to simulate the cooling system with the help of meteorological data for Bahal, Dist Bhiwani, Haryana (India), as shown in Table 1.

In the present work, modelling and simulation is done for proper sizing of flat-plate solar collectors with a water storage tank to match the load needed to the generator of the absorption system. The performance of a 10.5 kW solar driven LiBr-H_2O absorption cooling system is investigated numerically. In section 2, mathematical modelling of a solar absorption cooling system is provided. In the mathematical modelling section, modelling of an absorption system as well as that of the solar collector system is also provided. Section 3 presents the results and discussion pertaining to the validation of code. Section 4 concludes the work.

2. Mathematical model of solar absorption cooling system

Figure 1 shows the schematic diagram of a basic solar absorption cooling system. This system has

Table 1: Solar insolation, I (W/m^2), for Bahal, India

Time	Solar insolation (W/m^2)				
	May	June	July	August	Sept.
7:00	119.44	113.88	77.77	77.77	44.44
8:00	288.88	261.10	227.77	208.32	166.66
9:00	480.54	427.76	377.76	358.32	297.21
10:00	633.31	558.31	458.32	455.54	419.43
11:00	755.53	666.6	530.54	541.65	522.20
12:00	799.97	711.09	580.53	583.31	563.87
13:00	805.53	716.64	608.31	594.42	583.31
14:00	755.53	647.20	561.09	530.54	522.20
15:00	597.20	538.87	474.98	491.65	413.87
16:00	441.65	372.21	386.10	363.87	333.32
17:00	261.10	238.88	233.32	191.66	197.21
18:00	105.55	99.99	111.10	69.44	55.55

been the basis of most of the experience to date with solar air conditioning. Here, the solar energy is gained through the collector and is accumulated in the storage tank. Then, the hot water in the storage tank is supplied to the generator to boil off water vapour from a solution of Lithium Bromide and water. The water vapour is cooled down in the condenser and then passed to the evaporator where it again is evaporated at low pressure, thereby providing cooling to the required space.

Meanwhile, the strong solution leaving the generator to the absorber passes through a heat exchanger in order to preheat the weak solution entering the generator. In the absorber, the strong solution absorbs the water vapour leaving the evaporator. Cooling water from the cooling tower removes the heat by condensation. Since the temperature of the absorber has a higher influence on the efficiency of the system than the condensing temperature, the heat rejection (cooling water) fluid, is allowed to flow through the absorber first and then to the condenser. An auxiliary energy source is provided, so that hot water is supplied to the generator when solar energy is not sufficient to heat the water to the required temperature level needed by the generator. The mathematical modelling of the complete solar assisted absorption cooling system requires modelling of solar collector systems separately.

2.1 Absorption cooling system modelling

Energy balance

The system is modelled by treating each of the major components i.e. the generator, condenser, evaporator, heat exchanger and absorber as single control volumes. The heat addition or rejection at each component is then obtained by taking an energy balance across each component.

The rate of heat addition in the generator is the energy input to the cycle is given by the following equation:

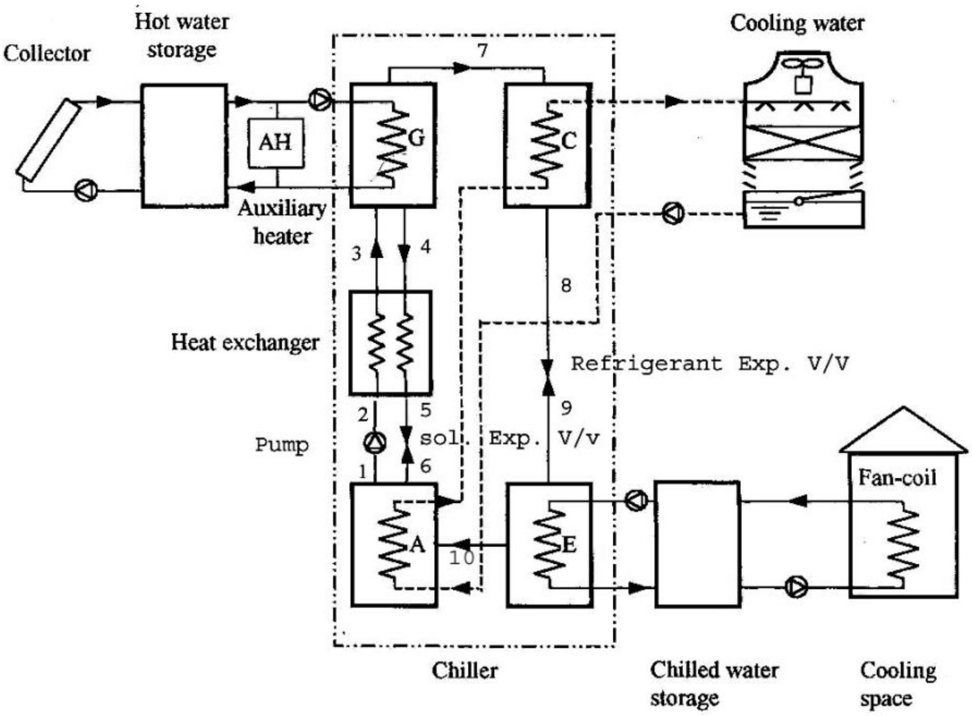

Figure 1: A basic solar absorption cooling system

$$Q_{gen} = m_7 h_7 + m_4 h_4 - m_3 h_3 \quad (1)$$

The rate of heat rejection out of the condenser is given by the following equation:

$$Q_{con} = m_7 h_7 - m_8 h_8 \quad (2)$$

The rate of heat absorption in the evaporator is given by the following equation:

$$Q_{evap} = m_{10} h_{10} - m_9 h_9 \quad (3)$$

The rate of heat rejection by the absorber is given by the following equation:

$$Q_{abs} = m_{10} h_{10} + m_6 h_6 - m_1 h_1 \quad (4)$$

An energy balance on the hot side of the solution heat exchanger is given by the following equation:

$$Q_{shx\text{-}h} = m_4 h_4 - m_5 h_5 \quad (5)$$

Similarly an energy balance on the cold side of the solution heat exchanger is given by the following equation:

$$Q_{shx\text{-}c} = m_3 h_3 - m_2 h_2 \quad (6)$$

The overall energy balance on the solution heat exchanger is satisfied if $Q_{shx\text{-}h} = Q_{shx\text{-}c}$ which is seen to be the case here.

Coefficient of performance (COP) is defined according to Figure 1 as follows:

$$COP = Q_{evap}/Q_{gen} \quad (7)$$

Usually the enthalpy of the refrigerant (water) is calculated by using the saturated and superheated property tables. The following formulations suggested by Stoeker and Jones (1982) have a great advantage for computer simulation of the system performance.

The enthalpy at exit of condenser (State 8) is given as:

$$H_8 = h_f(T_c) = A + BT_c + CT_c^2 \quad (8)$$

The enthalpy at the exit of the evaporator (State 10) is given as:

$$H_{10} = h_v(T_e) = A + BT_e + CT_e^2 \quad (9)$$

The enthalpy at the exit of the generator (State 7) is given as:

$$H_2 = h_v(T_g) = A + BT_g + CT_g^2 \quad (10)$$

The enthalpies of the solution of the refrigerant and the absorbent (H_2O-LiBr) at states 1 and 4 are expressed in a multi-variable polynomial in terms of the solution temperature, T (°C), and solution concentration, X, through the absorber and generator unit as given in ASHRAE charts by:

$$h = \sum_0^4 A_n X^n + T\sum_0^4 B_n X^n + T^2 \sum_0^4 C_n X^n \quad (11)$$

Where A, B and C are constant coefficients and are listed in ASHRAE (1985).

2.2. Solar energy modelling

The useful solar energy collected is transferred to the hot liquid storage tank from which the generator of the absorption system is supplied with input thermal energy. Necessary equations for modelling solar energy are taken from Duffie and Beckman (1991).

$$Q_u = A_c I \eta_c \quad (12)$$

Where Q_u is the useful energy collected in system collectors, A_c is the collector area, I is solar insolation and η_c is the collector efficiency;

$$\eta_c = F_R[(\tau\alpha) - U_L \frac{T_{in} - T_o}{I}] \quad (13)$$

In this equation, F_R is the collector heat removal factor, $(\tau\alpha)$ is the transmittance-absorptance product, U_L is overall heat transfer coefficient, T_{in} is the collector inlet temperature and T_o is the environment temperature. Perfect mixing within the tank is assumed. If the rate of heat addition and removal for a reasonable period of time Δt are assumed to be constant, equations can be written for each time interval:

$$T_s^{new} = T_s + \frac{\Delta t}{m_s C_{vw}}[Q_u - Q_L - Q_s] \quad (14)$$

where $Q_s = (UA)_s(T_s - T_o)$, Q_L is the extracted energy from the storage tank, T_s is the main storage temperature for the period and m_s is the storage tank mass. $(UA)_s$ is taken as 11.1 W/K as suggested by Duffie and Beckman (1991). In spite of the fact that $(UA)_s$ depends on the surface area of the storage tank, the average value of both $(UA)_s$ and the surface area of the storage tank are assumed on the basis of the fact that the surface area of the storage tank shows little change for selecting storage tank mass. Here, the heat transfer coefficient is assumed to be 0.72 W/m^2K.

An auxiliary heater at the exit of the storage tank boosts the temperature of the hot water from the storage tank temperature to the allowable reference temperature when the storage tank temperature drops below the allowable reference temperature. The auxiliary heater capacity is calculated as follows:

$$Q_{aux} = mC_{pw}(T_{ref} - Ts) \quad (15)$$

Where m is the mass flow rate used by the generator and T_{ref} is the minimum allowable hot water inlet temperature required for the generator to boil off the refrigerant from the refrigerant-absorbent solution.

The fraction of the total load met by non-purchased energy FNP is calculated as:

$$FNP = 1 - Q_{aux}/Q_{load} \quad (14)$$

where Q_{load} is exactly the same as Q_{gen} which is the generator load.

Input data required
The inputs data required for simulating the system consists of the following: system nominal capacity, generator temperature range, condenser temperature range, evaporator temperature range, absorber temperature, ambient temperature range, generator and condenser pressure, evaporator and absorber pressure, mass flow rate from generator to condenser, mass flow rate from absorber to generator, mass flow rate from generator to absorber. The system has been designed for 10.5 kW constant cooling load. The outputs include the surface area of the system components, coefficient of performance (COP) and the fraction of the total met by non-purchased energy (FNP).

3. Results and discussion

3.1 Validation

The computer program developed has been validated by comparing some of its results (FNP) with the similar study by Ileri (1995). It can be seen from Table 2 that, as compared with Ileri's 1995 study, the results of present study for reference temperature effects show the same general behaviour. Similarly, Ileri's study shows that a 353 K reference temperature is the best solution.

Table 2: Comparison of FNP for different reference temperature of the present study with those of Ileri's 1995 for the month of August

T_{ref} (K)	Present study FNP (%)	Ileri FNP (%)
353	100	94
358	89	–
363	69	58
368	–	56

Clearly, it can be said that there is a good agreement between the two studies.

3.2 Simulation results

Simulation results are discussed in this section for the performance of a 10.5 KW solar driven lithium bromide absorption cooling system. Figure 2 depicts the effect of the hot water inlet temperatures T_s on the system COP and flow ratio FR. It can be seen that an increase in this temperature resulted in the decreases of FR. This is due to increases in the mass fraction of concentration solution (X_G). While with an increase in this temperature, COP increases.

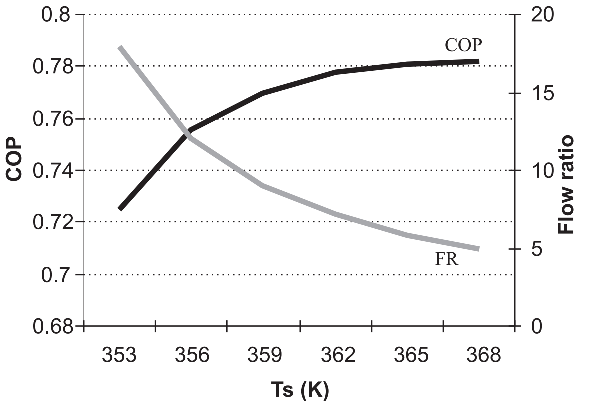

Figure 2: The effect of the hot water inlet temperatures on the system COP and FR (T_e = 280 K, Q_L = 10.5 KW, T_c = 306K)

Figure 3 depicts the effect of the hot water inlet temperature on the surface area of the system components. It can be seen that increase in this temperature results in the decrease of the absorber and solution heat exchanger surface area. As flow ratio decreases, the thermal energy extracted from the absorber also decreases and hence the temperature of the absorber increases, which further resulted in the increase of logarithmic mean temperature difference (ΔT_m) in the absorber and solution heat exchanger. By decreasing the heat capacity and increasing ΔT_m, heat transfer surface area normally decreases in these components.

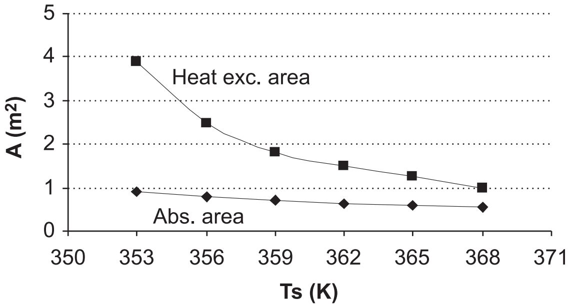

Figure 3: The effect of the hot water inlet temperatures on the surface area of the system components (Te = 280 K, QL = 10.5 KW, Tcool, in = 291 K)

Figure 4 depicts the effect of the reference temperatures on FNP. The investigation is carried out between 353 and 368 K because the increment in the system COP slows down considerably after a certain temperature. Furthermore, boiling occurs above 368 K and this is not desirable. To investigate the effect of storage tank reference temperature on the FNP, three reference temperatures of the storage tank, 353, 358 and 363 K were taken into consideration. It can be seen from Figure 4 that the 353 K reference temperature seems to be the best. It gives better results than that of the higher reference temperatures. In this study, a reference temperature below 353 K is not investigated, because it could cause a decrease in the system COP. However, below the 353 K reference temperature, insufficient refrigerant vapour is driven from the liquid solution.

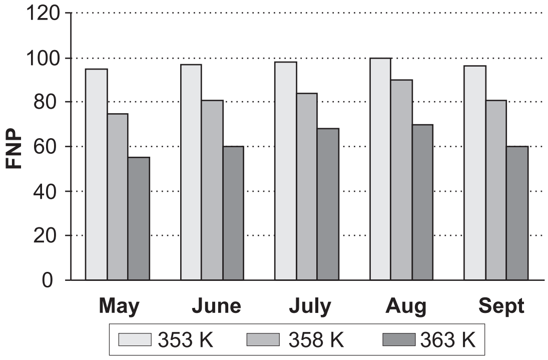

Figure 4: The effect of the reference temperatures on FNP

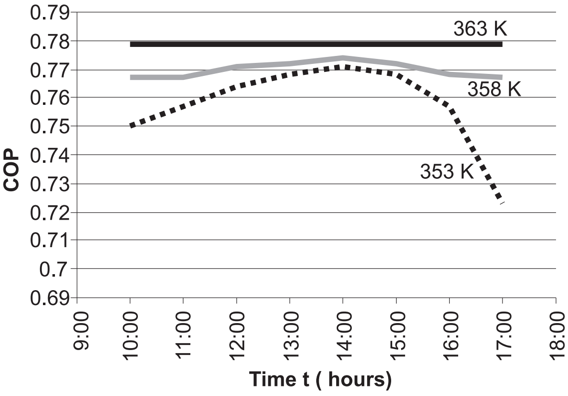

Figure 5: The effect of the reference temperatures on the system COP for a day in May

Figure 5 depicts the effect of the reference temperatures on the system COP for a day in May. As shown, unlike the effects caused by reference temperature on FNP, increasing the reference temperatures boosts the system COP. Also, COP remains unchanged at this high reference temperature of 363 K.

4. Conclusion

From the above study the following results can be drawn.

1. The hot water inlet temperature is found to affect the surface area of some of the system components. Increasing this temperature decreases the absorber and solution heat exchanger surface area, while the dimensions of the other components remain unchanged.
2. Although high reference temperature increases the system COP and decreases the surface area of system components, lower reference temperature gives better results for FNP than high reference temperatures do. For this study, a 353 K reference temperature is the best choice.

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