Mathematical modeling and performance analysis of a solar assisted vapour absorption refrigeration system

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Abstract. Climate change is the hottest topic today on earth. Traditional HVAC's harm the environment and stand out to be a major contributor to global warming. One of the many techniques to mitigate the current climate crisis is by utilizing the abundantly available solar radiation. Solar evacuated tube collectors are used to absorb solar radiation. This solar power is used to provide solar cooling technologies. Unlike the conventional HVAC's thermally driven absorption cooling is a boon in today's era. In the following research work, the study aims to provide numerical modeling of a solar assisted air-conditioning unit through the first law of thermodynamics using mass balancing equations for the province of Delhi in July. This research work's aim is to build and evaluate a Li-Br absorption cooler with a capacity of 1 KW. This paper aims to provide characteristic design features of components and measures the efficiency of the system at operating conditions. PCM based thermal energy storage tank is adopted which helps to supply cold air even at no sunlight hours. As a result of this research study, such solar assisted air conditioners turn out to be a chosen solution for sustainability.

1 Introduction

Greenhouse gas emissions are directly responsible for the rising temperatures worldwide. Heating, ventilation air conditioners have large energy acquisition and pose a negative impact on the local environment. They have raised tremendous concerns about the current climate threat as they are responsible for ozone depletion and global warming. Many organizations are searching for ways to fight climate change and resorting to sustainable alternatives. Reciprocating sorption chillers can be one method of limiting the usage of conventional systems and hence reducing reliance on fossil fuels. In the 1950s, numerous nations developed tiny sun absorption refrigeration systems. In the area of solar refrigeration employing solar collectors (flat plate), American Professor G.O.G.L. is well known for his contribution [1]. Sorption cooling includes both absorption and adsorption. Absorption machines do not require much of input power (shaft power) as they are

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thermally driven. In this way, absorption machines offer dependable and efficient cooling in areas with expensive or unreliable power sources, or in areas with access to waste, gas, geothermal, or solar heat. [2]. Traditional systems incorporate mechanical compression machines using electrical energy consumption whereas in absorption cooling thermal compression of refrigerant is employed. This area of research focuses discussion on an absorption system, which uses a refrigerant - absorbent pair that work together. The most commonly chosen working pairs are (1) lithium bromide – water, where the former is the absorbent and the latter is the refrigerant. It is one of the widely used pairs for space air-conditioning applications, (2) ammonia - water, where the refrigerant is ammonia & absorbent being water, but the lower efficiency of NH_3-H_2O compared with LiBr-H_2O restricts their use. NH_3 is poisonous and is not fit for residential applications.

[3] performed a life cycle analysis to examine the viability of the proposed design. COP achieved is bit lower but the system is economically friendly as compared to conventional HVAC's. [4] examined the system through a computational simulation using MATLAB. It incorporates analysis through 1st and 2nd law of thermodynamics. Various trends between the COP & temperatures of generator, absorber, and condenser are presented. [5] studies an ammonia - water absorption system for analysis. C-program is developed to find the thermodynamic properties of each component. Effect of various key parameters (eg. concentration effect, change in pressure) on performance of the structure is examined with graphs. [6] study the integration of PCM's with HVAC's to reduce energy consumption leading to maximization of the system's performance. The study also involves the making of 2 different systems and their efficiency is investigated. [7] includes construction and parameter analysis for each component of solar air conditioner. COP is calculated for two different working temperatures of the generator. A small thermal storage tank is also incorporated.

The study's goal is to analyse a LiBr-H_2O thermally driven absorption cooler which absorbs solar thermal energy using evacuated tube collectors. Practical problems that are faced by such an arrangement include crystallization, pressure drops & air leakage. To prevent crystallization to occur, the pressure of the condenser needs to be maintained at a particular value, irrespective of the temperature of the cooling water. This is achieved by monitoring the cooling water flow rate moving towards the condenser. Sometimes, addition of additives takes place to inhibit crystallization [1].

This paper reports design characteristics of different components of the system and presents a thermodynamic energy analysis of the vapour absorption refrigeration system under varied working conditions.

2 Geography description

The thermally powered solar absorption air-conditioning unit is planned and implemented for the province of Delhi, India. The site is located at 28.6° north latitude and 77.2° east longitude. In this area the average incident solar radiation is approximately 0.5 KWh/m2/day. In Delhi, for the month of July average ambient air temperature is 35℃ and humidity is 50% for the same. Under the current climatic conditions space air conditioning is required for human comfort.

3 System layout

The solar air conditioning system mainly comprises of evacuated tube solar collectors, a PCM thermal-storage unit & a vapor-absorption machine. The system's design and layout is depicted in Fig. 1.

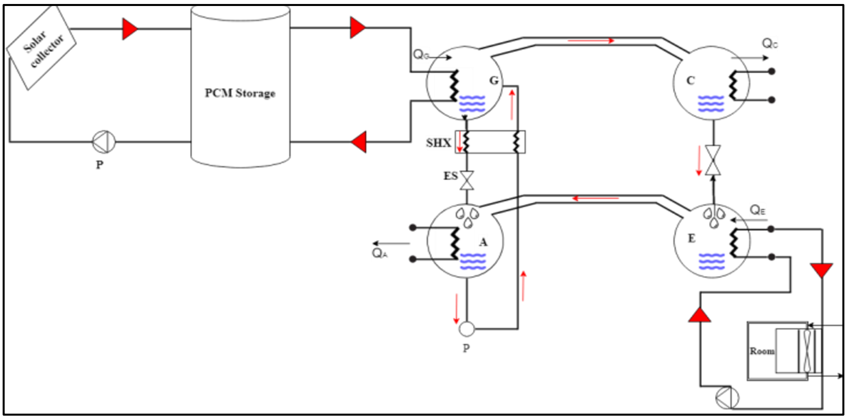

Fig. 1. Design layout for solar assisted water - libr absorption cooling system

4 Solar cycle system design

4.1 Solar collector

One of the crucial elements of a solar assisted air conditioning system, the solar collector, transforms the solar energy into thermal energy which further helps to drive the absorption chiller [8]. Evacuated tube collectors are used to absorb heat from sun's radiation. The heat transfer fluid incorporated for the solar system array is water because of a large value of thermal conductivity and heat capacity. Solar thermal energy then converts into heat output by the ETC collectors, which then uses that heat to warm water passing through them. In order to lessen losses from convection and re-radiation, the ETC collector is vacuumed [9]. The ETC collectors are placed under the sunlight to reach a temperature of 100°C. Once the given temperature is reached, the cycle starts. The amount of heat absorbed does not equal the actual heat that is transmitted to the heat transfer liquid because of the energy losses associated with the heat-transfer from the absorbent to the tubes.

Heat acquired by the solar collector is the useful heat required as an input to the generator in the absorption cycle. The following assumptions are adopted for the solar cycle system:

- Ambient temperature of the environment is 35°C and the pressure is 1 atm.
- A temperature difference of about 20°C is considered between the collector and the generator because of the losses that will take place.
- Area of each evacuated tube is 0.03 m^2.
- Efficiency of ETC is 0.70.
- Inlet heating water temperature to the generator is 92°C.
- Outlet temperature of water from generator is 88°C.

To find dimensions of the solar collector the following approach is used. The required area for the solar collector per cooling capacity is given as [10]:

1) Incident solar radiation for Delhi (G) = 0.5 $KWh/m^2/day$
2) Efficiency of solar collector (η) = 0.7 or 70%
3) Chiller COP = 0.76 or 76%

$$A = \frac{1}{G \cdot \eta \cdot COP} \frac{(m^2)}{(KW)}$$

$$= \frac{1}{0.5 * 0.7 * 0.76} \frac{(m^2)}{(KW)}$$

= 3.75 m^2 per KW

For 1KW absorption chiller, 3.75 m^2 coverage of area of the evacuated tubes is required. Area of each evacuated tube is 0.03 m^2 .

$\therefore 3.75 \div 0.03$

= 125 evacuated tubes are required.

4.2 PCM storage

PCMs are materials that are ideal for energy transfer and conservation purposes as they are able to absorb, store, and emit a significant quantity of heat energy at fairly consistent temperatures [11], [12]. These are integrated to enhance the operational efficiency of solar assisted air conditioners. A thermal energy storage system is installed between ETC and the vapour absorption system to stock the thermal energy with the aid of PCMs. The primary elements of this unit are a solar collector & a storage tank. A cylindrical vessel which is filled with a dense bed of PCM spheres, serves as the storage unit [13]. Heat transfer during charging & discharging process takes place b/w the PCM and the heat-transfer fluid. Direct cooling occurs in the morning and early afternoon. The excess energy is utilized to create and stock up cooling energy in PCM once the specified temperature is reached and there is enough radiation of sun. The PCM storage is discharged in the night [14].

5 Thermally driven vapour absorption cooling system

5.1 Working principle of the cycle

The main components in the design of a vapour-absorption cycle include ‘generator’, ‘condenser’, ‘refrigerant expansion valve’, ‘evaporator’, ‘absorber’, ‘solution pump’, ‘solution expansion valve’ and ‘solution heat exchanger’.

The heat gained by the heat-transfer fluid in the solar cycle is further utilized by the system’s absorption unit for the purpose of cooling. In the generator, high temperature heat transfer fluid transfers its heat to the working pair, LiBr-H_2O solution which is pumped from the absorber. Hence, heat gain takes place in the generator. Water (refrigerant) evaporates from solution and this high pressure super-heated steam moves further to the condenser whereas the remaining strong solution with high percentage of Li-Br moves towards the absorber with the help of a solution heat exchanger. Inside the condenser, heat rejection takes place and the refrigerant changes to a high pressure saturated liquid. Pressure decreases as the saturated liquid moves through the expansion valve. Then it further enters the evaporator, where evaporation occurs and cooling takes place. Low pressure saturated-vapour exits through the evaporator, moves and enters the absorber where the solution present absorbs it. The heat gathered in the evaporator is released when the refrigerant vapour gets absorbed and is condensed from vapour to a liquid state. The heat generated by the condensation of vapour of the refrigerant, caused due to the absorption in mixture, is carried away through the cooling water as it moves along the bundle of absorber tubes. The weak absorbent mixture is then pushed towards the generator to drive off the refrigerant by heating it up [15].

6 Numerical modelling of the absorption system

6.1 Assumptions for thermodynamic analysis

- Steady-flow & steady-state analysis.
- Negligible changes in kinetic & potential energy are considered across each component.
- Pure refrigerant i.e. water, only boils in the generator.
- Cooling in room is provided at a temperature of 21℃, which lies in the temperature range for thermal comfort (20℃ - 25℃).
- There are two working pressures involved ($P_{evaporator}$ and $P_{condenser}$). Expansion valve reduces the working pressure of fluid from $P_{condenser}$ to $P_{evaporator.}$
- No temperature loss occurs between the expansion valve and the evaporator.
- Temperature of cooling coil of the evaporator is 15℃. Since losses are involved the working fluid has a temperature of 18℃. After leaving the cooling coil the working fluid gains about 3-4℃.

6.2 Design parameters for the absorption cycle

Fig. 2 represents a schematic diagram of the absorption cycle.

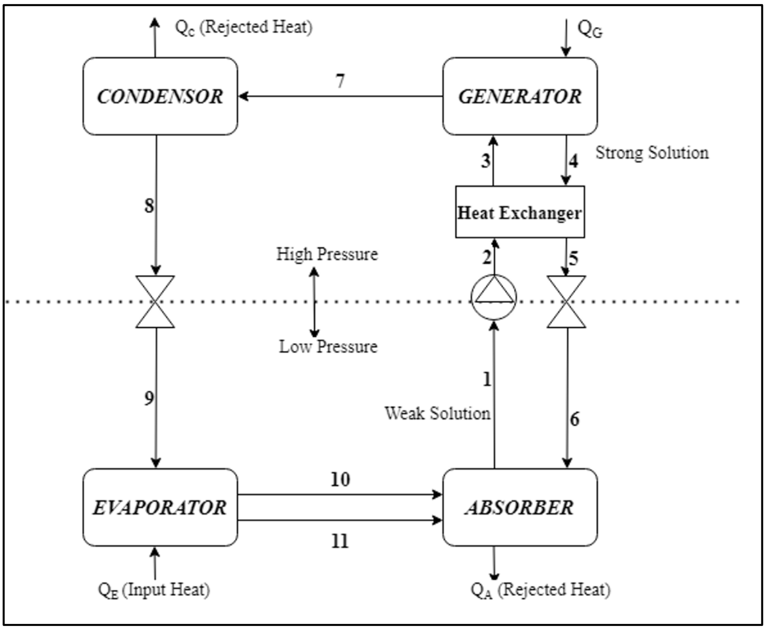

Fig. 2. Schematic diagram for absorption cooling system

Table 1 describes the operating conditions and temperatures of components that are adopted for carrying out the thermodynamic analysis.

Table 1. Design parameters

Parameters	***Symbol***	***Value***
Evaporator temperature	T_{10}	15°C
Generator solution exit temperature	T_4	75°C
Weak solution mass fraction	x_1	55% LiBr
Strong solution mass fraction	x_4	60% LiBr
Solution HX exit temperature	T_3	55°C
Generator (desorber) exit temperature	T_7	70°C

6.3 Thermodynamic model

Governing equations of mass balancing for a steady state & a steady flow model:
$\sum \dot{m}_\iota - \sum \dot{m}_o = 0$

Through the 1st law of thermodynamics and using mass balancing equations for every part :
$\sum (\dot{m}h)_i - \sum (\dot{m}h)_o + \sum Q_i - \sum Q_o + W = 0$

$Also,\ Q_C + Q_A - Q_G - Q_E = 0$

System calculations are performed at each point in Fig. 2 to specify values of P, T and h.

- Point 10:
  In the evaporator, the refrigerant is saturated water vapour.
  Temperature = T_{10} = 15°C and saturation pressure = 1.7051 KPa
  Enthalpy = h_{10} = 2528.9 KJ/Kg {from steam table}
- Point 11:
  Refrigerant is a saturated liquid.
  h_{11} = 62.98 KJ/Kg
- Point 9:
  T_9 = 15°C and P_9 = 1.7051 KPa
  $h_9 = h_8$ {since there is no loss in the expansion valve}
  h_9 = 131 KJ/Kg
- Point 8:
  At point 4, solution mass fraction = 60% Li-Br (strong solution) and temperature at the saturated state = 75°C
  → Saturation pressure = 4.82 KPa {LiBr-water chart}
  h_4 = 183.2 KJ/Kg
  At point 8, saturated liquid from condenser comes out. Temperature of condenser = 31.5°C
  Pressure at point 8 = pressure at point 4
  ∴ h_8 = 131 KJ/Kg {from steam table}
- Point 7:
  Exit temperature of superheated steam from the generator = 70°C

Pressure = 4.82 KPa
$h_7 = 2612.2$ KJ/Kg {from steam table}

- Point 6:
  Strong solution is obtained through the expansion valve and involves no losses.
  $T_6 = 44.5°C$
  P = 1.7051 KPa
  $\therefore h_6 = h_5$
  $h_6 = 115.235818$ KJ/Kg
- Point 1:
  Input mass fraction of saturated liquid = 55% {weak solution}
  Exit temperature of liquid = Temperature of Absorber = 34.9°C
  P = 1.7051 KPa
  $h_1 = 62.3995$ KJ/Kg {from steam table}
- Point 2:
  Pump work = $\dot{m}_2 h_2 - \dot{m}_1 h_1$
  $\dot{m}_2 = \dot{m}_{1,}\ x_2 = x_1$
  P = 4.82 KPa
  Since the work done by the pump is neglected.
  $\therefore h_2 = h_1 = 62.3995$ KJ/Kg
- Point 3:
  Exit temperature from solution heat exchanger = 55°C
  P = 4.82 KPa
  $h_3 = 124.7$ KJ/Kg {state: may be a sub-cooled liquid}
- Point 4:
  Generator solution exit temperature : $T_4 = 75°C$
  P = 4.82 KPa
  $h_4 = 183.2$ KJ/Kg {LiBr chart}
- Point 5:
  Exit Temperature of strong solution: $T_5 = 51.5°C$, P = 4.82 KPa
  Applying energy balance equation across solution heat exchanger:
  $\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5$
  $h_5 = 115.235818$ KJ/Kg

Now,
Performing calculations across each component of the absorption cooler:
Capacity of single-effect water Li-Br absorption cooler = $\dot{Q}_E$ = 1KW {assumed}
Let, $\dot{m}$ = mass flow rate of the refrigerant (Kg/sec)
$\dot{m}_{ss}$= mass flow rate of the strong solution
Circulation ratio= $\lambda = \frac{\dot{m}_{ss}}{\dot{m}}$

1) Evaporator
   $\dot{Q}_E = \dot{m}_{10} h_{10} + \dot{m}_{11} h_{11} - \dot{m}_9 h_9$
   $\dot{m} = \dot{m}_{10} = 0.000417\ Kg/sec$
   $\lambda = \frac{\dot{m}_{ss}}{\dot{m}} = \frac{x_{ws}}{x_{ss} - x_{ws}} = \frac{55}{60-55} = 11$
   $\therefore Strong\ solution: \dot{m}_{ss} = \lambda \dot{m} = 0.004587\ Kg/sec$
   $Weak\ solution = \dot{m}_{ws} = (\lambda + 1)\dot{m} = 0.005004\ Kg/sec$
2) Absorber
   $Q_A = \dot{m}_6 h_6 + \dot{m}_{10} h_{10} + \dot{m}_{11} h_{11} - \dot{m}_1 h_1$
   $\therefore Q_A = 1.271546\ KW$ {heat rejected}
3) Pump work is neglected

4) <u>Solution expansion valve</u>
There is no loss. → $\dot{m}_5 = \dot{m}_6, x_5 = x_6$
$\therefore h_5 = h_6$
5) <u>Solution heat exchanger</u>
$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5$
$\therefore h_5 = 115.2358\ KJ/Kg$
6) <u>Generator</u>
$Q_G = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3$
$\therefore Q_G = 1.305\ KW$ {heat input to the generator}
7) <u>Condenser</u>
$Q_C = \dot{m}_7 h_7 - \dot{m}_8 h_8$
$\therefore Q_C = 1.034\ KW$ {heat rejected}
8) <u>Refrigerant Expansion Valve</u>
Since there is no loss.
$\dot{m}_8 = \dot{m}_9$ and $x_8 = x_9$
$\therefore h_8 = h_9$

The COP for the absorption system is described through the following relation:

$$COP = \frac{Q_E\,(\text{evaporator heat load})}{Q_G(generator\ heat\ load)}$$

$\therefore \boldsymbol{COP = 0.76}$

6.4 Design of the generator heat-exchanger

Latent-heat of vaporization and sensible heat is obtained as an output when heat is fed into the generator. Experiments reveal that for such systems, the value obtained for heat-transfer coefficient, U ranges between 1600 -7500 W/m^2-°C. To evaluate the efficiency of a HX, the value of $(U.A)$ is used to represent the best heat transfer rate.

Employing single pass heat exchanger, the temperature b/w hot & cold fluid doesn't remain constant and varies with the distance along the heat-exchanger. For a HX that has two ends at which hot and cold streams enter or exit on either side, the logarithmic mean temperature difference is:

$$LMTD = \frac{\Delta T_O - \Delta T_I}{\ln(\frac{\Delta T_O}{\Delta T_I})}$$

Heat transfer is given by: $Q = U.A.LMTD$

$\Delta T_O = T_{12} - T_{13} = 92 - 55 = 37°C$
$\Delta T_I = T_7 - T_4 = 5°C$
$\therefore LMTD = 15.98°C$

Area of each evacuated tube is 0.03 m^2

$$U = \frac{Q_G}{A.LMTD}$$

∴ Heat transfer coefficient = 2711.72 W/m^2-°C, which lies between the said values.

7 Results

Results for the above research are tabulated in Table 2 for thermally driven water - libr absorption cooling system. Results for the above research are tabulated in Table 2 for thermally driven water - libr absorption cooling system. At each point h, P and T are specified along with the description of each part present in the absorption system. The COP evaluates the air-conditioner capacity. COP based on generator temperature = 75℃ is calculated to be equal to 0.76.

Table 2. Water-LiBr absorption cooling system calculations

Point	*h (KJ/Kg)*	$\dot{m}$ *(Kg/sec)*	*P (KPa)*	*T (℃)*	*%LiBr (x)*	*Remarks*
1	62.3995	0.005004	1.7051	34.9	55	
2	62.3995	0.005004	4.82	34.9	55	
3	124.7	0.005004	4.82	55	55	**'sub cooled liquid'**
4	183.2	0.004587	4.82	75	60	
5	115.235	0.004587	4.82	51.5	60	
6	115.235	0.004587	1.7051	44.5	60	
7	2612.2	0.000417	4.82	70	0	**'superheated steam'**
8	131	0.000417	4.82	31.5	0	**'saturated liquid'**
9	131	0.000417	1.7051	15	0	
10	2528.9	0.000417	1.7051	15	0	**'saturated vapour'**
11	62.98	0.000010	1.7051	15	0	**'saturated liquid'**
Description			***Working Temperature***			***KW***
Evaporator (Capacity)			$T_{10} = 15℃$			$Q_E = 1$
Absorber			$T_1 = 34.9℃$			$Q_A = 1.27$
Generator			$T_4 = 75℃$			$Q_G = 1.30$
Condenser			$T_8 = 31.5℃$			$Q_C = 1.03$
Coefficient of performance			COP			0.76

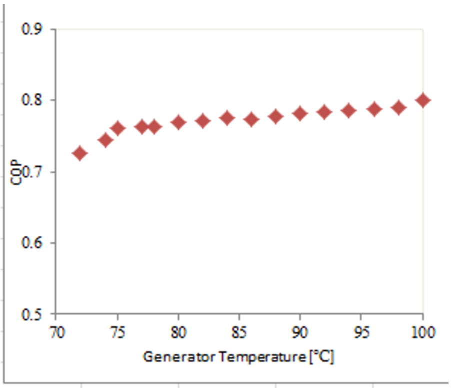

Fig. 3. Relation between COP and the generator temperature

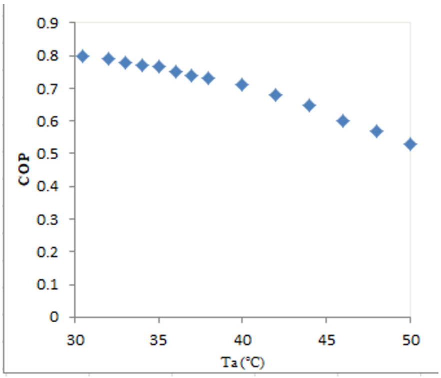

Fig. 4. Relation between COP and the absorber temperature

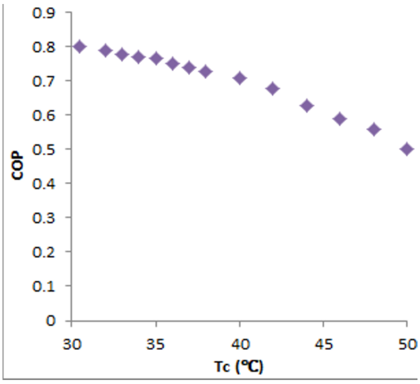

Fig. 5. Relation between COP and condenser temperature

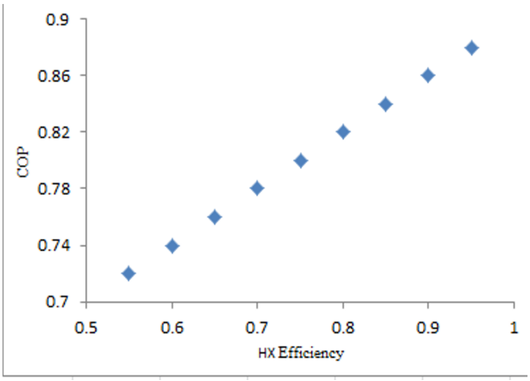

Fig. 6. Relation between COP of system and efficiency of HX

8 Discussions

Fig. 3 depicts the correlation between COP and different values of temperature of generator. It is noticed clearly that, as a rise occurs in the generator temperature, the value of coefficient of performance (COP) also increases. This implies that the system's performance is enhanced as the temperature of the generator rises. With Li-Br, the maximum attainable temperature is 100°C as above that crystallization may take place.

Fig. 4 and fig. 5 shows that the system's COP decreases with a rise in the temperature of the absorber and condenser respectively. So, the system performance & the absorption reaction is improved by reducing the absorber temperature, this statement is explained through the fact that since water absorption by Li-Br is a chemical reaction therefore it requires to be cooled down for achieving an improved performance. To achieve a better condensation rate in the condenser, the water vapour ought to be cooled down which is done using cooling towers or through the phenomena of natural air cooling. Since, water -

libr pair is considered for analysis hence, for cooling purposes water will be a better choice than the other because of the problem of crystallization that may occur [16].

Fig. 6, tells that the COP is increased as heat-exchanger effectiveness increases. The amount of energy required for the overall process is reduced before moving into the generator as the HX aids in increasing the strong solution temperature. An improvement in the COP is noticed, with the increase in the heat-exchanger effectiveness and a reduction in the energy requirement of the generator [16].

9 Conclusions

In this analysis, a mathematical model is articulated by applying the 1st law of thermodynamics to a vapour absorption refrigeration system. This thermally driven solar absorption air conditioning unit is planned and implemented for the province of Delhi, India. The system is integrated with evacuated tube collectors which absorbs sun's radiation and then transforms it into heat output, which warms the water that is passing through the collectors. A well-defined approach is used to find the dimensions of the collector. Area for each of the collector is calculated to be 3.75 m^2. A thermal energy storage unit is incorporated to increase the system's operational efficiency. Numerical modeling of the absorption unit is accomplished with the help of energy calculations for each component of the system, at each of the points from thermodynamic characteristics of the working fluid at the given working parameters. The COP evaluates the capacity of the air conditioner and is calculated to be 0.76. Design for the generator heat exchanger is proposed. The value obtained for the coefficient of heat-transfer for heat exchanger is calculated through the LMTD method and the obtained value lies between the mentioned range.

From the results and discussion using appropriate graphical methods it is discovered that the cycle's COP increases as the generator's working temperature rises, indicating an improvement in efficiency and performance. While the COP value is decreased with a rise in temperatures of absorber and condenser. From the graph, it is observed that an increase in the heat exchanger effectiveness occurs with a decrease in the energy needed in the generator which further leads to an improvement in the performance of the system. Additionally, it has been found that raising the generator temperature improves the vapour absorption cycle's COP but, this increase in temperature negatively impacts the system's efficiency and performance. Thus, a special attention must be given to this dominant reverse effect. Finally, this research leads the way to an improved thermal system. Hence, this methodology can be used and applied to a similar suitable system which will be economical and sustainable.

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