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Performance Evaluation of LiBr-H_2O Vapours Absorption System for an Office Building Cooling

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Authors' contributions

This work was carried out in collaboration among all authors. Author AA designed the study, wrote the protocol and wrote the first draft of the manuscript. Authors AA and RPS managed the literature searches. Authors AA and RSG managed the analyses of the study.

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ABSTRACT

The aim of this paper is to assess the energy performance by simulation of a single effect LiBr-H_2O vapour absorption system for office room cooling under Ludhiana (31°N) climate conditions. In an absorption cooling system, compressor which is using high grade energy, replaced by a combination of generator and absorber. The low grade heat is supplied to the generator which produces the cooling effect in the evaporator section. In the present work, lithium bromide is used as absorbent and water as refrigerant. A thermodynamic simulation of the cycle is carried out to investigate the effects of office room temperature and condenser temperature on the performance of the LiBr-H_2O vapour absorption system. It was concluded that coefficient of performance (COP) increases from 0.55 to 0.75 as office room temperature increases from 22 to 30°C for fixed value of condenser temperature of 46°C. The maximum COP was observed when system runs on least value of condenser temperature.

Keywords: Absorption refrigeration; LiBr-H_2O; simulation; coefficient of performance; room cooling.

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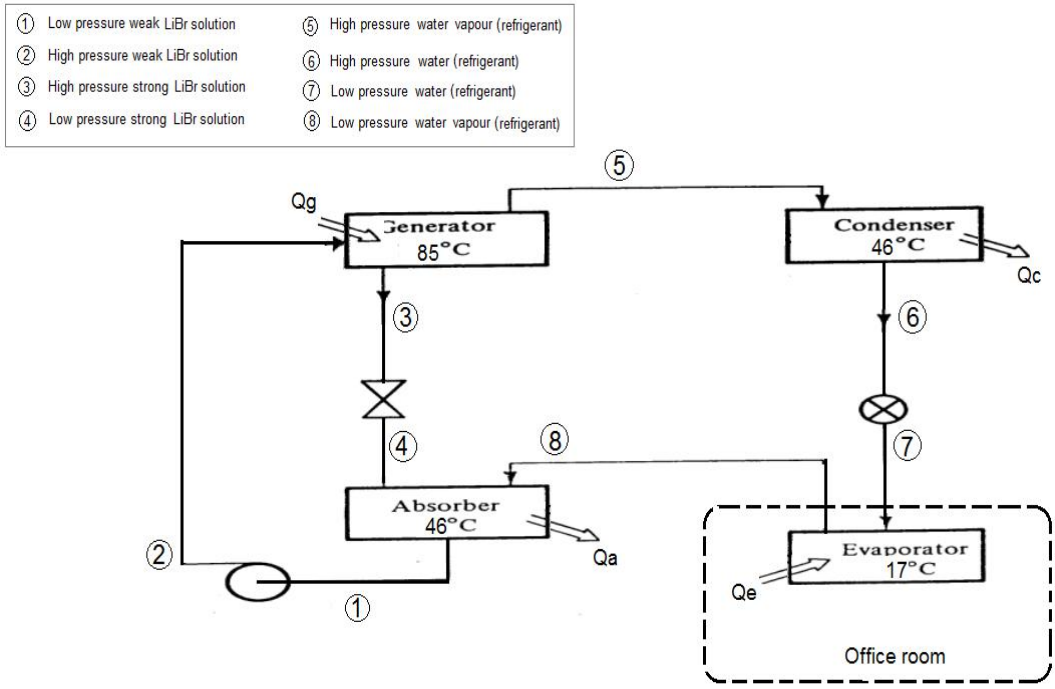
1. INTRODUCTION

Refrigeration and air-conditioning requirement, either for cooling of household, offices, hotels, laboratories or public buildings or preservation and chilling purposes of perishable items, are increasing considerably day by day. The conventional refrigeration and air-conditioning systems, known as the vapor-compression refrigeration process (VCRC), is most widely used today to meet these refrigeration and cooling demands. However, it is well-known that the vapor-compression systems are using ozone depleting refrigerants which is responsible for causing global warming and other environmental issues [1,2]. Research and development of Absorption refrigeration machine reported in many research papers is a suitable option to resolve the above challenges [3]. The vapour absorption refrigeration cycle (VARC) runs on thermal energy with small electrical energy requirement compared with VCRC [4]. The thermal requirement of the VARC system can be fulfilled by low-cost sources such as biomass [5], solar [6,7], exhaust heat from engines [8,9] and waste heat from industrial processes [10]. The principle of working of a VARS is similar to that of a vapor compression system. In an absorption cooling system, low grade heat from any source is supplied in the generator section. The heat evaporates the refrigerant which is further expanded to produce cooling effect. The compressor of the vapor compres-sion system is replaced with generator-absorber combination. The significant feature of the absorption system is to use thermal energy in comparison with the high grade electrical energy. In absorption cooling cycle, Lithium bromide (LiBr) is the most common absorbent with H_2O as the refrigerant and in smaller absorption chillers rarely$H_2O$is used as the absorbent and NH_3 as the refrigerant. However, LiBr has a very high affinity for H_2O and in addition, H_2O of course is relatively inexpensive and non-toxic [11]. Absorption cooling was invented by the French scientist Ferdinand Carr´einin1858 [12]. Solar energy constitutes a popular application of absorption refrigeration which is free, zero emission as well as a clean energy source. Pohl et al. [13], Grossman and Zaltash [14], Sumathy et al. [15], Asdrubali and Grignaffini [16], Hiebler et al. [17], Gogoi and Talukdar [18], Konwar and Gogoi [19] and Kanti Deand Ganguly [20] and many others studied the performances of absorption refrigeration processes using LiBr/H_2O as absorbant-refrigerant. According to the author's knowledge from review on vapour absorption refrigeration system, the very minimum effort has been done on thermodynamic analysis of a vapour absorption air-conditioning system for office room cooling for different room temperature settings as per requirement of different comfort level of different persons. Keeping this in view, this study investigates the performance of LiBr–H_2O based absorption refrigeration system for office room cooling under climate conditions of Ludhiana (31°N) to produce scientific results. The steady state mathematical model was formulated to investigate the effect of different office room temperature settings on coefficient of performance, circulation ratio, thermal load, concentration, system pressure and mass flow rate. In addition, the developed model can easily be extended to include different refrigerant-absorbent pairs for evaluating the thermal performance of single effect absorption cooling system.

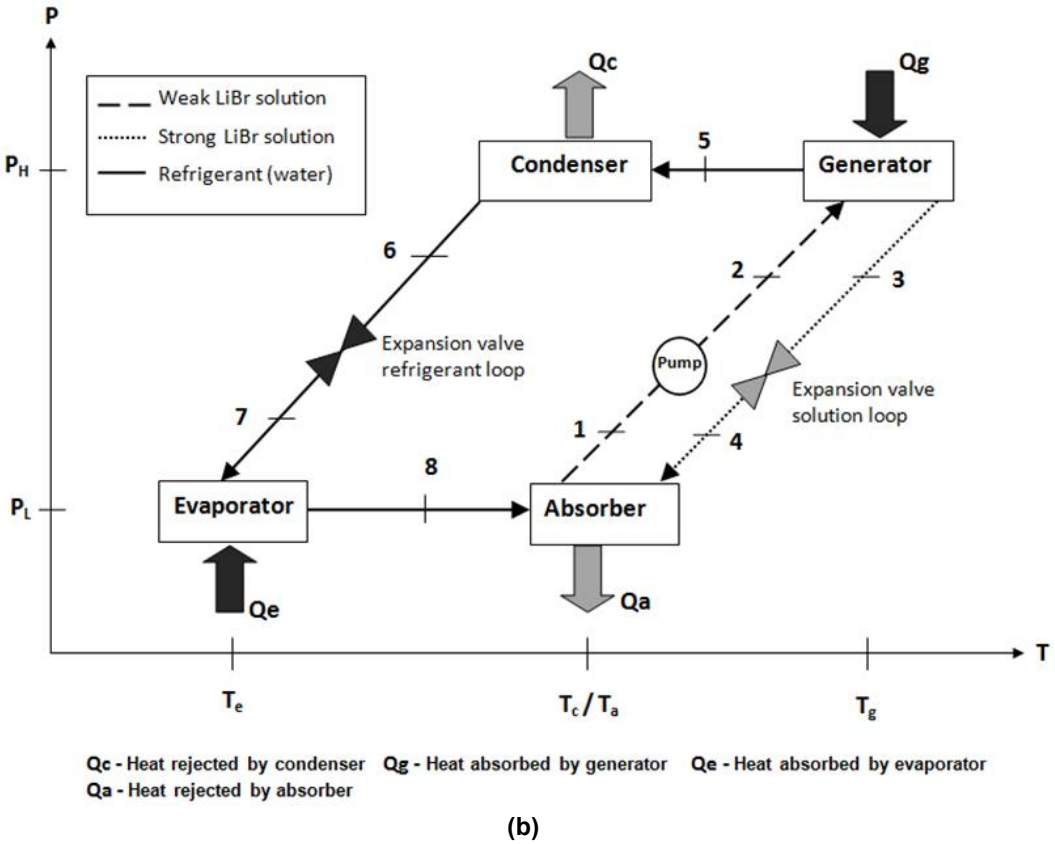
2. MATERIASL AND METHODS

2.1 System Description

In this work, the evaporator of LiBr–H_2O absorption based refrigeration system is kept inside the office room to be cooled as shown in Fig. 1(a). In the LiBr–H_2O absorption refrigeration system, LiBr is used as an absorbent and H_2O as refrigerant. The H_2O (liquid refrigerant) coming from the throttling valve (State 7) evaporates by absorbing the heat (Qe) from the office room (State 8). The refrigerant vapor coming from evaporator is absorbed and condensed into the LiBr–H_2O solution coming from the expansion valve of solution loop i.e. weak in water percentage at state point 4, resulting in weak LiBr–H_2O solution at state point 1. The heat produced (Qa) in absorber is rejected by cooling media. At state 1, the low pressure (P_L) LiBr– H_2O solution is pressurized by the solution pump to high pressure (P_H) solution (State 2) and pump it into generator. In the generator, heat (Qg) is supplied to weak LiBr–H_2O solution at high temperature to extract the refrigerant vapors and rich solution of absorbent returns back to the absorber. The vapor in superheated form condensed and rejected heat (Qc) in the condenser (state 6). Finally, condensed water (state 6) is throttled through expansion valve to low pressure (state 7). Fig. 1 (b) also illustrates the main components of a single-stage absorption refrigeration cycle represented in a pressure-temperature plot.



(a)



(b)

Fig. 1. (a) Schematic (b) P-T diagram of a absorption refrigeration system with different components

2.2 Mathematical Modeling

2.2.1 Model assumptions

In order to conduct the theoretical analysis of vapour absorption refrigeration cycle, the following basic assumptions have been made.

- Steady state processes.
- The outside air is considered as thermal reservoir and its temperature (To) is taken as 40°C (constant) for the month of June at Ludhiana (31°N).
- The refrigerant-absorbent are assumed to be ideal.
- No pressure drop in system except throttling valves.
- Saturated condition at the exit of condenser and evaporator.
- Pump is isentropic and its efficiency is set at a constant value η_p= 50%.
- The generator temperature (*Tg*) is taken as 85°C (constant).

2.2.2 Thermodynamic formulations

The thermal analysis depends on the governing equation of first law of thermodynamics in each component of the cycle. For this, the mass and energy balance equations in steady state processes for the different parts of the refrigerating cycle with respect to Fig. 1 are established as follows [21]:

Evaporator

Heat absorbed in evaporator, $Qe = m_8h_8 – m_7h_7$ (Watts) (1)

Condenser

Heat rejection from condenser, $Qc = m_5h_5 – m_6h_6$ (Watts) (2)

Absorber

Heat rejection from absorber, $Qa = m_8h_8 + m_4h_4 – m_1h_1$ (Watts) (3)

Mass balance, $m_4 + m_8 = m_1$ (Total mass) (4)

$m_4C_4+m_8C_8= m_1C_1$(Mass of LiBr) (5)

Generator

Heat added to generator, $Qg = m_5h_5 + m_3h_3 – m_2h_2$ (Watts) (6)

Mass balance, $m_3 + m_5 = m_2$ (Total mass) (7)

$m_3C_3+m_5C_5= m_2C_2$ (Mass of LiBr) (8)

Efficiency of solution pump

$$\eta_p = \frac{1}{\rho_1}\frac{P_1-P_2}{h_2-h_1} \quad (9)$$

The absorption system uses the phenomenon of chemical absorption and generation of the refrigerant. The heat to the generator can be applied from solar, geothermal, waste heat or natural gas to vaporizes the refrigerant. The refrigerant vapors are cooled in the condenser and further expanded to produce the chilled effect inside the evaporator. Only a small fraction of electrical energy required to run the solution pumps. Hence, the COP of the absorption system is defined as follows:

$$COP = \frac{Q_e}{Q_g} \quad (10)$$

2.3 Calculation Procedure of Vapour Absorption Refrigeration for Office Room Cooling

A computer program is developed in QBasic language for the simulation of vapour absorption refrigeration system. The program is constructed to allow for a sequential computation of all the governing equation of the absorption system given in section 2.2.2.The thermodynamic and transport properties of water, required to solve the mass and energy balances of the mathematical model, are obtained from thermodynamic property correlation developed by Harr et al. [22], valid up to a pressure of 815 bar. Meanwhile, for LiBr–H_2O solution, the following equations obtained by Patek and Klomfar [23], valid for full composition rate and from 273 to 500 K are used.

$$x = \frac{w/M_{LiBr}}{w/M_{LiBr}+(1-w)M_w} \quad (11)$$

$$P = \frac{1}{xM_{LiBr}+(1-x)M_w} \times \left[T - \sum_{i=1}^{8} a_i x^{m_i}(0.4-x)^{n_i}\left(\frac{T}{T_c}\right)^{t_i}\right] \quad (12)$$

$$h = \frac{1}{xM_{LiBr}+(1-x)M_w} \times \left[(1-x)h_f - h_c\sum_{i=1}^{30} a_i x^{m_i}(0.4-x)niTCTC-TOti \right. \quad (13)$$

$$\rho = \frac{1}{xM_{LiBr}+(1-x)M_w} \times \left[(1-x)\rho_f - \sum_{i=1}^{2} a_i x^{m_i}\left(\frac{T}{T_c}\right)^{t_i}\right]. \quad (14)$$

For the thermal analysis of the system, different operating parameter such as of the evaporator

temperature, condenser and absorber temperature are varied and the COP, mass flow rate, circulation ratio, concentration, system pressure and thermal load of the system is plotted. The input value of different operating parameters is given in Table 1. In this table, the five values of room temperature (Tr) are considered as per comfort level for different persons and the temperature in evaporator (Te) is considered 5°C less than room temperature based on efficient fan-coil evaporator [21]. Based on different designs of heat exchanger, the condenser and absorber temperature having 4, 6, 8 and 10°C higher than outside air temperature of 40°C are studied. The condenser and absorber temperature are assumed to be same for each set of operating condition for this study.

2.4 Cooling Load Calculation

The different heating loads that must be rejected from the space to maintain temperature of room air (Tr) at stable level as given in Fig. 2. Cooling load calculations is performed for anoffice room at Ludhiana (31°N). The dimension of this office room is about 10 m (E-W) × 5 m × 4 m high. The walls of room have 250 mm thick brick with 10 mm thick layer of cement on it. The ceiling of room has 0.15 m thick concrete. North and south wall have two windows of 1 m x 1.5 m wide having glass panes 6 mm thickness. In this room, there is one officer, one computer operator, two attendants, one receptionist, four visitors, one 500W fridge, one 300 W photocopier machine, two fans of 100W each, six tube rods of 40W each. As the temperature difference between the room air and the ambient is the major parameter for the cooling load calculation therefore, for this study, the maximum outdoor air temperature (To) of 40°C for the month of June at Ludhiana (°N) and inside room air temperature (Tr) of 22, 24, 26, 28 and 30°C (Table 1) are considered. The sample calculation of different heat loads on office room for outdoor air temperature of 40°C and inside room air temperature of 22°C, for cooling load calculation, is given in Appendix [24]. The cooling load of the room for different inside room air temperature of 30, 28, 26, 24 and 22°C comes out to be as 17, 20, 23, 26 and 29 kW respectively.

Table 1. The input data of different parameters

Sr. No.	Parameter	Symbol	Temperature (°C)
1.	Room / Evaporator temperature	Tr/Te	22/17, 24/19, 26/21, 28/23, 30/25
2.	Absorber and condenser temperature	Tc = Ta	44, 46, 48, 50

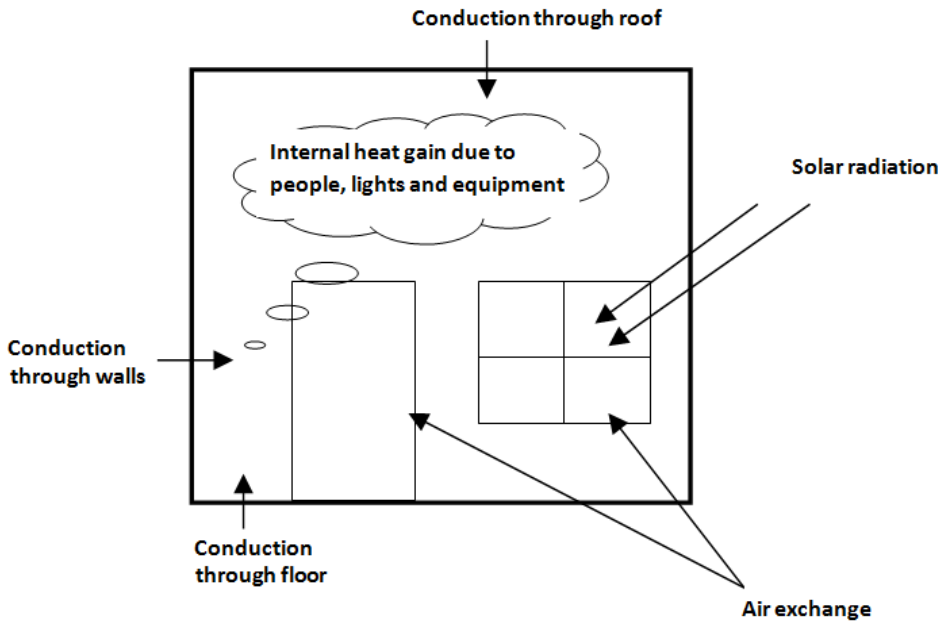


Fig. 2. Different heat loads on office room for cooling load calculation

3. RESULTS AND DISCUSSION

The results of parametric study of LiBr–H_2O based absorption refrigeration cycles to provide air conditioning for summer is presented in this section. Table 2 shows the thermodynamic properties at various state points and energy flows to and from each component of vapour refrigeration system in reference to one set of operating conditions as shown below.

Input data

Room air temperature, Tr = 22°C
evaporator temperature, Te = 17,
cooling load, Qe = 29 kW,
absorber temperature, Ta = Tc = 46°C and
outside air temperature, To = 40°C.

The variation of COP increases from 0.55 to 0.75 (Fig. 3) as office room temperature is increased from 22 to 30°C (i.e. increase in evaporator temperature from 17 to 25°C), for fixed value of absorber and Condenser temperature of 46°C. This is because of decrease in CR value 19.79 to 7.0 as can be seen in Fig. 4 and increase in Te from 17 to 25°C. Manu and Chandrashekar [25] also showed the increase in COP as CR decreases. Similar trend of increase in COP with increase in office room temperature can be observed for other values of absorber and Condenser temperatures. The best COP value of 0.78 is obtained with least value of absorber temperature (Ta) and condenser temperature (Tc) considered in this study.

Fig. 5 presents the heat load variations on the office room temperature for fixed value of absorber and Condenser temperature of 46°C and generator temperature of 85°C. It can be seen from Fig. 5 that when office room temperature increases from 22 to 30°C (i.e. increase in evaporator temperature from 17 to 25°C), there is a decrease in generator load (Qg) from 52.36 to 22.7 kW, absorber load (Qa) from 50.8 to 21.89 kW, condenser load (Qc) from 30.57 to 17.81 kW and pump work (Wp) from 2.5 to 0.47 W (Fig 7). This decrease in thermal loads i.e. Qg, Qa and Qc and pump work (Wp) is due to decreases in CR [25]. Aphornratana and Sriveerakul [26] also verified experimentally that, Qg, Qa and Qc decrease as circulation ratio (CR) decreases. It can be also seen from Fig. 5 that cooling load (Qe) on evaporative decreases with the increase in office room temperature which is due to decrease in the temperature difference of room and outside air (ΔT).

Fig. 6 demonstrates variations of the pressure as a function of office room temperature for constant absorber and Condenser temperature of 46°C and constant generator temperature of 85°C. As office room temperature increases from 22 to 30°C, there is an increase in low pressure side of the system, i.e. evaporator and absorber pressure, from 0.019 to 0.032 bar. This is well known fact that as saturation temperature increases saturation pressure also increases. However, there is no effect on high pressure side of the system, i.e. pressure in condenser and generator pressure remains constant at 0.101 bar. This is due to constant condenser temperature of 46°C. Manu and Chandrashekar [25] showed the similar findings.

Table 2. The output data corresponding to one set of operating conditions

State point	Hi (kJ/kg)	Mi (g/s)	Pi (bar)	Ti (°C)	Wi (%)
1	103.2	245.3	0.019	46	53.9
2	103.2	245.3	0.101	46.7	53.9
3	192	232.9	0.101	85	56.7
4	192	232.9	0.101	73.3	56.7
5	2658	12.4	0.101	85	0.00
6	192.6	12.4	0.101	46	0.00
7	192.6	12.4	0.019	17	0.00
8	2532	12.4	0.019	17	0.00
Coefficient of performance,		COP	=	0.55 (-)	
Circulation ratio,		CR	=	19.79 (-)	
Heat rejected by absorber,		Qa	=	50.8 KW	
Heat rejected by condenser,		Qc	=	30.57 kW	
Heat absorbed by evaporator,		Qe	=	29.0 kW	
Heat absorbed by generator,		Qg	=	52.36 kW	
Solution pump work,		Wp	=	2.518 W	

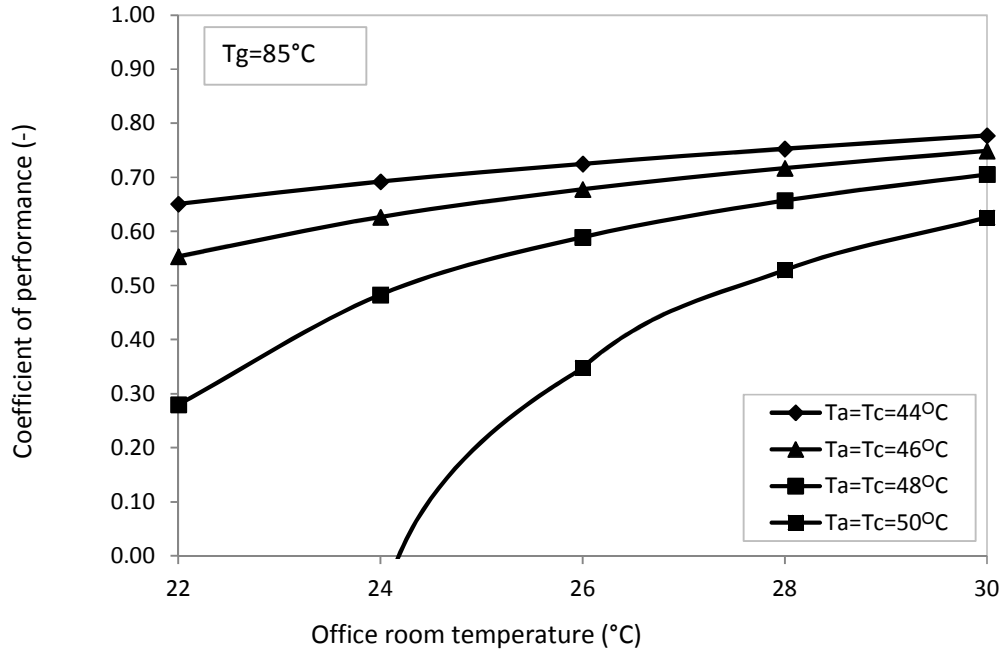


Fig. 3. Variation of coefficient of performance with the office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature

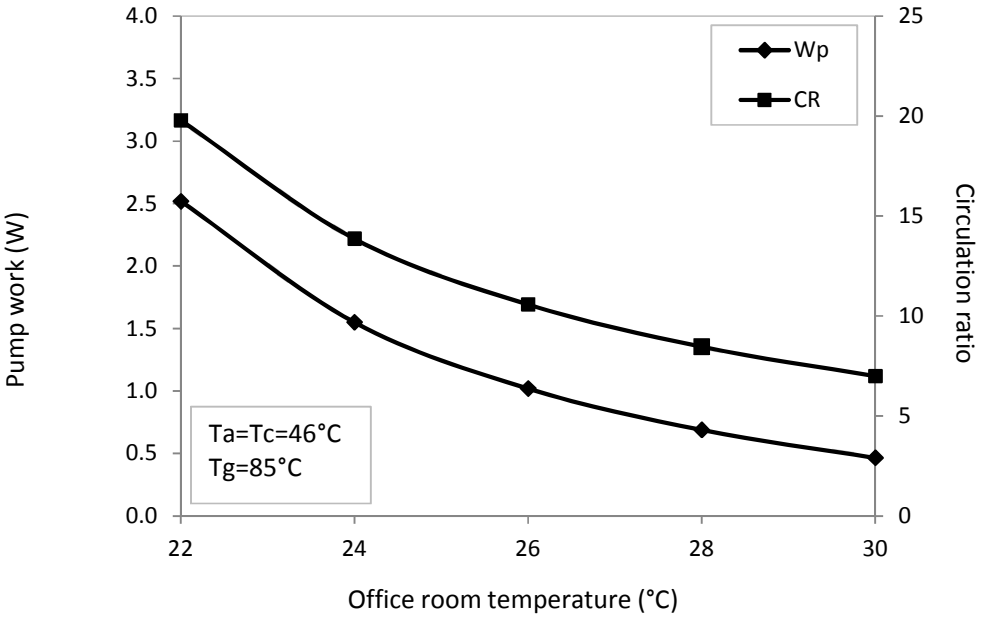


Fig. 4. Variation of concentration ratio and solution pump work with the office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature, Wp – Solution pump work, CR – Circulation ratio

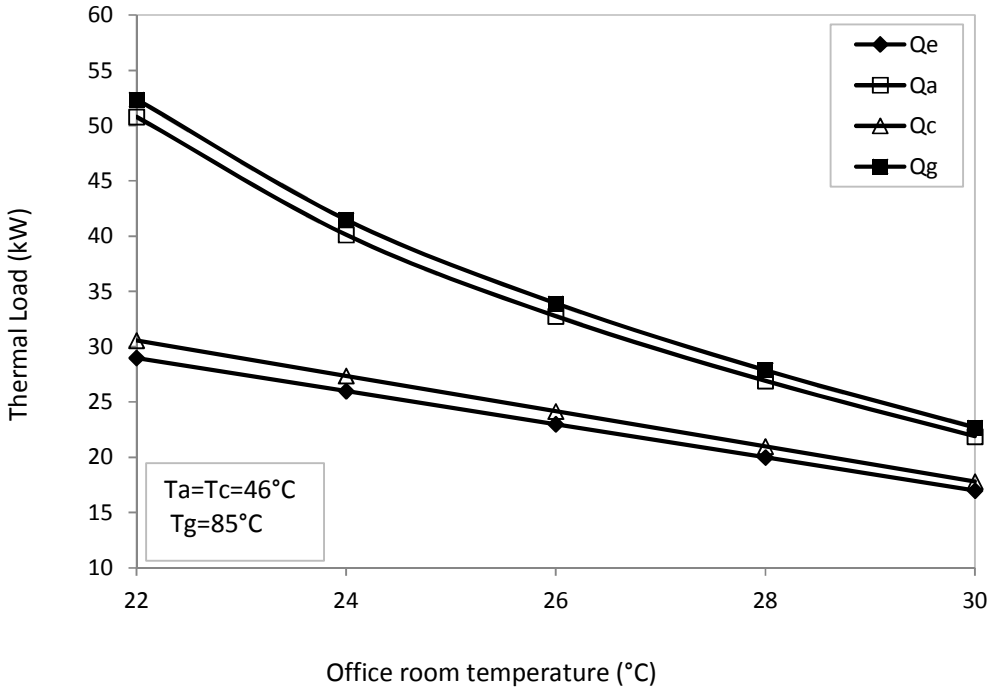


Fig. 5. Variation of thermal load with the office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature, Qe – Heat absorber in evaporator, Qa – Heat rejected in absorber, Qc – Heat rejected in condenser, Qg – Heat absorbed in generator

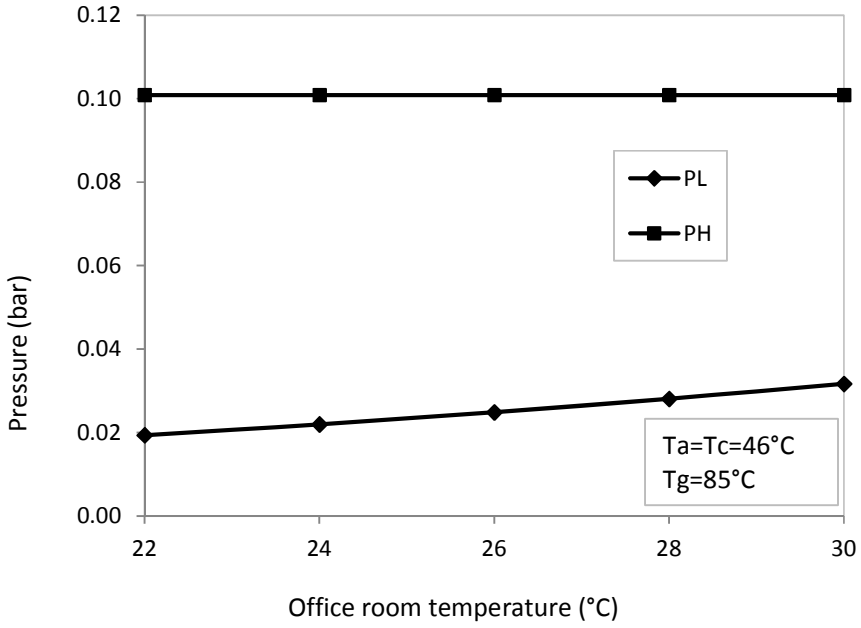


Fig. 6. Variation of pressure with the office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature, P_L – Low pressure, P_H – High pressure

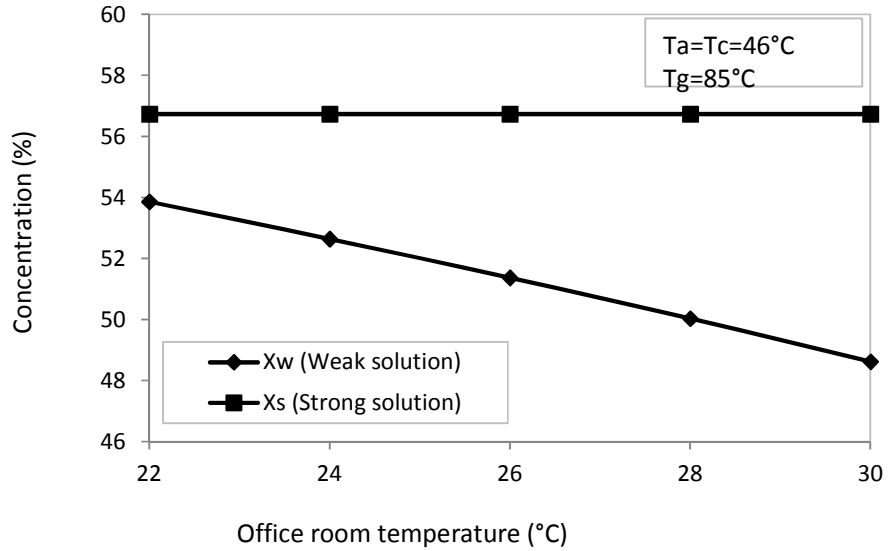


Fig. 7. Variation of concentration with the office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature

Fig. 7 shows the concentration as a function of office room temperature for fixed absorber temperature and condenser temperature of 46°C and fixed generator temperature of 85°C. The decrease in concentration (53.86 to 48.62%) of weak solution (w_1) leaving absorberwith increase in office room temperature is observed. This decrease in concentration is due to an increase in weak solution pressure (P_L) with the increase in office room temperature (Fig. 6). However, the concentration of strong solution coming out of the generator remain constant (w_1= 56.73%), this is due to a constant pressure of strong solution (P_H) coming out of the generator for all values of office temperatures (Fig. 6). The results of Fig. 7 are in accordance with the results presented in the references [25,27].

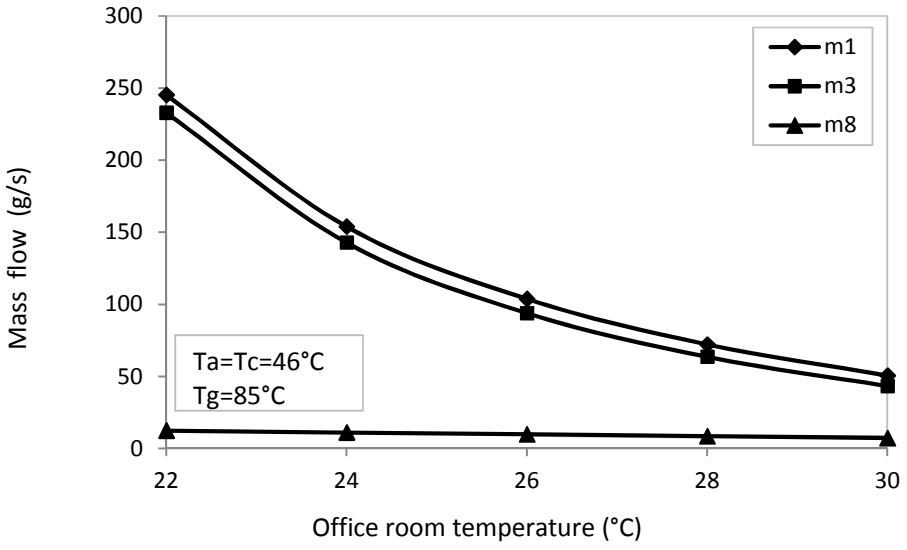


Fig. 8. Variation of mass flow with office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature, m_1 - mass flow rate of weak solution, m_3 - mass flow rate of strong solution, m_8 - mass flow rate of refrigerant in evaporator

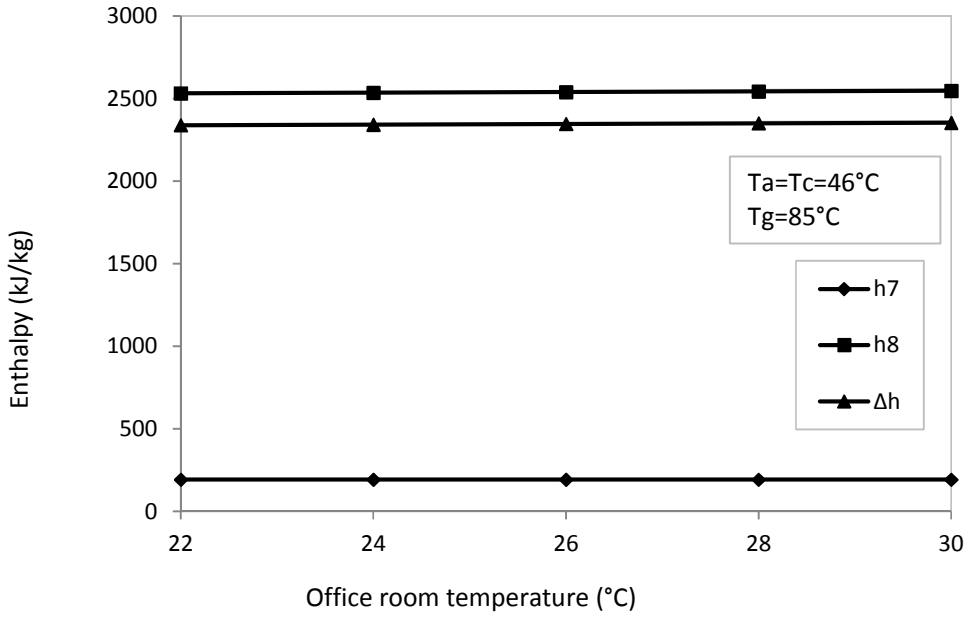


Fig. 9. Variation of enthalpy with office room temperature

Ta – Absorber temperature, Tc – Condenser temperature, Tg – Generator temperature, $\Delta h = (h_8-h_7)$ – enthalpy of evaporation

Fig. 8 shows of mass flow rate variations with room temperature at constant absorber and condenser temperature of 46°C and constant generator temperature of 85°C. As office room temperature increases, the decrease in weak solution mass flow rates (m_1) from 245.3 to 50.51 g/s, decrease in strong solution mass flow rates (m_3) from 232.9 to 43.29 g/sis observed which is because of decrease in CR (Fig. 4). It is also observed decrease in refrigerant mass flow rate (m_8) in evaporator from 12.4 to 7.223 g/s. This is because with an increase in the Te from 22 to 30°C, the cooling load on evaporator (Qe) decreases (Fig. 5) and enthalpy of evaporation ($\Delta h = h_8 - h_7$) in evaporator increases (Fig. 9), resulting in a decrease in mass flow rate (m_8) through evaporator as given by equation (3). The results of Fig. 8 are in line with the other studies [25,28].

4. CONCLUSION

In this study, a simulation of the single-stage LiBr–H_2O vapor absorption refrigeration system (VARS) for an office room cooling was done and the system performances were analyzed. This VARS is alternative to conventional VCRS. In this system, LiBr has been opted as absorbent and H_2O as refrigerantfor cooling purpose. The results revealed that the COP rises from 0.5538 to 0.7489 with 22 C to 30°C increase in room temperature for fixed value of absorber and condenser temperature of 46°C. The best COP is attained with least value of condenser and absorber temperature. Further, the CR value of the system showed its significance in deciding the performance of the system. Using computer program developed in QBasic language for thermodynamic analysis of LiBr–H_2O vapor absorption refrigeration system, the effect of office room conditions on coefficient of performance, circulation ratio, thermal load, concentration, system pressure and mass flow rate can be evaluated with higher degree of accuracy and faster rate

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COMPETING INTERESTS

Authors have declared that no competing interests exist.

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APPENDIX

The heat load calculations for outdoor air temperature of 40°C and inside room air temperature of 22°C, i.e. air temperature difference between outside and room air of 18°C, are as under:

a) Heat load due to conduction through roof, floor and walls

Heat gain through ceilings, floors, walls, and doors is due to temperature difference of air across them. The heat gain also depends upon the overall heat transfer coefficient of different components of room which further depend on thermal conductivity of material and heat transfer coefficient of outside and inside air of office room. The thermal conductivity of brick, cement, concrete, glass, wood is taken as 2.76, 1.56, 5.57, 3, 1.3 kJ/h-m-K respectively and the heat transfer coefficient of outside and inside air as 270 kJ/h-m^2-K and 150 kJ/h-m^2-K respectively. The overall heat transfer coefficient of different components of room is calculate as under

$U_{wall} = 1/(1/h_o+t_{brick}/k_{brick}+t_{cement}/k_{cement}+1/h_i)$
$= 1/(1/270 + 0.25/2.76 + 0.01/1.56 + 1/150) = 9.314$ kJ/h-m^2-K

$U_{door} = 1/(1/270 + 0.04/1.3 + 1/150) = 24.31$ kJ/h-m^2-K

$U_{glass} = 1/(1/270 + 0.006/3 + 1/150) = 80.838$ kJ/h-m^2-K

$U_{ceiling} = 1/(1/270 + 0.15/5.57 + 1/150) = 26.81$ kJ/h-m^2-K

Heat gain through different parts of the building can be estimated by using following

$Q_{North} = [(Ax\ U)_{wall} + (Ax\ U)_{door} + (Ax\ U)_{window}] (T_{o\ (max)} - T_r)$
$= [(10x4 - 2x1 - 2x1x1.5)\ 9.314 + (2x1)\ 24.31 + (2x1x1.5)\ 80.836] \times 18$
$= 11104.8$ kJ/h

$Q_{south} = [(Ax\ U)_{wall} + (Ax\ U)_{window}] (T_{o\ (max)} - T_r)$
$= [(10x4 - 2x1x1.5)\ 9.314 + (2x1x1.5)\ 80.836] \times 18 = 10568.8$ kJ/h

$Q_{East\text{-}West} = (Ax\ U)_{wall}\ (T_{o\ (max)} - T_r)$
$= [2(5x4)\ 9.314\ ] \times 18 = 6705.7$ kJ/h

$Q_{ceiling} = (Ax\ U)_{ceiling}\ (T_{o\ (max)} - T_r)$
$= [(10x5)\ 26.81\ ] \times 18 = 24124.2$ kJ/h

The total heat gain due to structure is therefore,

$Q_{structure} = 11104.8+10568.8+6705.7+24124.2 = 52{,}506$ kJ/h

b) Heat load due to solar radiation

Considering solar heat gain load as 36% total head gain load due to structure, the heat gain due to structure including solar radiation is given by

$Q_{(Structur+Solar\ rad.)} = 0.36 * 52{,}506 = 18{,}902.16$ kJ/h

c) Heat load due to air exchange

Both sensible and latent heat is considered for the infiltration as entry of outside air into the conditioning area influences both the humidity and temperature level of air. The number of air changes/24 hours is assumed to be 4.99 for office volume of 10 x 5 x 4 = 200m^3.
Therefore infiltration air = 4.99 x 200/24 = 41.6 m^3/h
Ventilation required for 8 persons in case of private office is: 8 x 12x10^{-3} x 3600 = 345.6 m^3/h

Therefore heat load based on total air exchange: (41.6+345.6) x ρ_{air} x c_p x (T_o-T_i) = 387.2 x (1.013/2872 x 298) x 1.004 (40-22) = 14599.4 kJ/h

d) Heat load due to internal heat gain

The primary sources of internal heat gain are equipments, occupants and lights operating within the cooling space.

(i) Heat load due to occupants = (2 clerk x 500) + (2 attendants x 600) + (4 persons x 450) + (1 receptionist x 600) = 4600 kJ/h
(ii) Heat load due to appliances= [(2 fans x 100) +(6 tubes x 40) + (1 photocopy machine x 300) + (1 refrigerator x 500)] x 3600/1000 = 4464 kJ/h

Total heat load due to internal heat gain = Occupants Load + Appliances load = 4600+ 4464 = 9064 kJ/h

The total Heat load = 52,506 + 18,902.16 +14599.4 + 9064 = 95071.5 kJ/h

Taking 10% safety factor, the total heat load on air conditioning apparatus is found to be: 95071.5 kJ/h x 1.10 = 104578.7 kJ/h or 29 kW

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