Al-Rafidain Engineering Journal (AREJ) Vol.29, No.1, March 2024, pp. 74-84

Thermodynamic Design of 10 TR Single-Effect LiBr-H_2O Absorption Refrigeration System

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Received: August 1st, 2023 Received in revised form: October 18th, 2023 Accepted: November 6th, 2023

ABSTRACT

The use of energy, which is perceived as a key element in the development of civilization and is necessary for all aspects of modern life, is one of the measures of the country's growth. Due to the high costs of conventional energy sources as well as the possibility of their depletion, it has become necessary to look for new and direct sources of energy. As a result, thinking began to return to absorption cooling systems due to their ability to work using direct thermal energy. Apart from their simplicity and lack of moving parts, absorption cooling systems are commonly used because they can operate by a heat source with relatively low temperatures, such as those generated by burning natural gas or by solar collectors, to produce a refrigeration effect directly by evaporating the refrigerant. This article presented a thermodynamic design of (10 TR) single-effect (LiBr/ H_2O) absorption refrigeration system. The capacity of the evaporator (35.17 KW) was used to determine the operating parameters for each component. Thermodynamic simulations are carried out on the basis of experimental correlations. To determine the various operational parameters of a vapor absorption refrigeration system under various operating conditions, a MATLAB code was developed. The effectiveness of the solution heat exchanger and the various temperatures of the generator, condenser, evaporator, absorber, are taken into consideration when calculating the coefficient of performance. The results obtained prove that the coefficient of performance increases by increasing the effectiveness of solution heat exchanger, generator and evaporator temperatures, and decreases by increasing absorber and condenser temperatures.

Keywords:

LiBr-H_2O; Absorption refrigeration systems; Air conditioning; Thermodynamic design.

1. INTRODUCTION

Due to the high internal loads in buildings and the growing demand for thermal comfort by its users, air conditioning has become one of the most important forms of energy consumption in countries with a hot climate [1]. Accordingly, during the hot summer months, the electricity supply network begins to have problems due to the increased demand for electrical energy for refrigeration equipment.

Electrically driven vapor compression chillers are installed in most buildings. In 2050, the energy consumed for air conditioning is expected to double tenfold [2]. In Iraq, total energy demand (48% in the residential sector) was exceeded by demand for refrigeration and air conditioning by 50 to 60 % [3]. This leads to an increase in carbon dioxide emissions, which can increase by 60% by 2030 compared to the beginning of the century, despite the urgent need to reduce them [4]. Mechanical compression chillers, on the other hand, use various types of halogenated organic refrigerants, such as hydrochlorofluorocarbons, which still contribute to ozone depletion; this is the reason why many of these refrigerants are banned or in the process of being banned. Solar-powered cooling systems, which achieve energy savings and reduce greenhouse gas emissions by more than 50% [5], seem to be an attractive alternative to conventional electrical driven compression units to enhance the energy efficiency of buildings. Where refrigerants are used with low electrical energy requirements and do not cause damage to the ozone layer.

The absorption refrigeration system, is one of the most important types of thermal driven refrigeration systems and is one of the oldest known refrigeration systems. When the world began to realize the real dangers associated with the depletion of conventional energy sources in recent years, it began to think about returning to the use of absorption refrigeration systems in order to be powered by direct thermal energy, which can be obtained in whole or in part from solar energy, the use of which has seen significant progress. In addition to the possibility of their direct operation, simplicity and lack of moving parts, absorption refrigeration systems are useful because they can work with a heat source with relatively low temperatures, such as waste heat or solar heat [6].

The absorption system is one of the most promising refrigeration system alternatives that can be considered for cooling applications in the current days, because the energy required to drive the refrigeration system is provided by a heat source, such as solar energy or burning natural gas. The role of the input heat is to separate the refrigerant from the refrigerant – absorbent solution in the generator and increase its pressure to condenser pressure, without the need for an electric compressor.

The refrigerant is absorbed and transported by the absorbent in vapors absorption refrigeration systems. Analyzes were carried out using a variety of absorbents and refrigerants. LiBr-H_2O and NH_3-H_2O are the two best and most appropriate combinations of absorbents and refrigerants. In the LiBr-H_2O pair, water acts as a refrigerant, whereas ammonia acts as a refrigerant in the NH_3-H_2O pair [7]. Thermodynamically, in a vapor absorption refrigeration system, these pairs have very high performance.

A. Bell et al. (1996) [8] created an experimental LiBr-H_2O absorption cooling system that was powered by solar energy. To observe all the activities, each component of the vapor absorption system was placed in evacuated glass cylinders. They found that the thermodynamic performance of the system involves an energy balance of each component. They also found that the cycle performance depends on the temperature of the generator, and that there is an ideal temperature for the generator when the COP is at its maximum. They reached the conclusion that when the system is run with low condenser and absorber temperatures, a reliable coefficient of performance can be obtained at a low generator temperature of up to 68 degrees Celsius.

S.F. Lee and S. A. Sheriff (2000) [9] performed a thermodynamic analysis of LiBr-H_2O absorption system for cooling and heating applications, they evaluated the effect of the temperature of the heat source only on both COP and exergetic efficiency. As expected, the lower cooling water temperature leads to a greater performance cooling coefficient and higher energy efficiency, according to the results of the parametric analysis of the absorption cooling system under various operating situations. Raising the heat source's temperature can raise the system's coefficient of performance, but doing so has the drawback of causing the system's coefficient of performance to drop if the temperature goes above a particular point.

M.B. Arun et al (2001) [10] compared the performance of LiBr-H_2O absorption systems with parallel flow and series flow with a double effect. They conducted their investigation using the idea of equilibrium temperature at low generator pressure. Comparisons between the series flow cycle's coefficient of performance and sensitivity to operating conditions were made. They concluded that the coefficient of performance of a parallel flow system is considered to be more sensitive to changes in the temperature of the evaporator than the condenser and the absorber. And the parallel flow system is more affected by the external heat input in the LP generator than the series flow system.

O. Kaynakli et al (2007) [11] conducted a theoretical study of how operational conditions affect the performance of the absorption refrigeration system. Then they analyzed the performance of single stage LiBr-H_2O absorption refrigeration system under different operating conditions using the first and Second Laws of thermodynamics. Investigations were conducted on how component thermal loads, performance coefficients, and efficiency ratios were affected by operating temperature and heat exchanger effectiveness. They concluded that with an increase in the temperature of the generator and the evaporator, the thermal loads on the absorber and the generator decrease, and as the generator's thermal load decreases, the performance coefficient rises.

Francis A. G. et al (2010) [12] designed and experimentally tested the performance of an outdoor LiBr-H_2O solar thermal absorption cooling system with a cold store. A prototype of a domestic experimental solar cooling system was developed and tested. The performance of the system as a whole, as well as the performance of the various system components, was assessed using physical measurements of daily solar radiation, ambient temperature, inlet and outlet fluid temperatures, mass flow rates, and electrical usage by component. Chilled water temperatures of up to 7.4 degrees Celsius in the results of the experiments demonstrated the new concept of the possible use of a cold store in cooling buildings of a domestic scale, which indicates its viability on this scale.

A.A.V. Ochoa et al (2016) [13] used LiBr-H_2O pair in the dynamic study of a single-effect absorption chiller. A mathematical model that takes into consideration the relationships of the convective coefficients of the absorption refrigeration process has been developed based on the concepts of mass, energy, and species conservation. The MATLAB platform was used to create this mathematical model. The model can simulate and anticipate how internal and exterior factors such as temperature, concentration, and pressure will react when power supply and thermal load are disrupted. According to the model, it is possible to understand and derive more information about its behavior, such as how an increase in the temperature of hot water in the chiller does not necessarily result in an increase in the COP of the system.

2. SYSTEM DESCRIPTION

The absorption cycle is distinct from the vapor compression refrigeration cycle because it is initiated by heat rather than work. As a result, because the absorption cycle performance coefficient is not determined in the same way as the regular vapor compression cycle, the two performance coefficients shouldn't be immediately compared. Some effort is required in absorption systems that employ a generator feed pump, but it is insignificant when compared to the compressor effort required in a normal vapor compression system.

The compressor is necessary to compress the refrigerant in the vapor compression cycle. In vapor absorption cycles, the conventional compressor is replaced by an absorber, a solution heat exchanger, a pump and a generator.

The absorption cycle of the absorbent-refrigerant pair is shown in Fig. 1. Heat is released when the absorbent in the absorber absorbs the refrigerant in the vapor phase. Then, through a solution heat exchanger, the strong solution of the absorbent and the refrigerant is pumped to the generator. In the generator, heat from the heat source separates the refrigerant from the solution.

After the refrigerant is separated from the solution, a weak solution (low percentage of refrigerant) is left in the generator. Through the expansion valve, weak solution that has just passed through the solution heat exchanger is sent to the absorber. In the solution heat exchanger, heat is transported from a weak solution to a strong solution (solution of the absorbent-refrigerant). As a result, the solution heat exchanger contributes to an increase in the system's coefficient of performance. Since it consumes much less pumping energy than compressors, pumps used in vapor absorption refrigeration systems are often neglected in the analysis. Solar energy, natural gas, and waste heat can all be used to power a vapor absorption refrigeration system. The condenser, evaporator, and throttling valve work in the same way as traditional vapor compression refrigeration systems [14].

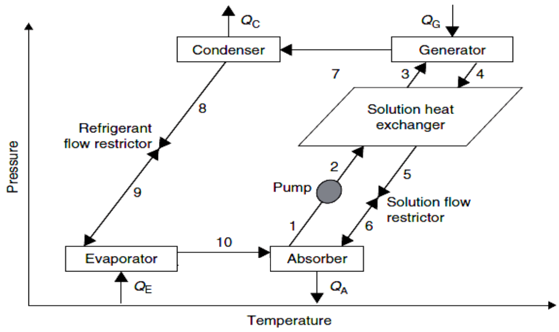

Fig. 1: Single-Effect LiBr-H_2O Vapor Absorption Cycle (ASHRAE).

LiBr-H_2O absorption refrigeration system has a better COP than NH_3-H_2O systems, where it ranges from 0.6 to 0.8 [15]. It operates at generator temperatures between 70 and 95 ℃ [16]. Water is used as a refrigerant in the absorber and condenser. Because the refrigerant in LiBr-H_2O systems is water vapor, the evaporator is unable to work at temperatures below 5 degrees Celsius, which is regarded as one of the system's key shortcomings.

In a LiBr-H_2O absorption system, water works as a refrigerant, pulling heat from the environment and dissipating it, while lithium bromide works as an absorbent, converting water vapor into a solution that can be circulated using a solution pump.

There are many air conditioning applications that use LiBr-H_2O absorption system. Applications requiring temperatures above 32°F can benefit from this system [17]. There are two types of LiBr-H_2O absorption refrigeration system available: single- effect and double-effect. The single-effect absorption chiller is mostly utilized for building cooling loads when water that is chilled at 6-7 ℃ is needed. The coefficient of performance (COP) changes slightly depending on the heat source and the temperature of the cooling water. When water is pressurized, single-effect chillers can operate at hot water temperatures of approximately 70 to 120 ℃ [18].

3. THERMODYNAMIC ANALYSIS OF THE SYSTEM

The Continuity equation is used to model the mass balance. Eq. (1) illustrates the mass balance in any state of the system. The sum of the masses exiting from each point in the cycle must equal the sum of the masses entering at that point.

$$\sum_{in} \dot{m_{in}} = \sum_{out} \dot{m_{out}} \quad (1)$$

The Continuity equation and the first law of thermodynamics, respectively, were used to model the energy balance. The energy added or subtracted at a particular point is equal to the difference between the energy delivered to the point and the energy coming out of the point, as shown by Eq. (2).

$$Q = \sum_{out} \dot{m_{out}}\, h_{out} - \sum_{in} \dot{m_{in}}\, h_{in} \quad (2)$$

Using the above equations, the following equations are explained for the analysis of the thermodynamics of the system:

3.1 GENERATOR

The solution pumped out of the absorber is the mass entering the generator ($\dot{m}_3$), and the mass leaving represents the quantity of steam produced ($\dot{m}_7$) as well as the solution going toward the heat exchanger ($\dot{m}_4$). Eq. (3) shows the mass balance.

$$\dot{m}_3 = \dot{m}_4 + \dot{m}_7 \quad (3)$$

The difference between the energy supplied by the solution ($\dot{m}_3 h_3$), which is transferred by a pump from the absorber to the generator, and the energy needed to generate vapor ($\dot{m}_7 h_7$), and the energy that the solution absorbed. ($\dot{m}_4 h_4$) and sent to the heat exchanger, is the energy required by the generator. Eq. (4) shows the energy balance.

$$Q_G = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3 \quad (4)$$

3.2 CONDENSER

The steam coming from the generator ($\dot{m}_7$) and the condensate ($\dot{m}_8$) are the two masses that enter and exit the condenser, respectively. Eq. (5) shows the mass balance in the condenser.

$$\dot{m}_7 = \dot{m}_8 \quad (5)$$

The difference between the steam energy ($\dot{m}_7 h_7$) supplied by the generator and the condensate ($\dot{m}_8 h_8$) coming out of the condenser is the energy that was taken out of the condenser. Eq. (6) shows the energy balance of the condenser.

$$Q_C = \dot{m}_7 (h_7 - h_8) \quad (6)$$

3.3 EVAPORATOR

After passing through the expansion device, the condensate from the condenser ($\dot{m}_9$) enters the evaporator, and after receiving energy from the fluid that needs to be cooled, that mass exits as an evaporated condensate ($\dot{m_{10}}$). Eq. (7) shows the mass balance in the evaporator.

$$\dot{m}_9 = \dot{m_{10}} \quad (7)$$

The energy difference between the energy supplied by the condensate ($\dot{m}_9 h_9$) and the energy necessary to evaporate it ($\dot{m_{10}} h_{10}$), is the energy absorbed by the coolant in the evaporator. Eq. (8) shows the energy balance of the evaporator.

$$Q_E = \dot{m}_9 (h_{10} - h_9) \quad (8)$$

3.4 ABSORBER

The solution, which is sending from the heat exchanger ($\dot{m}_6$) and the evaporated condensate ($\dot{m_{10}}$) from the evaporator, make up the bulk entering the absorber. While the strong solution that is pumped into the generator as a mass ($\dot{m}_1$) comes out of the absorber. Eq. (9) shows the mass balance in the absorber.

$$\dot{m}_1 = \dot{m}_6 + \dot{m_{10}} \quad (9)$$

Energy, that is collected from absorber, is difference between the energy which is supplied by the strong solution ($\dot{m}_1 h_1$) pumped to the generator and the energy supplied by the solution ($\dot{m}_6 h_6$) transferred from the heat exchanger and the evaporated condensate ($\dot{m_{10}} h_{10}$) from the evaporator. Eq. (10) shows the energy balance of the absorber.

$$Q_A = \dot{m}_6 h_6 + \dot{m_{10}} h_{10} - \dot{m}_1 h_1 \quad (10)$$

3.5 SOLUTION PUMP

The pump's mass balance, and work required to pumping the solution to the generator are shown in Eqs. (11) and (12).

$$\dot{m}_1 = \dot{m}_2 \quad (11)$$

$$W_p = \dot{m}_1 (h_2 - h_1) = \frac{\dot{m}_1 v_1 (P_2 - P_1)}{\eta_p} \quad (12)$$

3.6 SOLUTION HEAT EXCHANGER

Eq. (13), illustrating the technology of the effectiveness of the heat exchanger, is used to calculate the different temperatures in the solution heat exchanger. According to Eq. (14), energy absorbed by cold fluid, supplied from the absorber must be equal to energy which is supplied by hot fluid, from the generator at solution heat exchanger.

$$\varepsilon_{shx} = \frac{T_4 - T_5}{T_4 - T_2} \quad (13)$$

$$\dot{m}_3 h_3 + \dot{m}_5 h_5 = \dot{m}_2 h_2 + \dot{m}_4 h_4 \quad (14)$$

3.7 SOLUTION FLOW RESTRICTOR

According to Eqs. (15) and (16), mass flow rate and energy of solution remain unchanged at solution flow restrictor.

$$\dot{m}_5 = \dot{m}_6 \tag{15}$$

$$\dot{m}_5 h_5 = \dot{m}_6 h_6 \tag{16}$$

3.8 REFRIGERANT FLOW RESTRICTOR (EXPANSION VALVE)

According to Eqs. (17) and (18), there is no change in mass flow rate or energy of solution at expansion valve. By maintaining a constant enthalpy, refrigerant is cooled at expansion valve.

$$\dot{m}_8 = \dot{m}_9 \tag{17}$$

$$\dot{m}_8 h_8 = \dot{m}_9 h_9 \tag{18}$$

4. PERFORMANCE CALCULATIONS

Many assumptions and considerations are taken into account during the thermodynamic design of the vapor absorption refrigeration system. The design considerations are as follows:

- Refrigeration load: 10 TR
- Evaporator Temperature T_E: 5 to 12 ºC
- Condenser Temperature T_C: 30 to 45ºC
- Absorber Temperature T_A: 30 to 45ºC
- Generator Temperature T_G: 70 to 95ºC
- Effectiveness of SHE $\boldsymbol{\varepsilon_{shx}}$: 0.6 to 0.8
- Refrigeration Solution: LiBr-H_2O

This set of values was selected due to its consistency with the practical operating conditions of the absorption refrigeration system.

To model the system, these following assumptions had been established:

1. In a vapor absorption refrigeration system, the generator and evaporator pressures are kept at various pressures.
2. The operating pressure is maintained, during the runs.
3. It is assumed that, the refrigerant leaving condenser and evaporator is saturated liquid and vapor, respectively.
4. Before leaving to the absorber, the solution boils in the generator.
5. The evaporator did not experience any liquid spillover.
6. Cycle makes use of adiabatic throttling valves.
7. Pumping is an isentropic process.
8. Isolated system.

4.1 Calculations of Concentration

Eqs (19) and (20) given by Lansing [19] are used to calculate the concentration of the strong solution and the weak solution. Concentration at points 1, 2 and 3 will remain same as concentration of strong solution (X_{SS}), because in cycle, the strong solution passes through points 1, 2 and 3 on its way, from the absorber to the generator.

$$X_{SS} = \frac{49.04+1.125T_A-T_E}{134.65+0.47T_A} \tag{19}$$

Concentration at points 4,5 & 6 remains same as the concentration of weak solution (X_{WS}), because the weak solution in the cycle travels through points 4,5 & 6 on its way, from the generator to the absorber.

$$X_{WS} = \frac{49.04+1.125T_G-T_C}{134.65+0.47T_G} \tag{20}$$

From the generator to the absorber through the condenser and evaporator, the refrigerant travels during the cycle (points 7, 8 and 9). Since pure water works as the refrigerant, there is no concentration of LiBr at points 7, 8, 9 and 10.

It is necessary to remember that the concentration of the solution in the generator should be greater than the concentration of the solution in the absorber, and the concentration of the solution in the generator should not be exceed the value of (65%) in order to avoid crystallization of lithium bromide, which represents an essential issue in the operation of the system.

4.2 Temperature Calculation

As shown in Eq. (13), heat exchanger effectiveness is used to determine inlet and outlet solution heat exchanger temperatures.

$$T_5 = T_G - \varepsilon_{shx}(T_G - T_A)$$

4.3 Mass Flow Rate Calculations

Enthalpy differences at the condenser and evaporator is used to calculate the mass flow rate at Point 9, Eq. (8) [20, 21]:

$$\dot{m}_9 = \frac{Q_E}{(h_{10}-h_9)}$$

The difference between the mass flow of the refrigerant and its concentration is used to calculate the mass flow rates at points 3 and 4.

$$\dot{m}_4 = \dot{m}_9\left(\frac{X_{SS}}{X_{WS}-X_{SS}}\right) \tag{21}$$

$$\dot{m}_3 = \dot{m}_9\left(\frac{X_{WS}}{X_{WS}-X_{SS}}\right) \tag{22}$$

4.4 Pressure Calculations

The pressure calculation equation given by Lasing [19] is used to determine the pressure at the evaporator and absorber:

$$log_{10} P_E = 7.8553 - \frac{1555}{(T_E+273.15)} - \frac{11.2414\times10^4}{(T_E+273.15)^2} \tag{23}$$

$$log_{10} P_C = 7.8553 - \frac{1555}{(T_C+273.15)} - \frac{11.2414\times10^4}{(T_C+273.15)^2} \tag{24}$$

4.5 Thermal Properties Calculations for the Refrigerant and LiBr-H₂O Solution

The polynomial fit given by ASHRAE is used to obtain the enthalpy of the LiBr-H_2O solution [7].

Point 1: Enthalpy (h_1=h_2) of X_1 LiBr-H_2O solution at T_1.
Point 3: Enthalpy (h_3) of X_3 LiBr-H_2O solution at T_3.
Point 4: Enthalpy (h_4) of X_4 LiBr-H_2O solution at T_4.
Point 6: Enthalpy (h_6=h_5) of X_6 LiBr-H_2O solution at T_6.
Point 7: Enthalpy (h_7) of saturated water vapor at T_7 and P_7.
Point 8: Enthalpy (h_8=h_9) of water saturated liquid at T_8 and P_8.
Point 10: Enthalpy (h_{10}) of saturated water vapor at T_{10} and P_8.

4.6 COP of the System

The ratio of the heat absorbed in the evaporator and the power supplied / used to the generator and the pump represents the performance of the vapor absorption refrigeration system (coefficient of performance).

$$COP = \frac{Q_E}{Q_G + W_p} \tag{25}$$

Where the work done by pump (W_P) is neglected.

The maximum coefficient of performance for the absorption system is given by:

$$COP_{maximum} = \left[\frac{T_E}{T_C - T_E}\right]\left[\frac{T_G - T_A}{T_G}\right] \tag{26}$$

Where the temperatures in (Kelvin).

The ratio of actual coefficient of performance to the maximum coefficient of performance for the system is "Relative Performance Ratio", which is given by:

$$COP_{relative} = \frac{COP}{COP_{max}} \tag{27}$$

5. RESULTS AND DISCUSSION

The study was carried out under different operating conditions using (MATLAB R2019a). To check the validity of the program, the obtained results for evaporator capacity of 1 TR is compared with Lansing [19]. The result of comparison is very good as shown in Table 1. This comparison gives confidence in the following results and the results were as follows:

Table 1: Comparison of results between current study and Lansing [19].

Parameter	Current Study	Lansing [19]
Q_E (Kw)	3.517	3.517
Q_C (Kw)	3.7355	3.7475
Q_G (Kw)	4.6085	4.532
Q_A (Kw)	4.3384	4.3014
COP	0.7631	0.776
$COP_{max.}$	1.1688	1.1689
$COP_{rel.}$	0.6528	0.664

T_A (℃)	T_C (℃)	T_E (℃)	T_G (℃)	E_{shx}
40	40	7	90	0.8

5.1 Effect of solution heat exchanger effectiveness

The effect of solution heat exchanger effectiveness was calculated by keeping the temperatures of absorber, condenser, evaporator and generator equal to (32℃, 31℃, 7℃, 75℃) respectively, while effectiveness of solution heat exchanger was varied (from 0.6 to 0.8). It was found that, the effectiveness of solution heat exchanger does not affect on concentration of both strong and weak solution, the heat gained at evaporator, the heat released from condenser. While the heat input to the generator and the heat released from absorber decreases when the effectiveness of solution heat exchanger increases. The COP increases when the effectiveness of solution heat exchanger increases.

Fig. 2 shows the effect of the effectiveness of solution heat exchanger on COP, heat released from absorber, and heat input to the generator. The explanation for this is that increasing the effectiveness of solution heat exchanger leads to raising the temperature of the solution that enters the generator, and as a result, the difference in the temperature of the solution in the generator will decrease, and this leads to a decrease in the amount of heat exchanged in the generator, which results in better performance of the absorption refrigeration system. It is important to note that the area needed for the heat exchanger increases with increasing the effectiveness of solution heat exchanger. As a result, the effectiveness of solution heat exchanger was chosen to be equal to (0.8) in this study. As it is suitable in terms of performance and area.

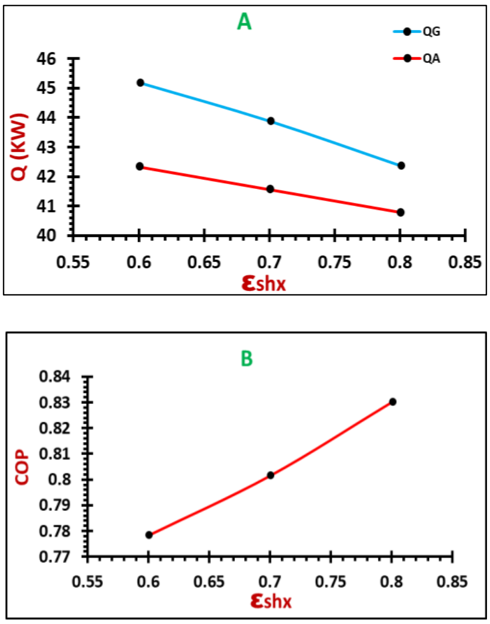

Fig. 2: Effect of the Effectiveness of Solution Heat Exchanger on: (A) heat released from absorber and heat input to the generator. (B) COP.

5.2 Effect of absorber temperature

The effect of absorber temperature was calculated by keeping solution heat exchanger effectiveness, and the temperatures of condenser, evaporator, and generator equal to (0.8, 31°C, 7°C, 75°C), respectively, while the temperature of absorber was varied (from 30 to 45°C). It was found that, concentration of strong solution, heat released from absorber and heat input to generator increase with increasing absorber temperature as presented in fig. 3. While the coefficient of performance decreases as absorber temperature increases. The temperature of absorber does not affect on weak solution concentration, the heat gained at evaporator, and the heat released from condenser. Since the amount of heat required for the generator increases with the increase in the absorber temperature as a result for the increase in the flow rate of the weak solution leaving the generator, the coefficient of performance of the system decreases. However, it is better to remove the heat from absorber, and reduce its temperature to prevent crystallization in the absorber tubes as a result of the increase in the concentration of the strong solution in it. According to the results that obtained, the best value chosen for the absorber temperature is (32°C).

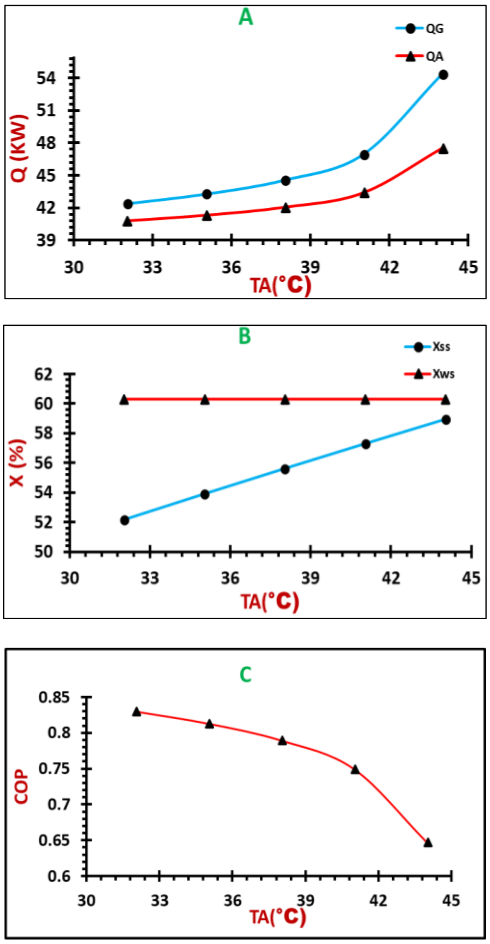

Fig. 3: Effect of temperature of absorber on: (A) heat released from absorber and heat input to generator. (B) strong and weak solution concentration. (C) coefficient of performance.

5.3 The effect of condenser temperature

The condenser temperature effect was calculated by keeping solution heat exchanger effectiveness and the temperatures of absorber, evaporator, and generator equal to (0.8, 32°C, 7°C, 75°C), respectively, while the condenser temperature was varied (from 30 to 45°C). It was found that the concentration of weak solution and the coefficient of performance decrease with increasing the condenser temperature. While the heat input to the generator and heat released from the absorber is increased very slightly till the temperature reaches about (40°C) then they increased in a considerable manner. The effect of the condenser temperature on the heat released from condenser is neglected. Furthermore, the temperature of condenser does not affect on strong solution concentration, and heat gained at evaporator. The above effects are depicted in Fig. 4. Increasing the condenser temperature causes an increase in the pressure of the generator and the condenser, and a decrease in the concentration of the weak solution in the generator while the generator temperature remains constant, which causes an increase in the flow rate of the solution that leaves the generator and enters the absorber. We infer that the refrigerant that evaporate in the generator decreases as the condenser temperature rises and the amount of heat required for the generator increases, which leads to a decrease in the system's performance. From the above argument, it can be concluded that the condenser temperature of (31°C) is considered the best.

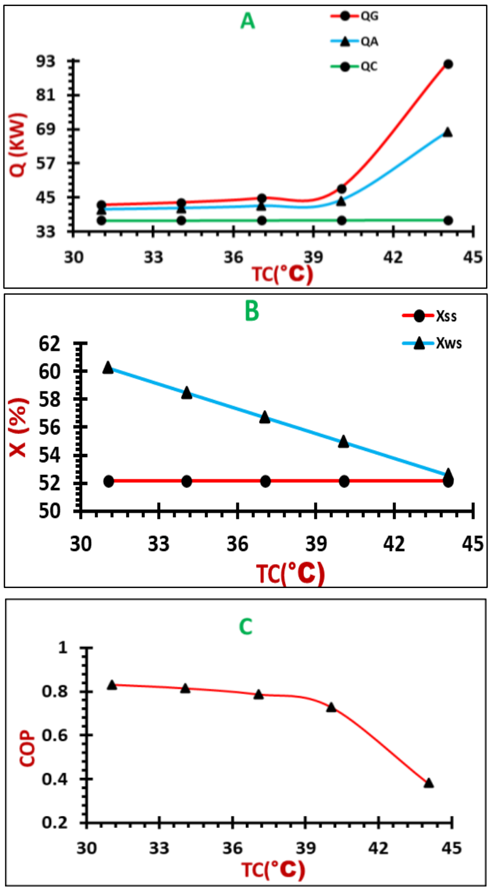

Fig. 4: Effect of temperature of condenser on: (A) heat released from condenser, heat released from absorber and heat input to the generator. (B) strong and weak solution concentration. (C) coefficient of performance.

5.4 The effect of evaporator temperature

The evaporator temperature effect was predicted by keeping solution heat exchanger effectiveness and the temperatures of absorber, condenser, and generator are constants and equal to (0.8, 32°C, 31°C, 75°C) respectively, while the evaporator temperature was varied (from 5 to 12°C). From Fig. 5 it can be noticed that the concentration of strong solution, heat released from absorber, and the heat input to generator decrease by increasing the temperature of evaporator, however the effect of evaporator temperature on heat released from condenser, the concentration of weak solution and heat gained at evaporator can be neglected. Coefficient of performance increases as the temperature of evaporator increases. As a result of the decrease in the amount of heat required to the generator due to the increase in the evaporator temperature, the coefficient of performance of the refrigeration system increases. The increase in the pressure of the evaporator and absorber is a result of the increase in the evaporator temperature. This will result in a decrease in the concentration of the strong solution in the absorber, which in turn reduces the flow of the solution that leaves the absorber to generate a sufficient amount of refrigerant vapor. This means that when the generator temperature remains constant, the flow rate of the weak solution that leaves the generator and enters the absorber decreases, and thus the amount of heat exchanged in the generator decreases. Moreover, the low flow rate of the weak solution reduces its ability to absorb refrigerant vapor in the absorber. Therefore, increasing the evaporator temperature leads to an increase in the coefficient of performance. According to theoretical calculations, the evaporator temperature (7°C) is considered the best choice.

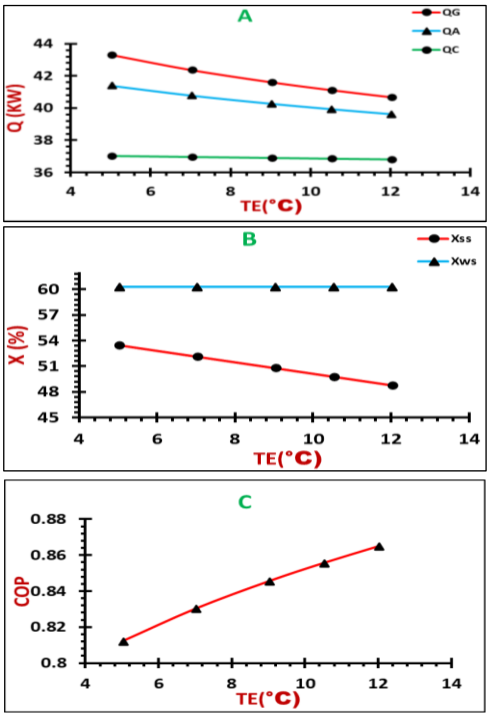

Fig. 5: Effect of the Evaporator Temperature on: (A) heat released from condenser, heat released from absorber and heat input to the generator. (B) concentration of strong and weak solution. (C) COP.

5.5 The effect of the generator temperature

The generator temperature effect was studied by setting the values of effectiveness of solution heat exchanger, the temperatures of absorber, condenser and evaporator at (0.8, 32℃, 31℃, 7℃), respectively. The generator temperature was varied (from 70 to 95℃). From Figure 7, as the temperature of generator increases, concentration of weak solution, heat released from condenser and coefficient of performance increase. However, the increase in COP, heat released from condenser and absorber are very small and can be neglected. Heat input to generator decreases very slightly, as the temperature of generator increases. Also, the temperature of generator does not affect on concentration of strong solution and heat gained at evaporator. The generator temperature should not be exceeded (92℃) in order to avoid crystallization of the lithium bromide, because the concentration of the weak solution should not be exceeded (65%). As shown from the above results, the value of (75℃) for generator temperature is regarded adequate.

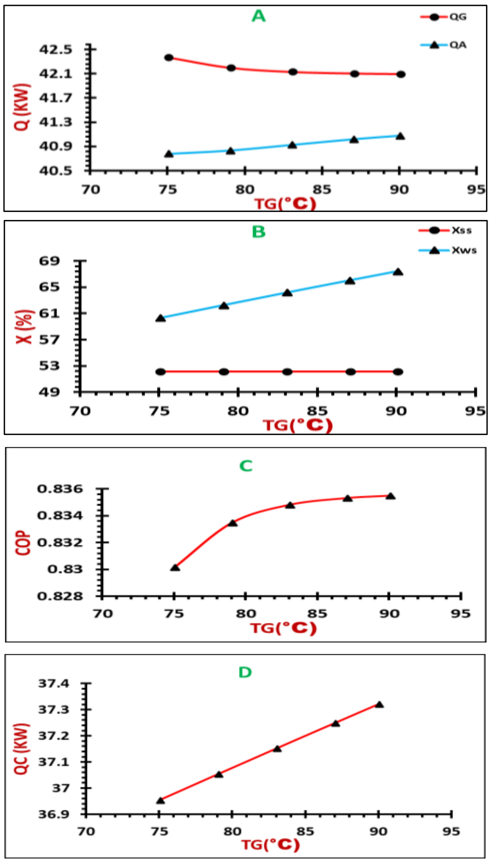

Fig. 6: Effect of the Generator Temperature on: (A) heat released from absorber and heat input to the generator. (B) concentration of strong and weak solution. (C) COP. (D) heat released from condenser.

From the above results, the COP computed is (0.83), and the maximum COP computed is (1.44), while the relative COP computed is (0.575).

Finally, Tables 2 and 3 summarize the results obtained after selecting the best operating temperatures and effectiveness of heat exchanger.

Table 2: Output results for vapor absorption refrigeration system.

RE (Ton)	QE (KW)	QC (KW)	QG (KW)	QA (KW)	COP	COP_{max}	$COP_{relative}$
10	35.17	36.953202	42.366604	40.777445	0.83013	1.44172	0.57579

Table 3: Operating conditions for vapor absorption refrigeration system.

State	T (℃)	P (kPa)	X ($kg_{LiBr}/kg_{Solution}$)	m (kg/sec)	h (kJ/kg)
1	32	0.9937	52.1344	0.1091	68.5743
2	32	4.4916	52.1344	0.1091	68.5743
3	59.52	4.4916	52.1344	0.1091	127.51
4	75	4.4916	60.2795	0.0944	184.4578
5	40.6	4.4916	60.2795	0.0944	118.3691
6	40.6	0.9937	60.2795	0.0944	118.3691
7	75	4.4916	Xref	0.0147	2634.6026
8	31	4.4916	Xref	0.0147	129.9255
9	7	0.9937	Xref	0.0147	129.9255
10	7	0.9937	Xref	0.0147	2513.7376

6. CONCLUSION

The goal of the current work is a theoretical design of (10TR) single-effect LiBr-H_2O absorption refrigeration system. Different design parameters have been studied. According to the results, the temperatures of generator 75℃, absorber 32℃, condenser 31℃, and evaporator 7℃ and the effectiveness of solution heat exchanger 0.8 have been selected. These predicted values will make it easier to design the various components of the system.

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NOMENCLATURE

Symbol	Definition	Unit
$LiBr$	Lithium Bromide	------
$\dot{m}$	Mass flow rate	kg/sec.
h	Enthalpy	kJ/kg
P	Pressure	kPa
Q	Heat transfer rate	kW
T	Temperature	℃
X	Concentration of solution	%
ε	Effectiveness of solution heat exchanger	------
W	Work done	kW
υ	Kinematic viscosity	$m^2/sec.$

SUBSCRIPT

Symbol	Definition
A	Absorber
C	Condenser
E	Evaporator
G	Generator
shx	Solution heat exchanger
SS	Strong Solution
WS	Weak Solution
in	Entering
out	Exiting
1,2,3,…	Position as mentioned in Figures

تصميم ديناميكي حراري لمنظومة تبريد امتصاصية بروميد الليثيوم - ماء أحادية التأثير سعة 10 طن

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الملخص

إن استخدام الطاقة، الذي ينظر إليه على أنه عنصر أساسي في تطور الحضارة هو ضروري لجميع جوانب الحياة الحديثة، وهو أحد مقاييس نمو البلد. نظراً لارتفاع تكاليف مصادر الطاقة التقليدية بالإضافة إلى احتمال استنفادها، فقد أصبح من الضروري البحث عن مصادر جديدة ومباشرة للطاقة. كنتيجة لذلك، بدأ التفكير للعودة إلى أنظمة التبريد بالامتصاص بسبب قدرتها على العمل باستخدام الطاقة الحرارية المباشرة. بصرف النظر عن بساطتها وقلة أجزائها المتحركة، فإن أنظمة تبريد الامتصاص شائعة الاستخدام لأنها يمكن أن تعمل بمصدر حراري ذي درجات حرارة منخفضة نسبياً، مثل تلك الناتجة عن حرق الغاز الطبيعي أو مجمعات الطاقة الشمسية، لإنتاج تأثير التبريد مباشرة عن طريق تبخير مائع التبريد. تقدم هذه المقالة التصميم الديناميكي الحراري لمنظومة تبريد امتصاصية (بروميد الليثيوم - الماء) أحادية التأثير سعة 10 طن. تم استخدام سعة المبخر (35.17 كيلو واط) لتحديد معاملات التشغيل لكل مكون. لقد نفذت المحاكاة الديناميكية الحرارية على أساس العلاقات التجريبية. لتحديد المعلمات التشغيلية المختلفة لمنظومة التبريد الامتصاصية في ظل ظروف تشغيل مختلفة، تمت البرمجة باستخدام برنامج الماتلاب. تم أخذ فعالية المبادل الحراري للمحلول ودرجات الحرارة المختلفة للمولد، المكثف، المبخر، ووعاء الامتصاص في الاعتبار عند حساب معامل الاداء. أثبتت النتائج التي تم الحصول عليها تثبت أن معامل الأداء يزداد بزيادة كل من فعالية المبادل الحراري للمحلول، درجات حرارة المولد والمبخر، ويقل بزيادة كل من درجات حرارة وعاء الامتصاص والمكثف.

الكلمات الدالة:

بروميد الليثيوم - ماء، أنظمة التبريد الامتصاصية، تكييف الهواء، تصميم ديناميكي حراري.