

Energy and Exergy Analysis of Single Effect Water-LiBr Vapour Absorption Refrigeration System

Natnale Sitotaw^{a,1,*}

^a College of Engineering and Technology, Department of Mechanical Engineering, Bonga University, Ethiopia

¹ sitotawnati2017@gmail.com

* corresponding author

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ABSTRACT

Vapor absorption refrigeration system (VARS) is one of the emerging technology in thermal-driven refrigeration systems and has too many benefits in comparison with another cooling system as their performance is good and the cost is low. Different mathematical models of various complexities have been developed for performance analysis, optimization, and design of such systems. In this study energy and exergy of a single effect LiBr vapor absorption refrigeration system is analyzed with different mathematical model developed so far. The effect of varying the temperature for each component of the system on the coefficient of performance (COP) and exergetic efficiency (COP_{ex}) is briefly discussed. The results show that increasing generator temperature from 86°C to 102°C for constant refrigerant mass flow rate causes COP of the system to increase, due to the decrease in circulation ratio. In another case, increasing the evaporator temperature from 2°C to 14°C for a constant temperature input of the generator has a positive effect on both the first and second law efficiency of the VARS as the load on the generator decrease. From the result, it can also be concluded that increasing absorber temperature has a negative impact on both COP and exergetic efficiency of VARS.

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1. Introduction

Through the past decades, the demand for refrigeration increases in substantial growth. According to the international institutes of refrigeration (IIR), conventional refrigeration eats up one-fifth of the electric consumption worldwide [1]. Vapor Absorption refrigeration system is an attractive method for using low-grade energy directly for cooling. This is the favorability of VARS compared to the conventional one. However, the net energy and exergetic efficiency of such units are often lower mainly due to high exergetic losses during isothermal boiling and condensation, and high parasitic load for heat rejection [2].

The operating principle of the VARS is close to that of the vapor compression system with two major differences. First, the VARS is a heat-propel thermal system: only thermal energy is interchanged with the surrounding with a limited transfer of mechanical energy. The second one is, unlike the vapor compression system, a secondary fluid in addition to the refrigerant known as liquid sorption medium or absorbent is used in the system [3]. VARS can be grouped based on several criteria, but the classification based on the working fluid is considered in this study. From various working pairs of VARS, the usual and commercial ones are water-lithium bromide (H₂O-LiBr) and ammonia-water NH₃-H₂O solution.

The two VARS have the same working principles with some crucial variability; the volatility of the absorbent, the difference in pressure and solubility ranges, lower latent heat of vaporization of ammonia compared to water. Thus, for the same duty, the refrigerant and absorbent mass-circulation rates are roughly double that of H₂O-LiBr [4]. Besides these main variations, due to the small boiling point difference between ammonia and, both ammonia and water boiled out from the solution in the generator. On account of this, three additional components are used in NH₃- H₂O VARS: a rectification column, a dephlegmator, and a sub-cooling heat exchanger. Furthermore, H₂O-LiBr VARS operate under vacuum pressure, whereas, NH₃- H₂O VARS work at a pressure much higher than the atmospheric pressure [5] and also, unlike refrigerant water, ammonia is both toxic and flammable.

H₂O-LiBr VARS is substantially used in medium and large-capacity air conditioning and refrigeration systems. In such a system, water is used as a refrigerant, and lithium bromide is used as an absorbent. Since water is the refrigerant, this system is not used under sub-zero temperature conditions [6]. The affinity of H₂O-LiBr pair, low volatility and corrosive action, low-pressure operation, and eliminating the need for rectifier makes it a preferred pair of refrigerant over ammonia-water mixture [7].

Shun and Sherif, [8] studied A H₂O-LiBr absorption refrigeration system for cooling and heating applications based on the first and second laws of thermodynamics. In their parametric analysis of VARS, the result shows that low cooling water temperature yields both higher cooling COP and higher exergetic efficiency. Besides this, increasing the generator temperature can improve COP of the absorption system, but as the heat source temperature increases beyond a certain optimal value, the COP of the system keeps constant and even decreased. Villa *et al.* [10] studied theoretically the performance of a single-stage H₂O-LiBr VARS driven by hot water. The equations are analysed using the EES-32 program (engineering equation solver version 32). The data generated by variation of the inlet temperature of the hot water were used for the calculation of a range of values of COP and exergetic efficiency with a maximum value of 0.7406 and 0.2409, respectively. Their analysis revealed that the major irreversibility was developed in the cooling tower, generator, and absorber with a value of 34, 32, and 16%, respectively.

2. Material and Method

Figure 1 shows the line diagram of a single effect H₂O-LiBr VARS. The system process takes place due to the interactivity of the four independent circuits: hot water, water-LiBr solution, cooling water, and chilled water circuits. From state 4 to 1, a weak solution from the absorber is forced to the generator through the solution heat exchanger (SHX) to desorb the refrigerant water from the absorbent (LiBr). A heat source provides heat to the generator, and the heat supplied evaporates the refrigerant H₂O at high pressure from state 1-5. Once the refrigerant evaporates, it is led to the condenser and the condenser dissipates heat to the environment while the refrigerant changes its phase from vapor to liquid state from state 5-6.

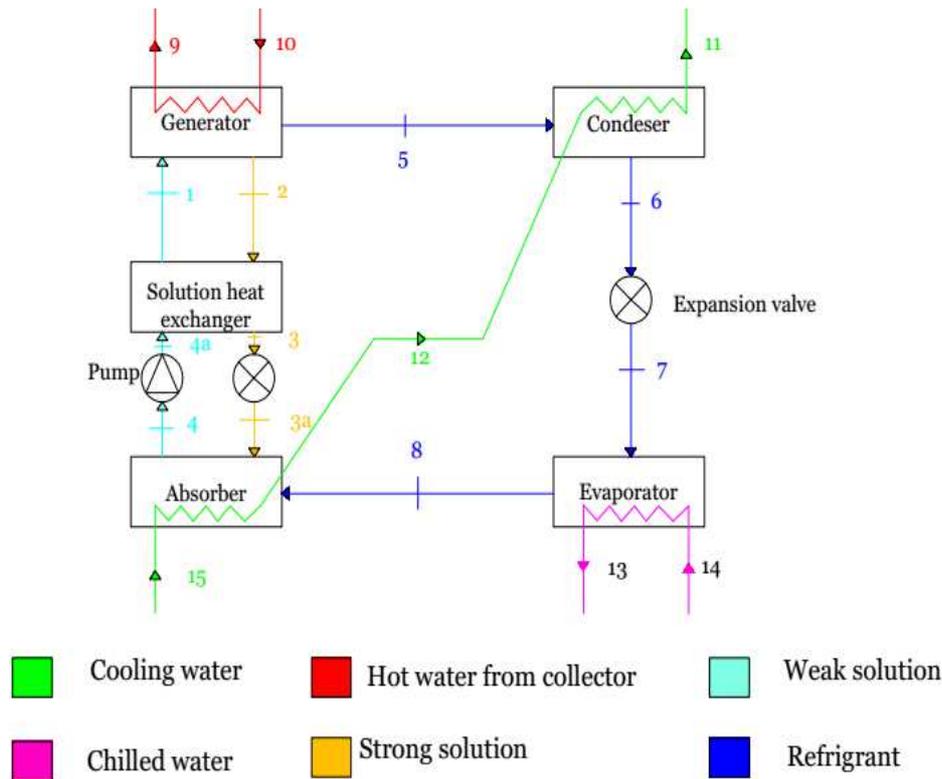


Figure 1. Schematic diagram of single effect VARS

Then, the refrigerant flows through the expansion valve from state 6-7, to reduce its pressure and subsequently leads to the evaporator. From state 7-8, the cooling process takes place inside the evaporator by absorbing heat from the utility. Finally, the water vapour from the evaporator flow to the absorber from state 8-4 where it mixes with the strong solution from state 8-2, which comes from the generator through the heat exchanger. The heat exchanger between generator and absorber, improves the efficiency of the chilled, by preheating the weak solution that returns to the generator.

The concentration of lithium bromide based on the mass fraction(x) can be expressed as [6].

$$X = \frac{m_L}{m_L + m_w} \tag{1}$$

Where m_L and m_w are the mass of anhydrous lithium bromide and water, respectively. The quantity of rich solution handled by the pump per unit mass of the vapour distilled is called circulation ratio (f) [6].

$$f = \frac{m_g}{D} = \frac{x_r}{x_r - x_w} \tag{2}$$

Where x_r is the concentration of the rich solution, D is the mass flow rate of distilled vapour, x_w is the concentration of a weak solution from and, $x_r - x_w$ is degassing range. Form this the quantity, weak solution per unit mass of distilled water is $(f - 1)$. In addition to these two property, molality which is the number of moles of a solute per kg of a solvent can be calculated for the LiBr mass fraction (X_{LiBr}) as given in the following equation [12].

$$m = \frac{X_{LiBr}}{(1 - X_{LiBr}) M_{LiBr}} \tag{3}$$

Where, X_{LiBr} and M_{LiBr} is the mole fraction and molar mass of LiBr respectively. Modelling of the single effect VARS is analysed by applying the mass, energy, and exergy balance for each component. Some simplifying assumptions as shown below are considered to design and simulate the system [4]:

- The complete system is in thermodynamic equilibrium.
- The study is made under steady-state conditions.
- Heat gain by the pump is insignificant as compared to the primary energy input for the generators.
- No pressure drop due to friction in the pipe.
- The solution is at a saturated state when leaving the generator and absorber, and the refrigerant is at a saturated state when leaving the condenser and evaporator.
- Heat loss from the system to the environment is neglected.
- Flow through the valve is isenthalpic.

The variation in potential and kinetic energy is In addition to this, the empirical equations of enthalpies, temperatures, concentrations, and vapour pressures of the H₂O-LiBr solutions as given by ASHRAE are used.

2.1 Entropy of water/LiBr solution

Exergy analysis of VARS demands a thermodynamic property called entropy, which is a quantifiable measure of microscopic disorder. Entropy change is created due to heat transfer by work or by mass transfer, and irreversibility such as mixing and friction [11]. Clausius in 1865 perceives that the entropy change of the system is given by,

$$ds \geq \left(\frac{\partial Q}{T}\right) \quad (4)$$

Equation 4 Called Clausius inequality. Where ∂Q is the heat flow, T is temperature, ds is the change in entropy of the system. Exergy, which is the work potential of a source, that is, the amount of energy that can be extracted as useful work can be calculated from this basic thermodynamic property. Entropy, unlike energy, has a one-directional flow. Meaning, the entropy of a given process is always increasing called the principle of increasing entropy. From Equation 4, the equality holds for an internally reversible process and the inequality for an irreversible process. Entropy is a property, and thus the value of entropy of a system is fixed once the state of the system is fixed [11]. However, unlike the entropy of a pure substance, the entropy of a mixture is somehow complicated due to the variation with both concentration and the state of the system. The molar entropy of a binary mixture 1 and 2 is given as [12].

$$\bar{S} = y_1 \bar{S}_{1(T,P)} + (1-y_1) \bar{S}_{2(T,P)} - y_1 \cdot v \cdot \bar{R} \left[\ln \left(\frac{m}{m_0} \right) - 1 \right] + \bar{S}_{(T,P,m)}^E \quad (5)$$

Where $\bar{S}_{1(T,P)}$ is the molar entropy of fluid 1 and $\bar{S}_{2(T,P)}$ is the molar entropy of fluid 2. The third term is the entropy generation in an ideal mixture (m_0 is the standard molality: $m_0 = 0.001 \text{ kmol kg}^{-1}$ of solvent) and the last term $\bar{S}_{(T,P,m)}^E$ is the additional entropy generation for a real mixture process. y_1 is the mole fraction of the solute. v and \bar{R} represent the dissociation number and the universal gas constant, respectively. The first two terms at the right side of (5) can be calculated as [13].

$$\bar{S} = \bar{S}_{0,(T,P)}^\infty + \int_{T_0}^T \frac{Cp}{\partial T} dT - \int_{P_0}^P \frac{\partial V}{\partial T} dP \quad (6)$$

Where, $\bar{S}_{0,(T,P)}^\infty$ is the entropy of a substance at the standard temperature and pressure. T , P , and V are temperature, pressure, and volume, respectively. Considering Eq. (5) and (6), to determine the entropy of water/LiBr solution, the correlation proposed is somewhat satisfactory regression results as in [12].

$$\bar{S} = y_{LiBr} \cdot \bar{S}_{LiBr}^\infty + (1 - y_{LiBr}) \cdot \bar{S}_{H_2O(T,P)}^1 - y_{LiBr} \cdot v \cdot \bar{R} \left[\ln \left(\frac{m}{m_0} \right) - 1 \right] + \bar{S}_{(T,P,m)}^E \tag{7}$$

Where the term \bar{S}_{LiBr}^∞ is the molar entropy of the ideal LiBr fluid, and $\bar{S}_{H_2O(T,P)}^1$ is the molar entropy of pure water. And each respective value is given below as stated by Kim and Ferreira, [12].

$$\bar{S}_{LiBr}^\infty = \bar{S}_{LiBr;0}^\infty - R \cdot \left[\frac{c_0}{2} \left(\frac{1}{T^2} - \frac{1}{T_0^2} \right) \right] \left[\frac{c_1}{3} \left(\frac{1}{T^3} - \frac{1}{T_0^3} \right) \right] + \left[\frac{c_2}{4} \left(\frac{1}{T^4} - \frac{1}{T_0^4} \right) \right] - R \cdot (b_{00} - \frac{b_{02}}{T^2}) \cdot (P - P_0^*) \tag{8}$$

$$\bar{S}_{H_2O}^1 = \bar{S}_{H_2O;0}^1 + R \left(d_0 \ln \frac{T}{T_0} + d_1 (T - T_0) + \frac{d_2}{2} (T^2 - T_0^2) - R \cdot (e_1 + 2e_2 \cdot T) (P - P_0^*) \right) \tag{9}$$

$$\bar{S}_{(T,P,m)}^E = y_{LiBr} \cdot v \cdot R \sum_{i=1}^6 \left[a_i + \frac{i \cdot b_i}{2 \cdot v} P + T \cdot \left(\frac{\partial a_i}{\partial T} + \frac{i}{2 \cdot v} \frac{\partial b_i}{\partial T} P \right) \right] m^{i/2} \tag{10}$$

Where: $a_i = \sum_{j=0}^2 a_{ij} T^{-j}$ (11)

$b_i = \sum_{j=0}^2 b_{ij} T^{-j}$ (12)

Where, R represents the universal gas constant in kJ/km.K and v are the dissociation number for the solute (2 for LiBr). The regression constant from Eq. 8 to Eq. 12 is given in Table 1.

Table 1. Constants in equation 8 to 11, [13]

J	0	1	2
a_{1j}	-2.196 x 10 ¹	4.93723 x10 ³	-6.55480 x10 ⁵
a_{2j}	-3.810475 x 10 ³	2.611535 x10 ⁶	-3.6699691 x 10 ⁸
a_{3j}	1.22808 x 10 ⁵	-7.7187923 x 10 ⁷	1.039856 x 10 ¹⁰
a_{4j}	-1.471674 x 10 ⁶	9.1952848 x 10 ⁸	-1.1894502 x 10 ¹¹
a_{5j}	7.765821 x 10 ⁶	-4.937566 x 10 ⁹	6.317554 x 10 ¹¹
a_{6j}	-1.5118920 x 10 ⁷	9.839997 x 10 ⁹	-1.273789 x 10 ¹²
b_{0j}	-4.41786 x 10 ⁻⁵	3.115 x 10 ⁻²	-4.3611
b_{1j}	3.074 x 10 ⁻⁴	-1.863 x 10 ⁻¹	2.738 x 10 ¹
b_{2j}	-4.08 x 10 ⁻⁴	2.16 x 10 ⁻¹	-2.517 x 10 ¹
c_j	-9.44 x 10 ⁻⁵	-5.8423 x 10 ⁸	0
d_j	1.197 x 10 ¹	-1.83 x 10 ⁻²	2.87 x 10 ⁻⁵
e_j	2.66 x 10 ⁻³	-3.865 x 10 ⁻⁶	7.4648 x 10 ⁻⁹
$\bar{S}_{LiBr;0}^\infty$	47.5562(kJ kmol ⁻¹)	$\bar{S}_{H_2O;0}^1$	0
T_0	273.15 K	P_0	0.6108 kPa

To determine the entropy of the H₂O-LiBr solution an iterative analysis as a function of entropy(x, T) is developed that takes the concentration and temperature of the solution at a given state point.

2.2 Exergy of a Control Volume

Exergy is defined as the maximum useful work acquired from an energy carrier at a given state in any process that brings the energy carrier into equilibrium with its environment. If a large reference environment is at temperature T_o , then the exergy of a material stream can be defined as [8].

$$e = (h - h_o) - T_o(s - s_o) \quad (13)$$

Where e is the exergy of the fluid, h is enthalpy, and s is the entropy. The subscript o refers to the reference state. Exergy balance for the steady-state process of a control volume can be expressed as [8].

$$\Delta E = \sum(m_e)_{in} - \sum(m_e)_{out} + [\sum(Q(1 - \frac{T_o}{T}))_{in} - \sum(Q(1 - \frac{T_o}{T}))_{out}] + \sum W \quad (14)$$

Where ΔE is the lost exergy or the irreversibility that occurred in the process. The first, second, third, and the last term on the right side of Eq.14 represent the exergy transfer by mass at the inlet, the Exergy transfer by mass at the outlet, the exergy transfer by heat, and exergy transfer by work of the control volume, respectively. Exergy transfer by heat is negligible as the system is assumed to be insulated from its environment. And, exergy transfer by work also has a trivial effect for VARS as the work done by the solution pump is very small. This implies Eq. 14 becomes:

$$\Delta E = \sum(m_e)_{in} - \sum(m_e)_{out} \quad (15)$$

For the exergy calculation of water/LiBr solution, the thermodynamic properties are essential. Mainly, enthalpy and entropy are indispensable to calculate the physical exergy. Assuming each component of VARS as a control volume; energy and exergy balance lead to the sets of heat transfer equations discussed in the following sections.

2.3 Performance of VARS

Performance analysis of VARS is based on the study of the first and the second law of thermodynamics. These two analyses rely on the heat and exergy transfer of the system, respectively. The mathematical model used for the heat transfer analysis of VARS refrigerant-absorber combinations is taken from Arora, [6]. Two main circuits in the VARS control the energy and exergy efficiency of the system.

2.3.1 Solution Circuit Energy and Exergy Balance

The general mathematical model for an absorption system for mass, energy, and exergy balance is; Let's say the amount of distilled water from the generator is D so that the strong and weak solution mass flow rate is $(D \cdot f)$ and $(D \cdot (f-1))$, respectively. Energy and exergy balance per unit mass of distilled water for each component of VARS is,

Generator: In the generator, the external hot fluid provides the heat to boil the solution and superheated vapor is generated and flows to the condenser. The energy and exergy balance of the generator is given in Eq. (16) and (17), respectively.

$$Q_g + f \cdot h_1 = h_5 + (f - 1) h_2 \quad (16)$$

$$\Delta E_g = f \cdot (h_1 - T_{o1}) - (f - 1)[h_2 - T_{o2}] + \frac{m_g}{D} [h_9 - T_{o9}] - (h_{10} - T_{o10}) \quad (17)$$

Where Q_g and ΔE_g are heat transfer and exergy gain in the generator. m_g is the hot water mass flow rate, and h , s , and T are specific enthalpy, specific entropy, and temperature, respectively.

The solution heat exchanger (SHX): In the solution heat exchanger the strong LiBr solution which comes from the generator exchanges heat with the weak LiBr solution from the absorber. The energy and exergy balance of SHX is given in Eq. (18) and (19), respectively.

$$Q_{SHX} = f(h_1 - h_4) = (f - 1)(h_2 - h_3) \quad (18)$$

$$\Delta E_{SHX} = f[h_4 - h_1 - T_o(s_4 - s_1)] + (f - 1)[h_3 - h_2 - T_o(s_3 - s_2)] \quad (19)$$

Where, Q_{SHX} and ΔE_{SHX} are heat transfer and exergy destruction in SHX, respectively.

Absorber: In the absorber, the water vapor from the evaporator is absorbed by the strong LiBr solution which comes from the solution heat exchanger, and the heat generated during absorption is rejected to water flow. The weak LiBr solution is pumped through the solution heat exchanger. The energy and exergy balance of the absorber is given in Eq. (20) and (21), respectively.

$$Q_a + f * h_4 = h_8 + (f - 1) h_3 \quad (20)$$

$$\Delta E_a = (h_8 - T_{0s_8}) + (f - 1) (h_3 - T_{0s_3}) - f(h_4 - T_{0s_4}) + \frac{m_a}{D} [(h_{15} - T_{0s_{15}}) - (h_{16} - T_{0s_{16}})] \quad (21)$$

Where, Q_a and ΔE_a are heat transfer and exergy destruction in the absorber, respectively.

2.3.2 Refrigerant circuit energy and exergy balance

The mass flow rate, as well as the composition, is the same at all sections for the refrigerant circuit.

Condenser: In the condenser, the superheated vapor is cooled down and condensed to a saturated liquid by the cooling water. The energy and exergy balance of the condenser is given in Eq. (22) and (23), respectively.

$$Q_c = h_6 - h_5 \quad (22)$$

$$\Delta E_c = (h_6 - T_{0s_6}) - (h_7 - T_{0s_7}) + \frac{m_c}{D} [(h_{11} - T_{0s_{11}}) - (h_{12} - T_{0s_{12}})] \quad (23)$$

Where, Q_c and ΔE_c are heat transfer and exergy destruction in the condenser, respectively.

Expansion valve: Inside the metering device, the saturated refrigerant which comes from the condenser is throttled to drop its pressure. It is assumed that the process occurs in an isenthalpic manner.

$$h_6 = h_7 \quad (24)$$

$$\Delta E_{exp} = (h_7 - h_6) - T_0 (s_7 - s_6) \quad (25)$$

ΔE_{exp} is exergy destruction in the metering device.

Evaporator: In the evaporator, the refrigerant evaporates by absorbing heat from the utility and the vapor is absorbed by the absorber. The energy and exergy balance of the evaporator is given by Eq. (26) and (27), respectively.

$$Q_e = h_8 - h_7 \quad (26)$$

$$\Delta E_e = (h_7 - T_{0s_7}) - (h_8 - T_{0s_8}) + \frac{m_e}{D} [(h_{13} - T_{0s_{13}}) - (h_{14} - T_{0s_{14}})] \quad (27)$$

Where, Q_e and ΔE_e are heat transfer and exergy destruction in the evaporator, respectively. The overall energy and exergy balance which is absorbed and rejected by the system gives:

Overall energy balance

$$Q_{received} = Q_{absorbed} \quad (28)$$

$$Q_G + Q_e = Q_C + Q_a \quad (29)$$

The total rate of exergy destruction of the absorption system is the sum of exergy destruction in each component and can be written as [10]:

$$\Delta E_{sys} = \Delta E_g + \Delta E_a + \Delta E_c + \Delta E_e + \Delta E_{SHX} \quad (30)$$

The coefficient of performance of the VARS is;

$$COP = \frac{Q_e}{Q_G + W_p} \quad (31)$$

Where, W_p is the pump work.

3. Result and Discussion

3.1 Effect of generator temperature on COP and COPex

Variation of COP and total exergy destruction rate (COPex) of a system with the variation of generator temperature is shown in Fig. 2. From Fig. 2 (a) It is observed that increasing the generator temperature (T_g) from 86°C to 102°C has a significant effect on COP by elevating it from 0.4 to 0.75 until it reaches its optimum and then after a negligible variation is observed. From Figure 2(b) as T_g increase from 86°C to 102°C the exergy destruction will increase beyond the optimum T_g so that a lot of energy will be lost without causing a significant effect. From the two Figures, it can also be dictated that increasing the condenser temperature (T_c) from 32°C to 40.5°C has a positive effect for both COP and COPex.

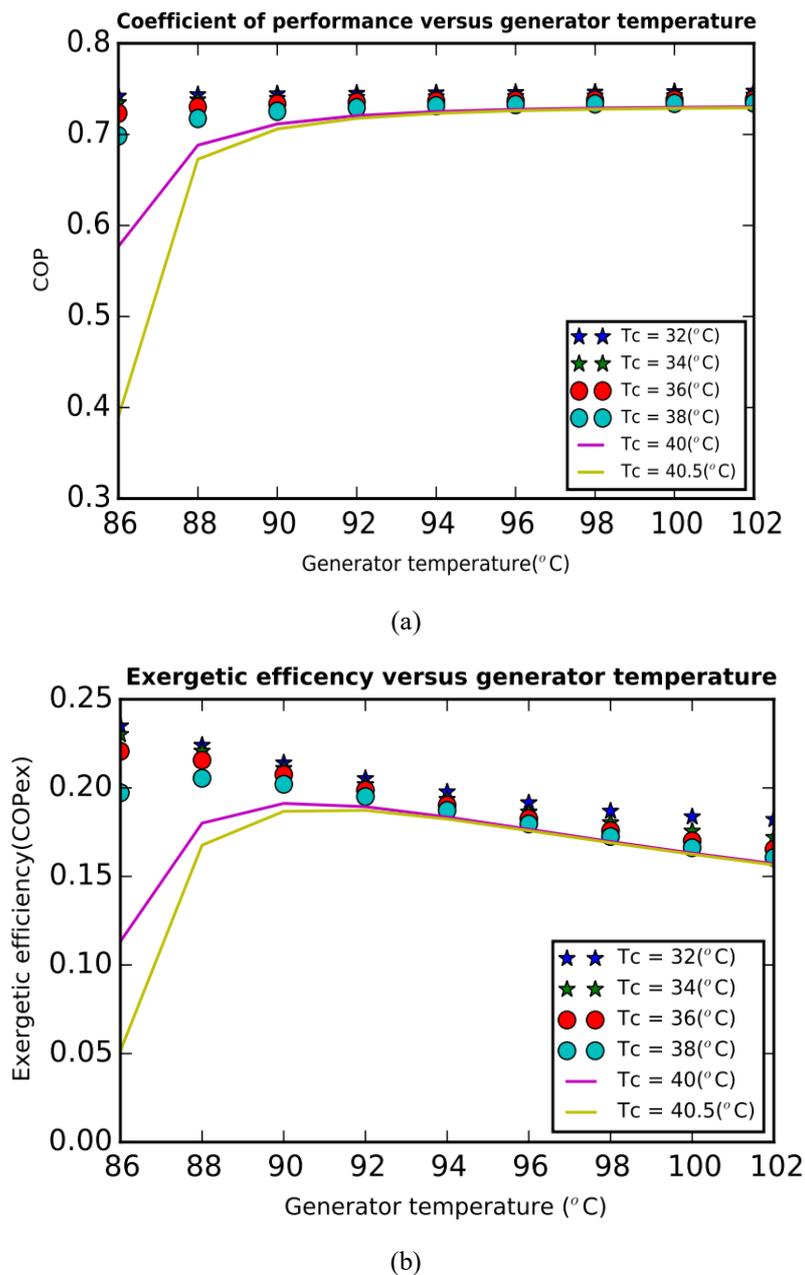


Figure 2. Variation of COP (a) and COPex (b) with generator and condenser temperature

3.2 Effect of evaporator temperature on COP and exergetic efficiency

It is observed from Figure 3 that increasing the temperature of the evaporator from 2⁰c to 16⁰c has elevated COP and exergetic efficiency from 0.755 to 0.795 and 0.154 to 0.34, respectively. This is due to increasing the evaporator temperature give rise to decreasing the amount of heat load required on the generator side. Again, the Figure also shows that increasing the condenser temperature (Tc) from 30⁰c to 38⁰c has a positive effect on both COP and COPex.

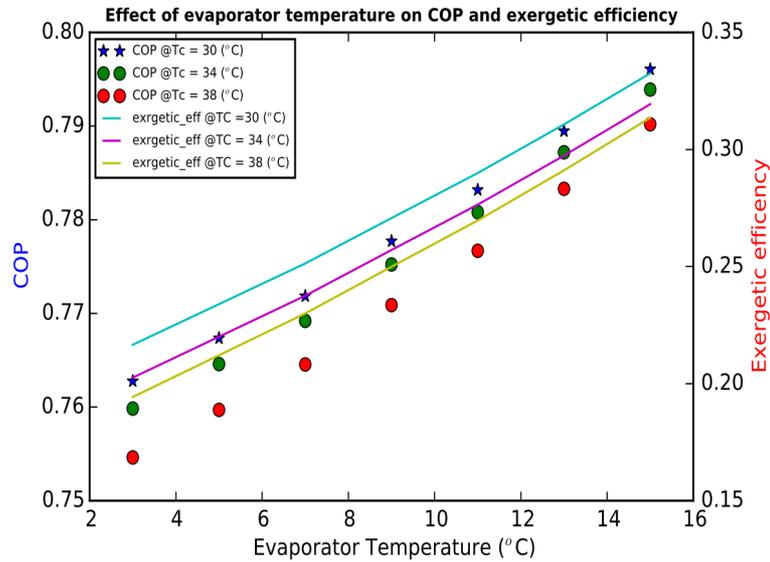


Figure 3. Variation of COP and COPex with evaporator and condenser temperature

3.3 Effect of absorber temperature on COP and exergetic efficiency of the system

Increasing the absorber temperature Ta from 25⁰c to 42.5⁰c for constant Tg has a negative impact on the COP and exergetic efficiency of the system as shown in Fig. 4. The analysis is conducted in the range of Tc (28⁰c to 35⁰c). As a result, the COP and COPex decrease from 0.800 to 0.765 and 0.35 to 0.22, respectively.

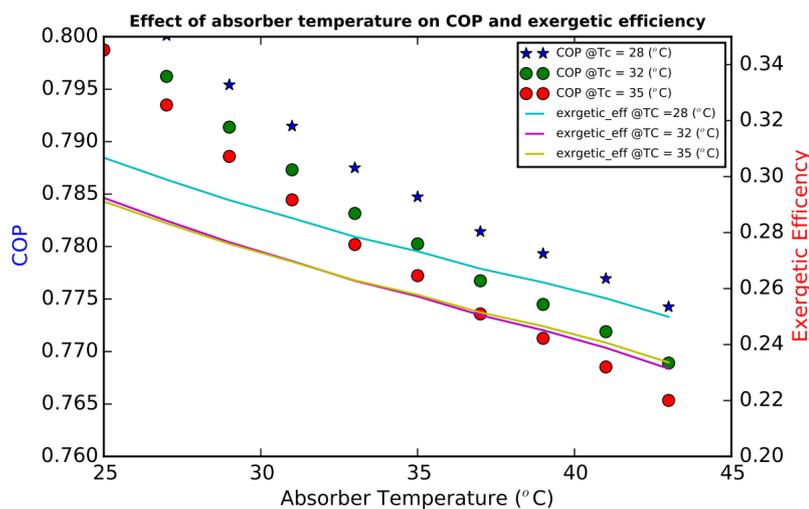


Figure 4. Variation of COP and exergetic efficiency with absorber and condenser temperature

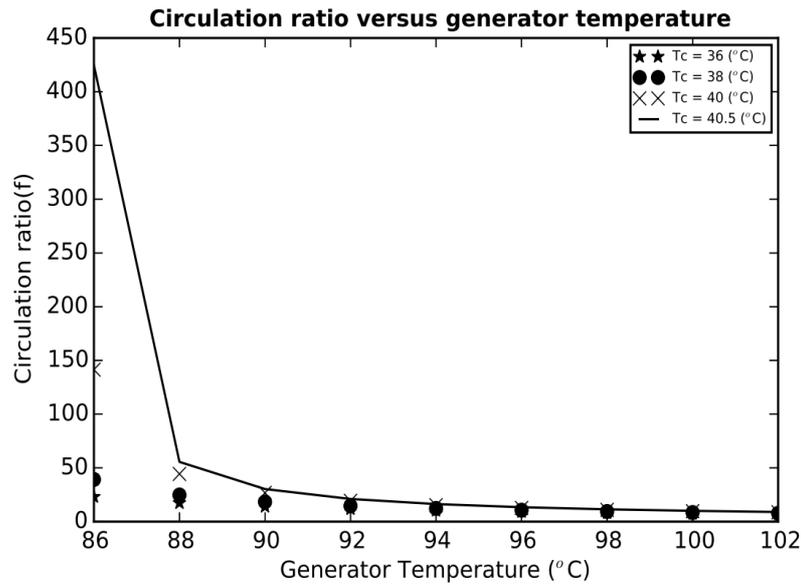


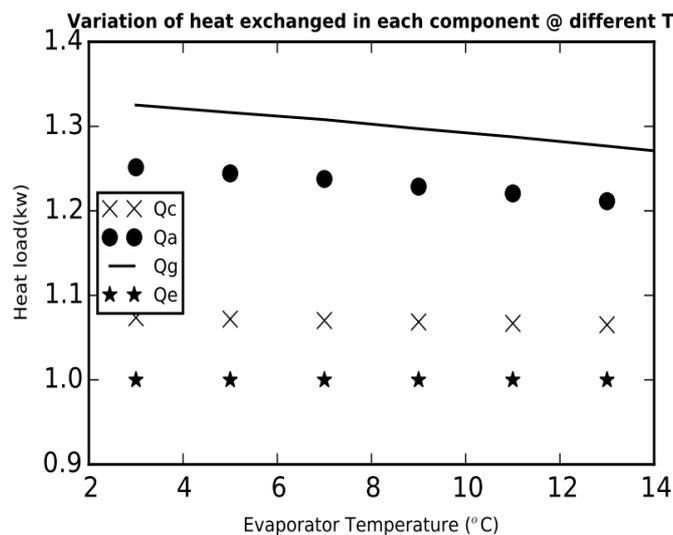
Figure 5. Effect of generator temperature on circulation ratio

3.4 Heat load and exergy destruction per kW of an evaporator load

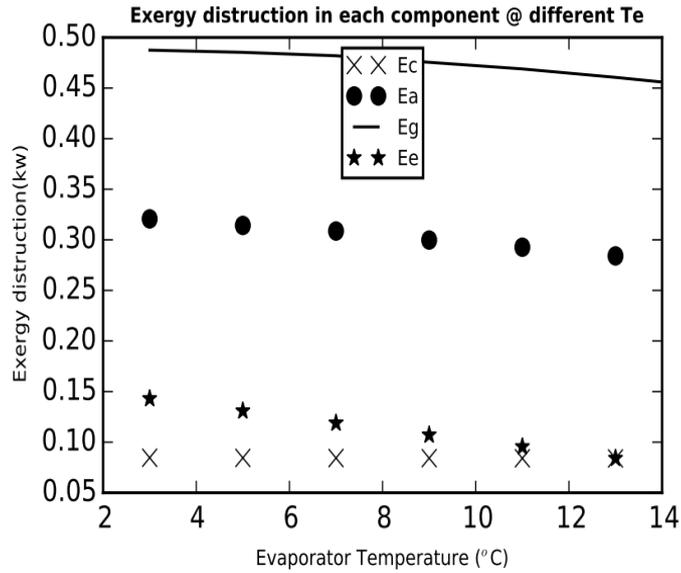
Whenever the study of the performance of VARS is considered, it is very important to analyze the heat load and exergy destruction on each component of VARS for variable input parameters. The response of the system, in heat load and exergy destruction to the variation of the generator and evaporator temperature per kW of an evaporator load, is analyzed.

3.5 Heat Load and exergy destruction versus evaporator temperature

Figure 6 (a) shows that assuming the evaporator temperature (T_e) varying from 2 °C to 14°C makes the generator and absorber load decrease. As T_e increase the concentration of rich solution increase and the circulation ratio (f) decrease. The decrease f , decrease the amount of heat provided to the generator. Fig. 7 (b) shows that whenever T_e increases exergy destruction in the evaporator, absorber and generator will decrease. This is because of the decrease in irreversibility due to mixing and temperature difference.



(a)

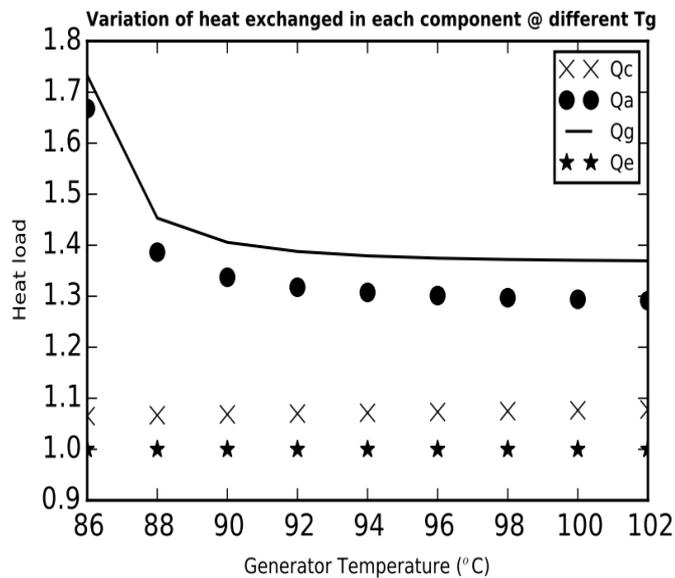


(b)

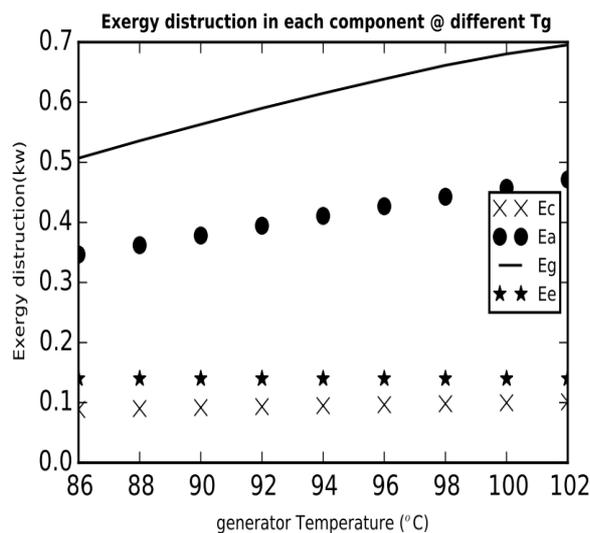
Figure 6. Effect of evaporator temperature on a) heat load b) exergy destruction

3.6 Heat Load and exergy destruction versus generator temperature

Figure 7 (a) shows that increasing generator temperature from 86 to 102°C makes the generator and absorber load decrease. As T_g increase, the concentration of rich solution increase and circulation ratio decreases too. This makes, with a constant mass flow rate a large amount of heat will be transferred. But, Fig. 7 (b) depicts that whenever T_g increases exergy destruction in absorber and generator will increase. This is because of the increase in irreversibility due to mixing and temperature difference.



(a)



(b)

Figure 7. Effect of generator temperature on a) heat load b) exergy destruction

4. Conclusion

From the results, the following main points are concluded: Increasing the generator temperature have a positive effect on COP of VARS, and a positive effect on exergetic efficiency until the system reach its optimum working temperature of T_g . So, for a good design of VARS engineers must conduct both COP and COP_{ex} of the system. Increasing the temperature of the evaporator (T_e) leads to a decrease in the required optimum temperature of the generator (decrease the amount of heat required on the generator side) cause to have a positive effect for both COP and COP_{ex} . Increasing the absorber temperature will lower both the COP and COP_{ex} of the system. This is due to as absorber temperature increases, the amount of degassing in the generator will decrease while the circulation ratio rises significantly. The circulation ratio depends on the generator temperature. Whenever the T_g reaches its cut-off temperature the circulation ratio will rise. This is due to the low vapor distilled from the generator. Far from the cut of temperature increasing generator temperature have a negligible impact on the circulation ratio.

References

x	mass fraction	-
f	circulation ratio	-
D	mass flow rate of distilled vapor	$kgsec^{-1}$
m	mass flow rate	$kgsec^{-1}$
M	molar mass	$kgmol^{-1}$
y	mole fraction	-
Q	heat load	$Jsec^{-1}$
T	temperature	K
s	Entropy	$Jsec^{-1}K^{-1}$
v	dissociation number	-
R	Universal gas constant	$Jmol^{-1}K^{-1}$
P	pressure	$kgm^{-3}sec^{-2}$

Subscripts

o	reference state
∞	ambient condition

References

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