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Irreversibility Analysis of Double Effect LiBr-Water Vapor Absorption Chiller

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Abstract

Irreversibility analysis of a double effect direct fired LiBr-water vapor absorption chiller with reverse parallel flow arrangement is presented in this research. The irreversibility method is a powerful tool for optimization of the performance of absorption refrigeration systems. Analysis is performed for 4 different operating conditions of the chiller. Mass flow rates, specific enthalpies, specific entropies, energy balance and irreversibility changes were calculated for all components of the system. The results reveal that irreversibility changes in condenser, evaporator and heat exchangers are less as compared to the irreversibility changes in generators and absorber. The results show that COP of the absorption chiller increases by increasing the temperature of high temperature generator. However, irreversibility changes also increase with this increase in temperature thus reducing the availability of energy. Thermo economic optimization of LiBr-water vapor absorption chiller can be done by using these results. These results play an important role for improvement of absorption system. It is observed that analysis of absorption system using the irreversibility method can improve potential thermodynamic performance, which is impossible through numerical techniques.

Keywords: Irreversibility, optimization, double effect Li-Br Water vapor absorption chiller, availability, generator, absorber.

1. Literature Review

A number of researchers have performed exergy / irreversibility analysis different for components/configurations of vapor absorption refrigeration systems. Bejan [1] performed a theoretical analysis based on entropy generation minimization. The method can be used for the thermodynamic optimization of refrigeration plants. Kotas [2] performed the analysis of thermal & chemical plants using exergy analysis method but the exergy analysis of absorption systems is not included. Szargut [3] presented energy and exergy balance of an NH₃-H₂O absorption refrigerator. Lee and Sherif [4] performed 1st & 2nd law analysis of absorption systems for both cooling and heating applications.

O. Kaynakli [5] studied the irreversibility in a coil absorber using LiBr-water solution. The irreversibility analysis of only coil absorber of vapor absorption cycle & this it is not performed for the other components of vapor absorption refrigeration cycle. M.M. Talbi [6] carried out an exergy analysis for a single-effect absorption refrigeration cycle with LiBr-water as the working fluid pair. Also this analysis was not performed for double effect absorption refrigeration cycle. Arzu Sencana [7] also performed exergy analysis of a single-effect LiBr-water absorption system for cooling & heating applications. It was performed for single effect LiBr-water absorption system. Abdul

Khaliq [8] performed energy & exergy analysis of double effect vapor absorption refrigeration system. They proved that exergy destructions were significantly larger in generators, absorbers, evaporator-2 and heat exchangers as compared to those in condenser-1, evaporator-1, throttling valves, and expansion valves. They further proved that with an increase in the generator-1 temperature the COP and Exergetic COP (ECOP) increased. On the other hand, the COP and Exergetic COP (ECOP) decreased with an increase in the absorber-1 temperature. But they did not perform this analysis for standard / commercial configuration of double effect direct fired reverse parallel flow vapor absorption refrigeration system rather they performed analysis with new configuration using two evaporators, two condensers and two absorbers. The system configuration was complex. Horuz and T.M.S. Callander [9] carried out an experimental investigation of the performance of commercially available gas-fired vapor absorption refrigeration system using NH₃water solution with NH₃ as the refrigerant and water as the absorbent and had a rated cooling capacity of 10 kW. The results of the system with variations in chilled water inlet temperature, chilled water level in the evaporator drum, chilled water flow rate and variable heat input were presented. D. K. Anand and B. Kumar [10] performed the calculation of LiBr-water solution properties over the complete range of useful temperatures and concentrations applicable to the absorption cycle. They made the charts for enthalpies and entropies of LiBr-water solution and calculated individual irreversibility for all the components of LiBr-water vapor absorption cycles, which is very useful information for irreversibility analysis but they did not perform any kind of irreversibility analysis for double effect

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LiBr water absorption cycle rather they provided base information for such analysis.

In this thesis, irreversibility analysis is performed for standard configuration of the direct fired, double effect, reverse parallel flow, vapor absorption chiller using aqueous LiBr as a working fluid which is being used by today's practical and commercial chillers at large scale. The irreversibility change at each component is determined. The irreversibility is equivalent to the exergy destroyed. All exergy losses existing in LiBr-water vapor absorption system are estimated. Irreversibility analysis is carried out for different operating conditions of the chiller. It is proved that by increasing temperature of Generator1, COP of chiller increases. It is also proved that increasing inlet temperature of chilled water, COP of chiller increases.

2. Irreversibility Analysis

System description

Analysis is performed for four different operating conditions of the chiller. For a double-effect, direct-fired, reverse parallel flow absorption chiller, the absorption cycle shows the properties of the solution at various concentration, temperatures, & pressures, as in Fig. 1.



Fig. 1. Double effect direct fired LiBr-water vapor absorption chiller with reverse parallel flow arrangement

For a typical double-effect, direct-fired, reverse parallel flow absorption chiller operating at a certain load (Fig 1), in the high-temperature generator (G1), the diluted LiBr solution is heated by the direct fired burner and the water is vaporized from the solution. As the water vapor is boiled off, LiBr solution becomes concentrated in the high temperature generator (G1). The concentrated solution leaves G1 & enters the high temperature heat exchanger (Hx-1) at temperature T₇ with concentration X₇ (state point 7). The concentrated solution cools to temperature T₈ as it leaves the heat exchanger (Hx-1). The water vapor that boils off from the LiBr solution leaves the high-temperature generator (G1) at temperature T₉ and flows through the tubes submerged in the diluted solution in the low-temperature generator (G2), the water vapor condenses to liquid form.

Latent heat of condensation released from the liquid refrigerant heats the diluted solution in G2 from temperature T_2 to temperature T_3 and boils off the water vapor from the diluted solution. Finally, in the low-temperature generator (G2), the diluted solution becomes partly concentrated and splits into two streams with same concentrations x_3 and x_{14}

when it leaves at the outlet. The stream of partly concentrated LiBr solution from low-temperature generator (G2) with concentration x_{14} combines with the concentrated solution from the high-temperature heat exchanger (Hx-1) and the mixture enters the low temperature heat exchanger (Hx-2) at state point 4 with temperature T_4 and concentration x₄. After it flows through the low-temperature heat exchanger (Hx-2), the temperature of the solution drops to T_5 . The solution then enters the absorber, sprays over the cooling water tube bundle and absorbs the water vapor from the evaporator. The condensed liquid refrigerant that flows inside the tubes in the low-temperature generator (G2) enters the trough of the condenser at a temperature T_{15} (state point 15). Water vapor boiled off in the low-temperature generator (G2) at temperature T_{10} is extracted by the condenser's low saturated pressure through the top passage and condenses to liquid as the cooling water removes the latent heat of condensation and drops the temperature to T_{11} . In the condenser, cooling water enters at temperature T_i (state point j) and leaves at Tk (state point k). The condensed liquid refrigerant from the tubes is cooled to a temperature T₁₁ and combines with the liquid refrigerant (from G2) condensed in the condenser. Liquid refrigerant at temperature T₁₁ passes through an expansion valve where it expands to very low pressure and temperature T_{12} (state point 12). Typically a vacuum of 6-8 mm Hg abs is maintained inside the evaporator in order to attain very low temperature of about 5-7°C. Evaporator spray pump (P4) is used to circulate the refrigerant to increase the evaporation rate. Chilled water enters the evaporator inside the tube bundle at temperature T_a (state point a) and leaves at temperature T_b (state point b). Heat is transferred from the chilled water to the refrigerant at an amount equal to the latent heat of vaporization and the refrigerant becomes saturated vapor at temperature T₁₃ (state point 13). Vaporized water in the evaporator is then extracted to the absorber because of its lower vapor pressure and is absorbed by the concentrated LiBr solution. In the absorber, concentrated solution is supplied and sprayed at temperature T_5 with a concentration x_5 (state point 5) by absorber spray pump (P2). It is cooled by the cooling water flowing inside the tube bundle at an entering temperature T_c (state point c) and leaving temperature T_d (state point d). As the water vapor from the evaporator is absorbed, the solution becomes diluted to a concentration of x_1 and its temperature drops to T_1 (state point 1). The cooling water removes the heat of absorption in the absorber. Diluted solution is then pumped to the low-temperature heat exchanger (Hx-2) and the low-temperature generator (G2) by a low temperature generator pump (P1). In the lowtemperature heat exchanger (Hx-2), diluted solution enters at temperature T_1 and leaves at T_2 (state point 2). After the diluted solution enters the low-temperature generator (G2), because of the release of the latent heat of condensation from the condensed water vapor that flows inside the tube bundle of the low-temperature generator (G2), the diluted solution is first heated and the water vapor then boils off. As the water vapor is boiled off, the LiBr solution becomes partly concentrated. At the outlet of the low-temperature generator (G2), partly concentrated LiBr solution is divided into two streams with equal concentrations x_3 and x_{14} . One stream (state point 3) flows through the high-temperature heat exchanger (Hx-1) and then enters the high-temperature generator (G1) through a high-temperature generator pump (P3).

The other stream (state point 14) combines with the concentrated solution from the high-temperature heat

exchanger (Hx-1) and enters the low-temperature heat exchanger (Hx-2) at state point 4. In the high-temperature heat exchanger (Hx-1), partly concentrated solution enters at temperature T_3 and leaves at T_6 . After that, partly concentrated solution flows into the high temperature generator (G1) and cycle is repeated.

3. Assumptions used in the irreversibility analysis

i. Refrigerant leaving the condenser and evaporator are at saturated state.

ii. The pressure loss in the lines is negligible.

iii. Steady state conditions are assumed, there is no loss of heat during each thermodynamic process involved.

iv. The atmospheric condition is assumed at 25 °C and 1 bar. v. Lithium bromide solution is assumed to be in equilibrium with the reference pressure and temperatures.

vi. Mass flow rates of refrigerant vapor boiled in the high-temperature generator (G1) and low-temperature generator (G2) are assumed to be equal.

vii. Mass flow rates of partially concentrated solution at state point 3 and 14 leaving the low-temperature generator (G2) are assumed to be equal.

viii. The work input required for the refrigerant pump (P4) and solution pumps (P1, P2 and P3) is negligible as compared to heat input in high temperature generator G1.

ix. Heat loss across the pumps (P1, P2, P3 & P4) is negligible.

x. Change in kinetic energy and potential energy is assumed to be zero.

xi. Enthalpies and entropies of solution/ refrigerant entering a section are taken as negative and leaving a section is taken as positive.

4. Irreversibility analysis

For the irreversibility calculation at each component of the absorption system, it is necessary, first to make the mathematical models and equations which govern the fundamental processes. The irreversibility (also stated as opposite to exergy) is the amount of exergy that is lost to the surroundings and which cannot be used anywhere. According to the gouy-stodola theorem, the irreversibility rate can be defined as

$$\dot{I} = T_0 S_{gen} \tag{1}$$

General form of steady state steady flow energy equation is given as

$$\dot{Q} - \dot{W} = \dot{m}\Delta(h + \frac{1}{2}V^2 + gz$$
 (2)

Steady state steady flow energy equation with negligible work transfer, kinetic energy and potential energy is given as

$$\dot{\mathbf{Q}} = \dot{\mathbf{m}}(\mathbf{h}_{0} \cdot \mathbf{h}_{i}) \tag{3}$$

To determine the status of the solution in the absorption cycle, the state points of the cycle are defined as follows:

1. Mean condition of the diluted solution leaving the absorber (A).

- 2. Dilute solution leaving the low-temperature heat exchanger (Hx-2).
- 3. Partly concentrated solution leaving the low-temperature generator (G2) and entering the high-temperature heat exchanger (Hx-1).
- 4. The condition of the mixture of partly concentrated and concentrated solutions entering the low-temperature heat exchanger (Hx-2).
- 5. Concentrated solution entering the absorber.
- 6. Partly concentrated solution leaving high-temperature heat exchanger (Hx-1).
- 7. Concentrated solution leaving high-temperature generator (G1).
- 8. Concentrated solution leaving high-temperature heat exchanger (Hx-1).
- 9. Water vapor leaving high-temperature generator (G1).
- 10. Water vapor leaving low-temperature generator (G2).
- 11. Low temperature condensed water leaving condenser.
- 12. Low temperature, low pressure wet water vapor entering the evaporator.
- 13. Low temperature, low pressure dry water vapor leaving the evaporator.
- 14. Partly concentrated solution leaving the low-temperature generator (G2).
- 15. High temperature condensed water leaving low-temperature generator (G2).
 - a. Chilled water coming in the evaporator.
 - b. Chilled water going out from the evaporator.
 - c. Cooling water coming in the absorber.
 - d. Cooling water going out from the absorber.
 - j. Cooling water coming in the condenser.
 - k. Cooling water going out from the condenser.
 - p. Inlet condition of natural gas.
 - q. Outlet condition of natural gas.

Each component of the absorption refrigeration system shown in the Fig 1 is modeled through the set of parameters, which characterize the mass, heat and irreversibility equations. Specific enthalpies (h), specific entropies (s) and LiBr-water solution concentration (x) of state points (1 to 15 and a, b, c, d, j, k, p, q) are found out from h-t-x graph (duhring chart), h-t-x, s-t-x and hydrocarbon properties/ steam Tables. First of all heat source input temperature is fixed. At this temperature, LiBr-water solution temperature and concentration are assumed in high temperature generator. After that all the values of temperature and concentration across each component are determined by iteration method and each component is balanced thermodynamically.

5. High-temperature generator (G1)

Mass balance

It is assumed that mass flow rate of refrigerant (water vapor) leaving high temperature generator is equal to the mass flow rate of refrigerant (water vapor) leaving low temperature generator G2 (which is half of the mass flow rate of the refrigerant entering the evaporator at state point 12).

$$\dot{m}_9 = \dot{m}_{10} = 0.5 \times \dot{m}_{12} \tag{4}$$

$$\dot{m}_6 = \dot{m}_7 + \dot{m}_9 \,. \tag{5}$$

Energy balance

(10)

$$Q_{61} = \dot{m}_7 h_7 + \dot{m}_9 h_9 - \dot{m}_6 h_6 \tag{6}$$

$$Q_{G1} = \dot{m}_g \times LHV_g \tag{7}$$

Where

 \dot{m}_a = Mass flow rate of the gas to the burner of high temperature generator.

$$LHV_g = Q_{G1}/\dot{m}_g \tag{8}$$

Irreversibility

$$\Delta I_{G1} = T_0[(\dot{m}_7 \ s_7 + \dot{m}_9 \ s_9 - \dot{m}_6 s_6 + \dot{m}_g (s_q - s_p)] \tag{9}$$

Low-temperature generator (G2)

Mass balance $\dot{m}_2 = \dot{m}_3 + \dot{m}_{10} + \dot{m}_{14}$

$$\dot{m}_3 = \dot{m}_{14} = 0.5 \times (\dot{m}_2 - \dot{m}_{10}) \tag{11}$$

 $\dot{m}_{10} = 0.5 \times \dot{m}_{12}$ (12)

$$\dot{m}_9 = \dot{m}_{10} = \dot{m}_{15} \tag{13}$$

Energy balance

 $Q_{G2} = \dot{m}_3 h_3 + \dot{m}_{10} h_{10} + \dot{m}_{14} h_{14} - \dot{m}_2 h_2$ (14)

Irreversibility

 $\Delta I_{G2} = T_0 [(\dot{m}_3 s_3 + \dot{m}_{10} s_{10} + \dot{m}_{14} s_{14} - \dot{m}_2 s_2 + \dot{m}_9 (s_{15}$ *s*₉)] (15)

Absorber (A)

Mass balance $\dot{m}_1 = \dot{m}_5 + \dot{m}_{13}$ (16)

Energy balance

 $Q_{abs} = \dot{m}_1 h_1 - \dot{m}_5 h_5 - \dot{m}_{13} h_{13}$ (17)Irreversibility

 $\Delta I_{abs} = T_0 [(\dot{m}_1 s_1 - \dot{m}_5 s_5 - \dot{m}_{13} s_{13} + \dot{m}_{cw} (s_d - s_c)] \quad (18)$

High temperature heat exchanger (Hx-1) Mass balance

 $\dot{m}_3 = \dot{m}_6$ (19)

 $\dot{m}_7 = \dot{m}_8$ (20)

 $\dot{m}_7(h_8 - h_7) + \dot{m}_3(h_6 - h_3) = 0$ (21)

Irreversibility $\Delta I_{Hx-1} = T_0 [\dot{m}_7 (s_8 - s_7) + \dot{m}_3 (s_6 - s_3)]$ (22)

Low temperature heat exchanger (Hx-2)

Mass balance

 $\dot{m}_4 = \dot{m}_5$

(24) $\dot{m}_{1} = \dot{m}_{2}$

Energy balance

$$\dot{m}_1(h_2 - h_1) + \dot{m}_4(h_5 - h_4) = 0 \tag{25}$$

Irreversibility

$$\Delta I_{Hx-2} = T_0 [\dot{m}_1 (s_2 - s_1) + \dot{m}_4 (s_5 - s_4)]$$
(26)

Condenser (C) Mass balance $\dot{m}_{11} = \dot{m}_{10} + \dot{m}_{15}$ (27)

Energy balance $Q_{cond} = \dot{m}_{11}h_{11} - \dot{m}_{10}h_{10} - \dot{m}_{15}h_{15}$ (28)

Irreversibility ΔI

$$\Delta I_{cond} = T_0 \Big[(\dot{m}_{11} s_{11} - \dot{m}_{10} s_{10} - \dot{m}_{15} s_{15}) + \dot{m}_{cw} (s_k - s_j) \Big]$$
(29)

Evaporator (E)

Mass balance $\dot{m}_{...} = \dot{m}$

$$m_{12} - m_{13}$$
 (50)

(20)

(37)

Energy balance $Q_{evap} = \dot{m}_{13}h_{13} - \dot{m}_{12}h_{12}$ (31)

Irreversibility $\Delta I_{evap} = T_0[(\dot{m}_{12} (s_{13} - s_{12}) + \dot{m}_{chw}(s_b - s_a)]$ (32)

Effectiveness of the heat exchangers 1 and 2

$$\varepsilon_1 = \frac{n_6 - n_3}{h_7 - h_3} \tag{33}$$

$$\epsilon_2 = \frac{h_2 - h_1}{h_4 - h_1} \tag{34}$$

Mass balance at the mixing point 4

 $\dot{m}_4 = \dot{m}_8 + \dot{m}_{14}$ (35)

Energy balance at the mixing point 4 $\dot{m}_4 h_4 = \dot{m}_8 h_8 + \dot{m}_{14} h_{14}$ (36)

Refrigerant mass flow through the evaporator From equation 4.30 $\dot{m}_{12} = \frac{\dot{Q}_{evap}}{h_{13} \cdot h_{12}}$

Where

 $h_{11} = h_{12}$ (before and after throttling)

Mass flow rate of the refrigerant entering the absorber A $(\dot{m}_{12} = \dot{m}_{13})$ is inversely proportional to the concentration of the solution at point 1 (x_1) leaving the absorber A.

Similarly mass flow rate of the solution entering the absorber (\dot{m}_5) is inversely proportional to difference of the concentration of the solution entering and leaving the absorber $(x_5 - x_1)$. Mathematically it can be presented by the following equation

$$\frac{\dot{m}_5}{\dot{m}_{12}} = \frac{x_1}{(x_5 - x_1)} \quad . \tag{38}$$

Mass flow rate of the concentrated solution entering the absorber (m_5) can be calculated from the equation 37.

Table 1.A. Double effect direct fired LiBr-water vapor absorption chiller

(Analysis 1)

(23)

State	Toursetout	C-lation	Mass flows m	Enthalpy	Entropy,
Point	°C	Concentration	(kg/sec) x 10 ⁻³	n (kJ/kg)	s (KJ/Kg K)
1	34.8	58	0.023417	92.75	2.2440
2	77.0	58	0.02341	181.5	2.5325
3	78.0	59.1	0.011331	185.2	2.4746
4	84.0	62	0.021906	214.1	2.4015
5	38.0	62	0.021906	118.9	2.1437

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6	121.5	59.1		0.011	331	269.2		2.6966
7	147.0	65		0.010	575	335.2		5.1000
8	90.0	65		0.010	575	245.2		2.3349
9	145.0	0		0.000	755	2769.	1	6.8729
10	78.0	0		0.000	755	2640.	4	7.6400
11	37.0	0		0.001	511	155.1		0.5537
12	6.3	0		0.001	511	155.6		0.5537
13	6.3	0		0.001	511	2511.	4	8.9925
14	78.0	59.1		0.011	331	185.1		2.4746
15	48.2	0		0.000	755	201.7		0.6800
System water properties								
State		Temperature,	Ent	halpies,	Entro	opies, s	Tem	perature
Points		°C	h k	J/kg	kJ/kg	g K.	Diffe	erence °C
a		11.0	46.	192	0.165	58		
b		6.3	26.	566	0.095	56	4.7	
c		30.8	129	.132	0.447	79		
d		33.5	140	.400	0.484	19	2.7	
J		33.5	140	.400	0.484	19		
k		35.5	148	.750	0.512	20	2.0	
Gas Properties								
Gas Inlet Pressure, H		External Temper	External Temperature		Natural Gas		Δs_g	
(bar) °		°C	°C		L.H.V (kJ/kg)		(kJ/	kg k)
(bar)		^o		===== (:			(
(bar) 1.20-1.25		150		45000			0.566	55

 Table 1.B.
 Double effect direct fired LiBr-water vapor absorption chiller

(Analysis 1)					
Mass flow re	sults				
Inputs					
Ref. Capacity	3.56	kW			
h ₁₃	2511.400	kJ/kg			
h ₁₁	155.002	kJ/kg			
X ₁	58.00	%			
X5	62.00	%			
Outputs					
m ₁₁ =	m : ₁₂	0.001511	kg/sec		
$\dot{m}_4 =$	m [·] ₅	0.021906	kg/sec		
$\dot{m}_{1} =$	m ₂	0.023417	kg/sec		
m' ₃ =	m ₆	0.011331	kg/sec		
m . 14		0.011331	kg/sec		
$m'_{7} =$	m' ₈	0.010575	kg/sec		
m ₃		0.001511	kg/sec		
m' ₉ =	m . 15	0.000755	kg/sec		
m . 10		0.000755	kg/sec		

Table 1.C. Energy and irreversibility analysis of double

 effect LiBr-water vapor absorption chiller (Analysis 1)

System energy balancing				the system		
Components	Internal Heats kW	Components	Internal Heats kW	Components	Internal Heats kW	
Qgen-1	2.587	Qgen-1	2.587	Qgen-1	2.587	
Qgen-2	1.939	Qgen-2	1.939	Qgen-2	1.939	
Qabsorber	4.228	Qabsorber	4.228	Qabsorber	4.228	
Qcond	1.913	Qcond	1.913	Qcond	1.913	
Qevap	3.560	Qevap	3.560	Qevap	3.560	
Hex-1	0.952	Hex-1	0.952	Hex-1	0.952	
Hex-2	2.085	Hex-2	2.085	Hex-2	2.085	

 $COP = Qevap/Qgen-1 \times \eta comb = 1.170$

m̀ _g	0.057×10^{-3}	Q _{rej}	6.140
\dot{m}_{chw}	181.000x10 ⁻³	Q _{ct}	7.361
m _{cw}	375.000x10 ⁻³	ΔT_{chw}	4.700
$\dot{m}_{cw}/\dot{m}_{chw}$	2.070	ΔT_{ct}	4.700
η_{comb}	0.85		



Fig. 2. Generator (G1) temerature versus COP







Fig. 4. Generator (G1)temperature & Irreversibility changes



Fig. 5. Gen. 1 temp versus irreversibility change in G1



Fig. 6. Gen. 1 temp versus irreversibility change in G2



Fig. 7. Gen. 1 temp versus Irreversibility change in absorber



Fig. 8. Gen. 1 temp versus irreversibility change in Condenser



Fig. 9. Gen. 1 temp versus irreversibility change in Evaporator



Fig. 10. Gen. 1 temp versus irreversibility change in high temp heat exchanger (Hex-1)



Fig. 11. Gen. 1 temp versus irreversibility change in low temp heat exchanger (Hex-2)

6. Imperfect mass and heat transfer

At higher temperature in high temperature generator (G1), water is boiled off at higher temperature and vapor becomes superheated. This high temperature, high pressure vapors travel from high temperature generator (G1) to low temperature generator (G2), dissipates its heat to partially concentrated solution in lower temperature generator (G2) and is converted to liquid form. Perfect heat transfer cannot take place and some of heat loss occurs during heat transfer process. This heat loss causes irreversibility. Similarly in solution heat exchangers, high temperature stream gives its heat to low temperature stream. But the heat lost by high temperature stream is never equal to heat gain by low temperature stream, it means some of heat is lost during heat transfer process and causes irreversibility. At increased operating temperatures, heat transfer rate also increases and the heat transfer irreversibility will be increased.

When chiller is operating at part load, then temperature in high temperature generator G1 is low at same pressure, therefore, amount of water vapor boiled off is less as compared to that at full load (high temperature). Thus at low temperature mass flow of water vapor leaving from HTG is reduced. Consequently the heat transfer to partially concentrated solution will be reduced and solution returning from HTG will be less concentrated (containing more water). This is due to imperfect mass transfer and causes irreversibility, because availability of useful energy will be reduced.

7. Losses due to mixing

Losses due to mixing are because of evaporation of the refrigerant from a concentrated solution in the generator which requires more heat to evaporate than that required for pure state. The refrigerant leaves the generator in the state of superheated vapor because the temperature required for the generator is much higher than the evaporation temperature of pure refrigerant at the same pressure. The energy spent in the generator in the form of superheat, causes a thermodynamic loss and extra cooling in the condenser is required. Therefore, as evident from the Table 1C, 2C, 3C & 4C, the generator and absorber have the maximum changes in irreversibility. Condenser and evaporator have less irreversibility due to the pure fluid being used by them. The irreversibility change in the evaporator is due to the temperature difference between the evaporating refrigerant and the environment.

8. Losses due to circulation

Losses depend upon solution circulation ratio. When solution circulation ratio increases the losses also increase and with decrease in solution circulation ratio, losses are decreased. For example, irreversibility in absorber can be decreased by reducing circulation ratio of the solution and using more efficient heat exchanger. Circulation ratio of solution can be decreased by increasing the solution concentration in the generator when the temperature or pressure decreases and vice versa. When the circulation ratio of solution is decreased, heat output at the absorber and heat input to the generator are increased and COP will be improved. When circulation ratio of the solution is increased, heat output at the absorber and the heat input to the generator are decreased and COP will be reduced. $\hat{\mathsf{F}}\mathsf{i}\mathsf{gure}$ 2 shows the variation of the COP of the absorption chiller with the heat source temperature supplied to the high temperature generator. It can be seen that the COP of the system increases with increase in the heat source temperature. The absorption system's performance is greatly affected by the operating temperatures. The COP will increase with increasing heat source temperature. But at the same time, the heat transfer will increase in all the heat exchangers due to this increase in heat source temperature Thus irreversibility will be increased at increased heat transfer and hence increase of COP will be smaller. Figure 3 shows the change in COP of the absorption chiller with the inlet temperature of chilled water. COP of the chiller increases when inlet temperature of chilled water increases. With increase in inlet temperature of chilled water, higher cooling capacity can be produced from the evaporator. COP will increase with increase in inlet temperature of chilled water but from this analysis, it is proved that the increase in COP is relatively small. The reason for this behavior is the effect of other variables of the system such as high temperature generator G1 temperature, absolute pressure in the evaporator and temperature of cooling water entering the absorber. As chilled water inlet temperature increases, the pressure in the evaporator also increases thus causing increase in the heat transfer slightly. The increase of the pressure in evaporator and absorber results the decrease in the mass fractions slightly.

Figure 4 shows that irreversibility changes in the high temperature generator and absorber are significantly more than the irreversibility changes in condenser and evaporator. This is due to the heat of mixing in the high temperature generator and absorber which is not present in pure fluids in condenser and evaporator. Figure 5 shows that irreversibility in high temperature generator decreases with increase in heat source temperature (analysis 2) but with further increase in source temperature, there is remarkable increase in irreversibility (analysis 3 and 4). This is due to larger heat transfer rates at higher temperature thus causing the increased heat transfer irreversibility. Figure 6 shows that irreversibility change in low temperature generator is small for two practical readings of the chiller. Irreversibility change for analysis 3 is considerable but with further increase in heat source temperature, irreversibility has decreased. The reason for this reduction in irreversibility is that, high temperature, high pressure superheated vapor dissipates its heat effectively to the solution in low temperature generator (G2) and completely changes to liquid, thus increasing COP. Figure 7 shows that irreversibility in absorber is significant. It is due to the fact that larger entropy generation occurs due to mixing of two fluids in the absorber. It can also be observed that with increase of source temperature (at 147°C, 149°C, 151.67°C), irreversibility increases (analysis 1-3) but there is a decrease in the irreversibility at higher temperature (153.5°C) for the analysis 4. The reason for this decrease is that, there is considerable increase in solution concentration in the high temperature generator for analysis 4. Solution circulation ratio can be reduced by increasing the concentration of solution in the generator. Figure 8 shows that irreversibility in condenser is less as compared to those in generators and absorber due to pure fluid being used in condenser. Their general trend is to decrease with increase in heat source temperature. Figure 9 shows that irreversibility in evaporator is less as compared to those in generators and absorber due to pure fluid being used in evaporator. Irreversibility in the evaporator decreases with increase in heat source temperature and becomes minimum at G1 temperature of 151.67°C (analysis 3), which is very near to design values of typical double effect vapor absorption chiller. For further increase in temperature (analysis 4) irreversibility in the evaporator has been increased. It means that G1 temperature of 151.67°C is the ideal for the irreversibility in the evaporator. Irreversibility will increase if we go above or below to this design value of G1 temperature. Figures 10 and 11 show that irreversibility changes in high temperature heat exchanger are negative because high temperature stream rejects heat which results in entropy decrease and low temperature stream gains heat which results in entropy increase. But heat gain by low temperature stream is always less than heat rejected by high temperature stream (second law of thermodynamics). Therefore, decrease in entropy of high temperature stream is more than increase in entropy of low temperature stream causing net change in irreversibility of heat exchanger as negative. With increase of heat exchanger effectiveness, irreversibility will be reduced and therefore change in irreversibility will also be reduced. High temperature heat exchanger irreversibility is due to large temperature difference between the two streams of fluid which results in higher heat transfer irreversibility. In high temperature heat exchanger, finite temperature difference between two streams is more as compared to that in low temperature heat exchanger. Therefore, finite time heat transfer irreversibility will be higher in high temperature

heat exchanger as compared to that in low temperature heat exchanger. With increase in heat source temperature, heat transfer in both heat exchanger increases. This increased heat transfer results in increased heat transfer irreversibility.

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