

**ABSTRACT**

This work presents exergy analysis of a double effect parallel flow, single effect and half effect absorption systems for comparison. A computer program is developed for the thermodynamic properties of lithium bromide-water solutions by the author in FORTRAN codes for the exergy analysis. The coefficient performance (COP) and the exergetic coefficient performance (ECOP) of the double effect parallel flow absorption systems are higher than the single effect and the half effect cycles. For the double effect cycle COP and ECOP are found as 1.196 and 0.284, and for the single effect cycle COP and ECOP are found as 0.68 and 0.254, respectively. For the half effect cycle COP and ECOP are found as 0.455 and 0.24, respectively. For each component the exergy loss is calculated. Most of the irreversibilities are found in the evaporator and in the absorber. It is concluded that the performance of the evaporator and the absorber is crucial for the three cycles. Better design and improving of these two components will directly improve and affect positively the performance and the working conditions of the three cycles.

**KEYWORDS:** Cooling, Absorption, Half-Single-Double effect, ECOP.

**Nomenclature**

$COP$	coefficient of performance
$e$	specific exergy (kJ/kg)
$\dot{E}$	exergy flow rate (kW)
$h$	specific enthalpy (kJ/kg), (kJ/kMol)
$\dot{m}$	mass flow rate (kg/s)
$P$	pressure (kPa)
$\dot{Q}$	heat flow rate (kW)
$s$	specific entropy (kJ/kg K)
$T$	temperature (K)
$\dot{W}$	power (kW)

**Greek letters**

$\eta$  efficiency

**Subscripts**

$A$	absorber
$C$	condenser
$D$	destruction
$en$	energy
$ex$	exergy
$E$	evaporator
$EXV$	expansion valve
$HE$	heat exchanger
$HPG$	high pressure generator
$L$	loss
$LPG$	low temperature generator
$OC$	overall cycle

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P pump  
0 environment conditions

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## INTRODUCTION

In industrial process a lot of thermal energy is obtained by burning fossil fuel to produce heat for their purpose and after this processes heat is rejected as waste. This waste heat can be converted to a useful cooling by using a heat operated refrigeration system, such as an absorption refrigeration cycle. The growing need for building cooling and refrigeration in industry all over the world makes the absorption cooling cycles driven with low temperature heat energy interesting. The main causes of this growing need are the increasing requirements for higher living standards, comfort and the increasing thermal load of buildings. The absorption cooling cycle is an essential component of the combined heating, cooling, and power systems. The absorption cooling is a kind of energy saving and environmental friendly technology and absorption cooling technology is attracting more and more attention in our age. The power consumption for cooling can be reduced by using low temperature waste heat, geothermal or solar energy. Absorption systems seem to provide many advantages, but vapor compression systems still dominate all market sectors.

There are about 40 refrigerant compounds and about 200 absorbent compounds available in absorption cooling that the most common working fluids are water/NH<sub>3</sub> and LiBr/water. The absorption chillers using LiBr-H<sub>2</sub>O solution offer very good efficiency than the other solutions for cooling and refrigeration at over 0 °C temperature of evaporator. But there is a risk of salt crystal formation called solution crystallization that happens in the case of high absorber temperature, air leak into machine, or low ambient temperature. The ammonia-lithium nitrate, the ammonia-water solution, or other appropriate solutions can be used better for producing cold at temperatures below than 0 °C. The COP of an absorption cycle depends on three external temperatures; evaporation, ambient, and generation (driving) temperatures. The triple effect cycle has the best COP among the double effect, the single effect, and the half effect cycles. However, the double effect cycle has better COP than the single effect and the half effect cycles. The half effect cycles has the lowest COP and the single effect cycle presents better COP than the half one. The details of these cycles, their configuration and their differences can be found in literature. The single and the double effect absorption cycles have more commercial use than the triple one and the half one [1, 2]. The air cooled double effect systems are better than the single effect and the half effect cycles, because they are more flexible, efficient, independence upon water and without cooling tower. In the double effect cycle the same driving heat source produces refrigerant vapor twice that in these process two vapor generators are needed. The most common configurations between in lots of configuration of double effect cycles are in-parallel and in series cycle layouts. In series means that the entire flow goes through both generators without dividing into two streams. In parallel cycle the solution stream split among high and low pressure generator. In-series layouts are better in the cooling capacity than the in-parallel layouts however the in-parallel layouts have higher COP [2]. The double effect cycles needs heat energy at between 120-160 °C temperatures, and the single effect cycle needs heat energy at between 90-120 °C temperatures, while the half effect cycles needs heat energy at between 50-90 °C temperatures, [1, 2].

Farshi *et al.*, studied on exergo-economic analysis of double effect absorption refrigeration systems, and they found that lower total investment costs were obtained when the condenser temperatures were low and the evaporator temperatures were high [3]. Inzunza *et al.*, have done the comparison of the performance of single-effect, half-effect, double-effect in series and inverse absorption cooling systems operating with the mixture H<sub>2</sub>O/LiBr. They found that for the generation temperature between 100 °C and 110 °C, the COP of the single effect was up to 0.89, for the generation temperature of over 55 °C the COP of the half effect was up to 0.44. They also found that the most efficient one is the double effect systems, which the COP is up to 1.48. They observed that for low temperatures the half effect systems work better than any other [4]. Inzunza *et al.*, also studied the comparison of the performance of single-effect, half-effect, double-effect in series and inverse and triple-effect absorption cooling systems operating with the NH<sub>3</sub>-LiNO<sub>3</sub> mixture. They obtained that the COP values of H<sub>2</sub>O/LiBr are higher than the COP values of NH<sub>3</sub>-LiNO<sub>3</sub>. However with NH<sub>3</sub>-LiNO<sub>3</sub> refrigeration solution, the evaporator temperature can be as low as -50 °C [5]. Talukdar and Gogoi have done the exergy analysis of a combined vapor power cycle and boiler flue gas driven double effect water-LiBr absorption refrigeration system. They concluded that for integration with power cycle, the double effect absorption system is more appropriate and better than the single effect [6].

Colorado and Rivera have obtained the performance comparison between a conventional vapor compression and compression-absorption single-stage and double-stage systems used for refrigeration [7]. They concluded that the

compression power of the cascade cycles was 45 % lower than in compression cycles. Kaynakli *et al.*, have done the energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources; they have reported that higher temperatures of the heat sources increases the exergy destruction of the high pressure generator [8]. Avanesian and Ameri have done the energy, exergy, and economic analysis of single and double effect LiBr–H<sub>2</sub>O absorption chillers and they showed that the double effect absorption chillers are more economical than the single effect [9]. Bouaziz and Lounissi in their study named the energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling, they found that the COP of the proposed system is better than the conventional one [10].

The goal of this study is to investigate and compare the irreversibility and the exergetic coefficient of performance (ECOP) of the half effect, the single effect and the double effect absorption systems. The three cycles and their working conditions are taken from the reference [1, 11].

**MATERIALS AND METHODS**

The schematic diagram of the double effect absorption cooling cycle, the single effect absorption cycle and the half effect absorption cycle are given in Figure 1, 2 and 3. A half effect absorption system consists of two absorbers, a condenser, two generators, an evaporator, two heat exchangers, two pumps and three expansion valves. A single effect absorption system consists of an absorber, a condenser, a generator, an evaporator, a heat exchanger, a pump and two expansion valves. A double effect absorption system consists of an absorber, two condensers, two generators, an evaporator, two heat exchangers, two pumps and four expansion valves. At the double effect cycle, the solution that is pumped from the pump1 is heated in the heat exchanger2 and firstly enters the low pressure generator which is heated by the condanser1, after that the liquid solution is pumped with pump2 to the heat exchanger1 and then enters the high pressure generator. The vapor taken from the high pressure generator, condensates in the condanser1 and some of the heat energy is transferred into the low pressure generator. The liquid enters condenser2 and mixes with the vapor coming from the low pressure generator. After that the liquid transferred from the expansion valve4 evaporates in the evaporator to obtain cooling.

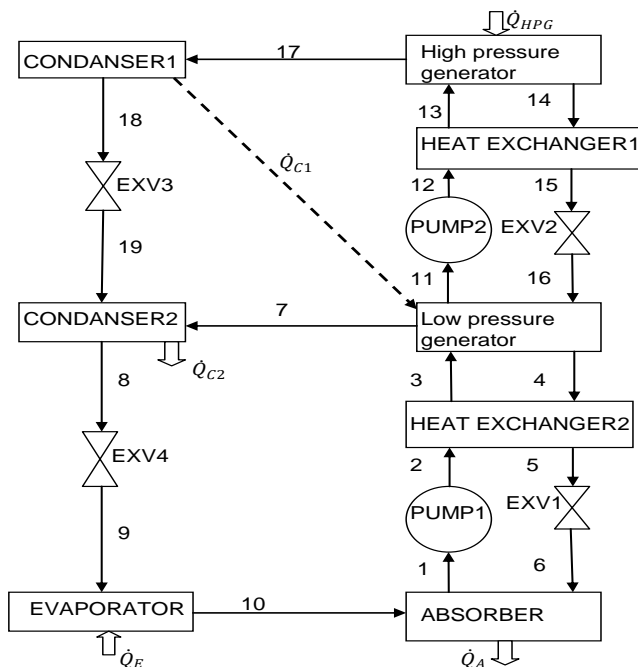


Figure 1. Double effect absorption cooling cycle.

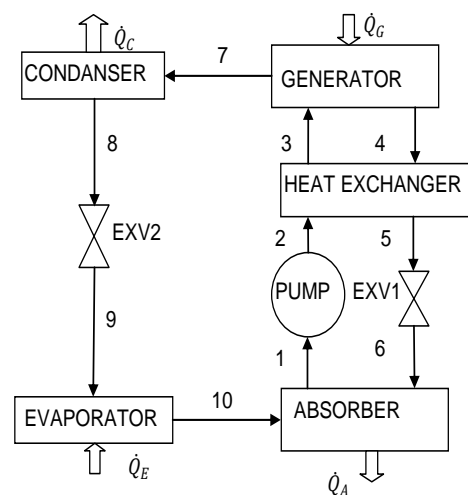


Figure 2. Single effect absorption cooling cycle.

The single effect cycle has a refrigerant cycle (7-10) and H<sub>2</sub>O-LiBr solution cycle (1-6). The generator is supplied with a heat source. The evaporated H<sub>2</sub>O is conducted to the condenser that gives heat to the atmosphere to change the phase of H<sub>2</sub>O from vapor to liquid. The refrigerant H<sub>2</sub>O is expanded in an expansion valve to reach the evaporation pressure. The cooling process is obtained in the evaporator and the refrigerant evaporates again and

then is conducted to the absorber. The vapor is mixes with H<sub>2</sub>O-LiBr solution coming from the generator and the absorber releases heat. After that the weak H<sub>2</sub>O-LiBr solution is pumped to the generator by passing through the heat exchanger which increases solution temperature. The cycle starts once again in the generator.

The half effect absorption system has two H<sub>2</sub>O-LiBr solution circuits. The evaporated H<sub>2</sub>O is produced in the generator2. The condensed H<sub>2</sub>O in the condenser is expanded in the expansion valve3 and then evaporated in the evaporator. The evaporated refrigerant in the evaporator is conducted to the absorber1. The vapor mixes with H<sub>2</sub>O-LiBr solution coming from the generator1. And then the weak H<sub>2</sub>O-LiBr solution is pumped to the low pressure generator1 by passing through the heat exchanger1 which increases solution temperature. Some part of H<sub>2</sub>O evaporates in the low pressure generator1 and goes directly to high pressure absorber2. At the high pressure absorber2 low concentrated H<sub>2</sub>O-LiBr is formed and pumped to the high pressure generator2 by passing through the heat exchanger2. The cycle starts once again in the generator2.

In this study, the thermodynamic and the mathematical modeling are explained as follows for the three cycles and the thermodynamic analysis is done. In this study, these assumptions are utilized in the analysis of the three cycles: The pressure drops in the pipeline and in the components are neglected, the cycles are at steady state and steady flow cycles, refrigerant leaving the condenser is saturated liquid at condenser pressure, the pump process is adiabatic, the pressure reducing valve is an adiabatic process, refrigerant leaving the evaporator is saturated vapor at evaporator pressure, solution leaving the generators and the absorber are assumed to be saturated in equilibrium conditions at its respective temperature and pressure, refrigerant is pure water, direct heat transfer from the components to the surroundings is negligible. There is no mass inlet or outlet of the cycles so that the chemical exergy of the streams are not taken into calculation. The physical exergy of the streams is taken as the total exergy. The equations of the calculation of the single effect cycle are given in Table 1 for of each component and for overall cycle.

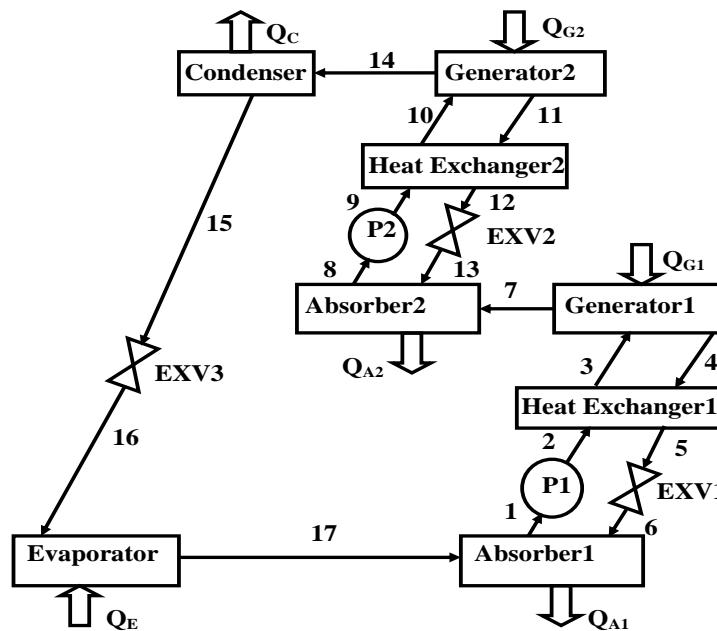


Figure 3. Half effect absorption cooling system.

Table 1. For each component and for overall of the single effect absorption cooling cycle mass, energy, and exergy equations [12, 13].

Component	Mass Equation	Energy Equation	Exergy Equation
Absorber	$\dot{m}_6 + \dot{m}_{10} = \dot{m}_1$	$Q_A = \dot{m}_{10}h_{10} + \dot{m}_6h_6 - \dot{m}_1h_1$	$E_1 = \dot{m}_1(h_1 - h_0 - T_0(s_1 - s_0))$ $E_6 = \dot{m}_6(h_6 - h_0 - T_0(s_6 - s_0))$ $E_{10} = \dot{m}_{10}(h_{10} - h_0 - T_0(s_{10} - s_0))$
Pump	$\dot{m}_1 = \dot{m}_2$	$W_{P1} = \dot{m}_1(h_2 - h_1)$	$E_2 = \dot{m}_2(h_2 - h_0 - T_0(s_2 - s_0))$

Heat exchanger	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_4 = \dot{m}_5$	$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5$	$E_3 = \dot{m}_3(h_3 - h_0 - T_0(s_3 - s_0))$ $E_4 = \dot{m}_4(h_4 - h_0 - T_0(s_4 - s_0))$ $E_5 = \dot{m}_5(h_5 - h_0 - T_0(s_5 - s_0))$
Expansion Valve1	$\dot{m}_5 = \dot{m}_6$	$\dot{m}_5 h_5 = \dot{m}_6 h_6$	
Generator	$\dot{m}_3 = \dot{m}_4 + \dot{m}_7$	$\dot{m}_3 h_3 + Q_G$ $= \dot{m}_4 h_4 + \dot{m}_7 h_7$	$E_7 = \dot{m}_7(h_7 - h_0 - T_0(s_7 - s_0))$
Condenser	$\dot{m}_7 = \dot{m}_8$	$\dot{m}_7 h_7$ $= \dot{m}_8 h_8 + Q_C$	$E_8 = \dot{m}_8(h_8 - h_0 - T_0(s_8 - s_0))$
Expansion Valve2	$\dot{m}_8 = \dot{m}_9$	$\dot{m}_8 h_8 = \dot{m}_9 h_9$	$E_9 = \dot{m}_9(h_9 - h_0 - T_0(s_9 - s_0))$
Evaporator	$\dot{m}_9 = \dot{m}_{10}$	$\dot{m}_9 h_9 + Q_E = \dot{m}_{10} h_{10}$	
Overall cycle	$(\dot{Q}_A + \dot{Q}_C)_{outletenergy} = (Q_G + W_P + \dot{Q}_E)_{inletenergy}$ $COP = \dot{Q}_E / (W_P + \dot{Q}_G)$ $W_P = \dot{m}_{in}(h_{in} - h_{out}) = \Delta P / \rho$ $\dot{E} = Q(1 - \frac{T_0}{T})$ $\dot{E}_{D,G} = \dot{E}_3 + \dot{E}_G - \dot{E}_7 - \dot{E}_4$		$\dot{E}_{D,C} = \dot{E}_7 - \dot{E}_8 - \dot{E}_{C1}$ $\dot{E}_{D,HE} = \dot{E}_2 + \dot{E}_4 - \dot{E}_3 - \dot{E}_5$ $\dot{E}_{D,E} = \dot{E}_9 + \dot{E}_E - \dot{E}_{10}$ $\dot{E}_{D,EXV} = \dot{E}_{in} + \dot{E}_{out}$ $ECOP = \dot{E}_E / (W_{P,tot} + \dot{E}_G)$

## RESULTS AND DISCUSSION

To calculate the enthalpy and the entropy values of the streams a computer program written by the author in FORTRAN codes is used. The equations used in the program to calculate the enthalpy and the entropy values of the streams are taken from the reference [14, 15]. However, the reference state values are taken for 50 % concentration H<sub>2</sub>O-LiBr, at 100 kPa pressure, 25 °C temperature, and as h<sub>0</sub>=49.2 kJ/kg and s<sub>0</sub>=0.1867 kJ/kgK for the mixture of H<sub>2</sub>O/LiBr.

In Table 2 energy, exergy, destructed exergy, COP, ECOP, exergy and energy balance for overall and for each component of the double effect parallel flow absorption cycle are given. For the three cycles the generator heat energy is taken as 4000 kW for the sake of comparison. However the temperature of the heat energy is different for each cycle. In Table 3 and in Table 4 energy, exergy, destructed exergy, COP, ECOP, exergy and energy balance for overall and for each component of the single effect cycle and the half effect cycle are given.

**Table 2. Energy, exergy, destructed exergy, COP, ECOP, exergy and energy balance for overall and for each component of the double effect parallel flow absorption cooling cycle.**

Absorber heat energy- exergy destruction	$Q_A = 6326$ kW, $E_A = E_{D,A} = 492$ kW
Condanser1 heat energy-exergy--exergy destruction	$Q_{C1} = 2325$ kW, $E_{C1} = 470.5$ kW, $(E_{D,LPG} + E_{D,C1}) = 172$ kW,
Condanser2 heat energy-exergy--exergy destruction	$Q_{C2} = 2459$ kW, $E_{C2} = E_{D,C2} = 193$ kW, $\eta_{ex,C2} = 0.02$
Evaporator heat energy- exergy	$Q_E = 4782$ kW, $E_E = 343$ kW, $E_{D,E} = 687$ kW
High pressure generator heat energy	$Q_{HPG} = 4000$ kW, $E_{HPG} = 1209$ kW
Low pressure generator heat energy exergy destruction	$Q_{LPG} = Q_{C1} = 2780$ kW, $E_{LPG} = 364$ kW, $(E_{D,LPG} + E_{D,C1}) = 203$ kW
COP	1.196
ECOP	0.284
Inlet Energy=Outlet Energy $\rightarrow (Q_{HPG} + Q_E = Q_A + Q_{C2}) \rightarrow (4000 + 4782 = 6326 + 2459) \rightarrow 8782 \approx 8785$	
Overall Cycle(inlet exergy ( $E_{OC} = E_{HPG} + E_E = 1209 + 343 = 1552$ ))	
Overall Cycle(inlet exergy=outlet exergy=Lost + Destructed) ( $E_{D,OC} = (E_{D,LPG} + E_{D,C1}) + E_{D,A} + E_{D,C2} + E_{D,E} + E_{D,others} = 172 + 492 + 193 + 687 = 1544 \approx 1552 =$ inlet exergy)	
Error=(1552-1544)/ 1552=0.005	

**Table 3. Energy, exergy, destructed exergy, COP, ECOP, exergy and energy balance for overall and for each component of the single effect absorption cooling cycle.**

Absorber heat energy- exergy destruction	$Q_A = 3780$ kW, $E_A = E_{D,A} + E_{A,Loss} = 245$ kW,
Condenser heat energy-exergy--exergy destruction	$Q_C = 2941$ kW, $E_{C,Lost} + E_{D,C} = 198$ kW

Evaporator heat energy- exergy	$Q_E = 2720.7 \text{ kW}, E_E = 229.3 \text{ kW}, E_{D,E} = 422.7 \text{ kW}$
Generator heat energy- exergy	$Q_G = 4000 \text{ kW}, E_G = 905 \text{ kW}, E_{D,G} = 261.2 \text{ kW}$
COP	0.68
ECOP	0.254
Inlet Energy=Outlet Energy $\rightarrow (Q_G + Q_E = Q_A + Q_C) \rightarrow (4000 + 2720.7 = 3780 + 2941) \rightarrow 6720.7 = 6721$	
Overall Cycle(inlet exergy ( $E_{OC} = E_G + E_E = 905 + 229.3 = 1134.3$ ))	
Overall Cycle(inlet exergy=outlet exergy=Lost + Destroyed)	
( $E_{D,OC} = E_{D,C} + E_{D,A} + E_{D,G} + E_{D,E} = 198 + 245 + 261.2 + 422.7 = 1126.9$ )	
inlet exergy=outlet exergy $\rightarrow 1134.3 \approx 1126.9$	
Error=(1134.3-1126.9)/ 1134.3=0.006	

**Table 4. Energy, exergy, destroyed exergy, COP, ECOP, exergy and energy balance for overall and for each component of the half effect absorption cooling cycle.**

Absorber1 heat energy- exergy destruction	$Q_{A1} = 2001 \text{ kW}, E_{A1} = E_{D,A1} + E_{A1,Loss} = 76.4$
Generator1 heat energy- exergy	$Q_{G1} = 2061 \text{ kW}, E_{G1} = 337.8 \text{ kW}$
Absorber2 heat energy- exergy destruction	$Q_{A2} = 1933 \text{ kW}, E_{A2} = E_{D,A2} + E_{A2,loss} = 239.1 \text{ kW}$
Generator2 heat energy- exergy	$Q_{G2} = 1939 \text{ kW}, E_{G2} = 150.5 \text{ kW}$
Condenser heat energy-exergy--exergy destruction	$Q_C = 1879.7 \text{ kW}, E_C = E_{C,Lost} + E_{D,C} = 46 \text{ kW}$
Evaporator heat energy- exergy	$Q_E = 1818.5 \text{ kW}, E_E = 117 \text{ kW}, E_{D,E} = 241.5 \text{ kW}$
COP	0.455
ECOP	0.24
Inlet Energy=Outlet Energy $\rightarrow (Q_{G,TOT} + Q_E = Q_{A,TOT} + Q_C)$ (2061+1939+1818.5=2001+1933+1879.7) $\rightarrow 5818.5 \approx 5813.7$	
Overall Cycle(inlet exergy ( $E_{OC} = E_{G,TOT} + E_E = 337.8 + 150.5 + 117 = 605.3$ ))	
Overall Cycle(inlet exergy=outlet exergy=Lost + Destroyed)	
( $E_{D,OC} + E_{Loss,OC} = E_C + E_{A1} + E_{A2} + E_{D,E} = 46 + 76.4 + 239.1 + 241.5 = 603$ )	
inlet exergy=outlet exergy $\rightarrow 605.3 \approx 603$	
Error=(605.3-603)/ 605.3=0.004	

For a refrigeration system, the coefficient performance (COP) and the exergetic coefficient performance (ECOP) are the most important thing for evaluation and consideration. As can be seen in these tables the double effect cycle has the highest COP, and ECOP values which are 1.196 and 0.284. The COP and the ECOP of the single effect cycle which are 0.68 and 0.254 are higher than the half one COP, and ECOP values which are 0.455 and 0.24. The reason is that the temperature of the heat energy given to the double effect cycle is about 175 °C, the temperature of the heat energy given to the single effect cycle is about 105 °C, and the temperature of the heat energy given to the half effect cycle is about 50 °C. Higher heat source temperatures means higher COP and ECOP for the absorption cooling cycles. The evaporators have the maximum destroyed exergy rates between other components of the three cycles. The absorbers of the three cycles have higher destroyed exergy rates between other components except evaporators. The destroyed exergy of pumps, heat exchangers and expansion valves are small so that they can be ignored. Better design of the evaporators and the absorbers will directly affect and improve the working conditions and the performance of the overall cycles.

These results showed that the double effect systems have higher COP and ECOP values than the single effect and half effect systems. However the single effect systems required fewer components to operate than the double effect and the half effect ones. Therefore the single effect systems are less expensive and simpler than the double effect and half effect ones. These results are in good agreement with the literature. 0.004 % Error is happened in all this calculation of the three cycles which can be ignored. These results showed that the half effect systems are very appropriate for using low temperature heat energy in cooling. These results are in good agreement with the literature.

## CONCLUSION

Absorption cooling cycles are environmental and can use solar or waste heat for cooling with very small electric power and they can decrease the costs of cooling. This work presents exergy analysis of a double effect parallel flow, single effect and half effect absorption systems for comparison. A computer program is developed for the thermodynamic properties of lithium bromide-water solutions by the author in FORTRAN codes for the exergy analysis. The double effect parallel flow absorption systems have better advantages than the single effect and the

half effect absorption systems. The coefficient performance (COP) and the exergetic coefficient performance (ECOP) of the double effect parallel flow absorption systems are higher than the single effect and the half effect cycles. For the double effect cycle COP and ECOP are found as 1.196 and 0.284, and for the single effect cycle COP and ECOP are found as 0.68 and 0.254, respectively. For the half effect cycle COP and ECOP are found as 0.455 and 0.24, respectively. For each component the exergy loss is calculated. Most of the irreversibilities are found in the evaporator and in the absorber. It is concluded that the performance of the evaporator and the absorber is crucial for the three cycles. Better design and improving of these two components will directly improve and affect positively the performance and the working conditions of the three cycles.

## REFERENCES

- [1] ASHRAE Fundamentals Handbook, "Chapter 1. Thermodynamics and refrigeration cycles", 2001.
- [2] P. Srikihirin, S. Aphornratana, and S. Chungpaibulpatana, "A review of absorption refrigeration technologies", *Renewable & Sustainable Energy Reviews*, VOL. 5, pp. 343-372, 2001.
- [3] L. Garousi Farshi, S.M.S. Mahmoudi, M.A. Rosen, M. Yari, and M. Amidpour, "Exergoeconomic analysis of double effect absorption refrigeration systems", *Energy Conversion and Management*, VOL:65, pp.13-25, 2013.
- [4] L.A. Domínguez-Inzunza, M. Sandoval-Reyes, J.A. Hernández-Magallanes and W. Rivera, "Comparison of the performance of single-effect, half-effect, double-effect in series and inverse absorption cooling systems operating with the mixture H<sub>2</sub>O/LiBr", *Energy Procedia*, VOL: 57, pp.2534-2543, 2014.
- [5] L.A. Domínguez-Inzunza, J.A. Hernández-Magallanes, M. Sandoval-Reyes, and W. Rivera, "Comparison of the performance of single-effect, half-effect, double-effect in series and inverse and triple-effect absorption cooling systems operating with the NH<sub>3</sub>-LiNO<sub>3</sub> mixture", *Applied Thermal Engineering*, VOL: 66, pp:612-620, 2014.
- [6] K. Talukdar, and T.K. Gogoi, "Exergy analysis of a combined vapor power cycle and boiler flue gas driven double effect water-LiBr absorption refrigeration system", *Energy Conversion and Management*, VOL:108, pp.468-477, 2016.
- [7] D. Colorado, and W. Rivera, "Performance comparison between a conventional vapor compression and compression-absorption single-stage and double-stage systems used for refrigeration", *Applied Thermal Engineering*, VOL:87, pp: 273-285, 2015.
- [8] Omer Kaynakli, Kenan Saka, and Faruk Kaynakli, "Energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources", *Energy Conversion and Management*, VOL:106, pp.21-30, 2015.
- [9] T. Avanesiana, and M. Ameri, "Energy, exergy, and economic analysis of single and double effect LiBr-H<sub>2</sub>O absorption chillers", *Energy and Buildings*, VOL: 73, 26-36, 2014.
- [10] Nahla Bouaziz, and D. Lounissi, "Energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling", *International Journal of Hydrogen Energy*, VOL:40, pp.13849-3856, 2015.
- [11] B.H. Gebreslassie, M. Medrano, and D. Boer, "Exergy analysis of multi-effect water-LiBr absorption systems: From half to triple effect", *Renewable Energy*, VOL:35, pp.1773-1782, 2010.
- [12] K. Annamalai, I.K. Puri, *Advanced thermodynamics engineering*, CRC Press LLC, 2002.
- [13] I. Dincer, M.A. Rosen, *EXERGY, energy, environment and sustainable development*, 1st ed., Elsevier Ltd., 2007.
- [14] H.T. Chua, H.K. Toh, A. Malek, K.C. Ng, K. Srinivasan, "Improved thermodynamic property fields of LiBr-H<sub>2</sub>O solution", *International Journal of Refrigeration*, VOL:23, pp:412-429, 2000.
- [15] Y. Kaita, "Thermodynamic properties of lithium bromide-water solutions at high temperature", *International Journal of Refrigeration*, VOL: 24, pp:374-390, 2001.