

ABSTRACT

Absorption cooling cycles have the advantage of being environmental and using solar or waste heat for cooling with very small electric power. This work presents exergy analysis of a double effect parallel flow absorption system. For the exergy analysis a computer program is developed for the thermodynamic properties of lithium bromide-water solutions by the author in FORTRAN codes. The exergy analysis of the cycle has showed that the exergetic coefficient performance (ECOP) is about 0.28 and that means the cycle is an efficient cycle. The exergy destruction of each component of the cycle is calculated. Most of the irreversibilities in the cycle occurred in the evaporator and in the absorber which both is about 74 % of the total irreversibility. That means the efficiency of the evaporator and the absorber is crucial for the double effect absorption cycles. To improve the performance and the working conditions of the cycle better design and improving of these two components is essential.

KEYWORDS: Absorption, Cooling, 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

η	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
P	pump
0	environment conditions,

INTRODUCTION

In a foreseeable future, there is a growing need for refrigeration in industry and building cooling all over the world. The main causes of this growing are the increasing thermal load of buildings, higher living standards and increasing requirements for comfort. Using low temperature waste heat, solar or geothermal energy can reduce power consumption for cooling. For cooling purposes absorption chillers using a LiBr-H₂O solution offer very good efficiency than the other solutions at over 0 °C temperature of evaporator. But there is a risk of salt crystal formation called solution crystallization. This happens when there is air leak into machine or high absorber temperature and low ambient temperature. To produce cold at temperatures below than 0 °C, the ammonia-water solution, ammonia-lithium nitrate or other appropriate solutions can be used. The COP of an absorption cycle depends on three external temperatures; evaporation, generation (driving) and ambient temperatures. The half effect cycles presents the lowest COP, the single effect cycle has better COP than the half one. The double effect cycle has better COP than the single one; however the triple effect cycle has the best COP among them. The details of these cycles, their configuration and their differences can be found in literature. The single and the double effect absorption refrigeration cycles have more commercial use than the triple one and half one [1, 2]. The air cooled double effect systems are better than the single one because they are more flexible, efficient, independence upon water and without cooling tower.

Double effect means that the same driving heat source produces refrigerant vapor twice. For this process two vapor generators are needed. Many configuration of double effect cycles are obtained and the most common ones 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 means that the solution flow going to the high pressure generator does not go to the lower pressure generator, the solution stream split among both. In-series layouts are better in the cooling capacity than the in-parallel layouts; however the in-parallel layouts have higher COP [3].

Izquierdo *et al.*, have experimentally and analyzed solar-powered single-and double-effect directly air-cooled LiBr-H₂O absorption prototype built as a single unit. They found that the COP of the single effect mode is between 0.54-0.63 and the COP of the double effect mode is doubled versus that of the single one [3]. Ventas *et al.*, have analyzed two-stage double-effect ammonia/lithium nitrate absorption cycle and they concluded that maximum COP is about 1.25 and adiabatic absorber offers better performance than the diabatic version [4]. Vasilescu and Ferreira have analyzed solar driven double-effect absorption cycles for sub-zero temperatures and they found that a double effect ammonia- lithium nitrate solution is an appropriate solution for this process [5]. Gomri and Hakimi studied on the second law analysis of double effect vapor absorption cooler system and they concluded that the performance of the system increases with increasing low pressure generator temperature and the highest exergy losses occurs in absorber and high pressure generator [6]. Colorado and Rivera studied on the performance comparison between a conventional vapor compression and compression-absorption single-stage and double-stage systems used for refrigeration [7]. They found that the compression power in cascade cycles was 45 % lower than in compression cycles. Li *et al.*, studied on performance analysis of solar air cooled double effect LiBr/H₂O absorption cooling system in subtropical city. They found that the performance was decreasing with increasing the collector temperature [8]. Talukdar and Gogoi studied on exergy analysis of a combined vapor power cycle and boiler flue gas driven double effect water-LiBr absorption refrigeration system. They concluded that the double effect absorption system is better and more appropriate than the single effect for integration with power cycle [9]. Farshi *et al.*, about exergo-economic analysis of double effect absorption refrigeration systems they found that lower total investment costs were obtained when the evaporator temperatures are high and the condenser temperatures were low [10]. Avanesian and Ameri have studied on energy, exergy, and economic analysis of single and double effect LiBr-H₂O absorption chillers. They concluded that the double effect absorption chillers are more economical than the single ones [11]. Bouaziz and Lounissi have done 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 [12]. Kaynakli *et al.*, in their study named energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources; they have reported that exergy destruction of the high pressure generator increases with higher temperatures of the heat sources [13]. Inzunza *et al.*, studied the comparison of the performance of single-effect, half-effect, double-effect in series and inverse absorption cooling systems operating with the mixture H₂O/LiBr. They found that the COP of the single effect was up to 0.89 for the generation temperature between 100 °C and 110 °C, the COP of the half effect was up to 0.44 for the generation temperature of over 55 °C. They also found that the double effect systems were the most efficient that the COP were up to 1.48. They mentioned that half effect systems work better than any other for low temperatures [2]. 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 $\text{NH}_3\text{-LiNO}_3$ mixture. They found that the COP values of $\text{H}_2\text{O/LiBr}$ are higher than the COP values of $\text{NH}_3\text{-LiNO}_3$. However with $\text{NH}_3\text{-LiNO}_3$ refrigeration solution the evaporator temperature can be as low as $-50\text{ }^\circ\text{C}$ [14]. Cimsit *et al.*, have done the thermo-economic optimization of $\text{LiBr/H}_2\text{O-R134a}$ compression absorption cascade refrigeration cycle. They found that the cascade cycle has the potential to reduce electric energy consumption about 50 % [15].

The goal of this study is to investigate the irreversibility and the exergy efficiency of the components of the cycle, and to obtain the exergetic coefficient of performance of the double effect parallel flow absorption system. The cycle and working conditions is taken from the reference [1].

MATERIALS AND METHODS

The schematic diagram of a double effect parallel flow absorption system is given in Figure 1. 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.

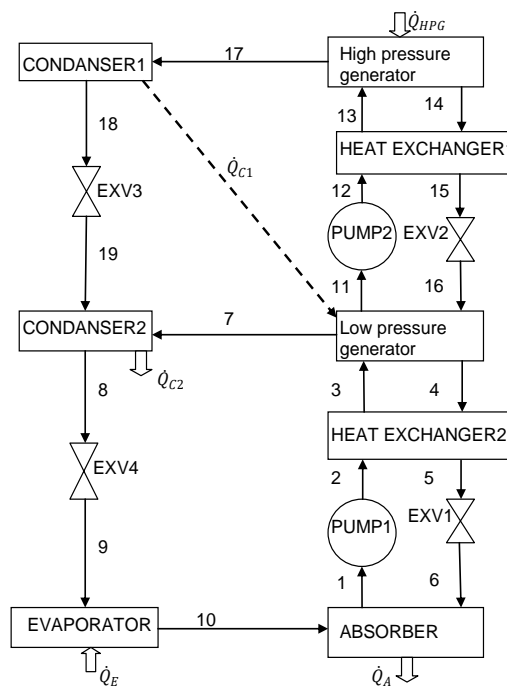


Figure 1. Schematic diagram of a double effect parallel flow absorption system

In this study, these assumptions are utilized in the analysis of the cycle: The pump process is adiabatic, the cycle is at steady state and steady flow cycle, pressure drops in the pipeline and in the components are neglected, the pressure reducing valve is an adiabatic process, refrigerant leaving the condenser is saturated liquid at condenser pressure, 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, direct heat transfer from the components to the surroundings is negligible, refrigerant is pure water. In this study the thermodynamic analysis of the cycle which is introduced in the previous section is done and the thermodynamic and the mathematical modeling explained as follows. There is no mass inlet or outlet of the cycle so that the chemical exergy of the streams are not taken into calculation. The total exergy is taken as the physical exergy of the streams. The equations of the calculation of the cycle are given in Table 1 for of each component and for overall cycle.

Table 1. For of each component and for overall cycle mass, energy, and exergy equations [16, 17].

Component	Mass Equation	Energy Equation	Exergy Equation
Pump1	$\dot{m}_1 = \dot{m}_2$	$W_{P1} = \dot{m}_1(h_2 - h_1)$	$E_1 = \dot{m}_1(h_1 - h_0 - T_0(s_1 - s_0))$ $E_2 = \dot{m}_2(h_2 - h_0 - T_0(s_2 - s_0))$
Heat exchanger2	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_4 = \dot{m}_5$	$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5$	$E_3 = \dot{m}_3(h_3 - h_0 - T_0(s_3 - s_0))$ $E_4 = \dot{m}_4(h_4 - h_0 - T_0(s_4 - s_0))$ $E_5 = \dot{m}_5(h_5 - h_0 - T_0(s_5 - s_0))$
Expansion Valve1	$\dot{m}_5 = \dot{m}_6$	$\dot{m}_5 h_5 = \dot{m}_6 h_6$	$E_6 = \dot{m}_6(h_6 - h_0 - T_0(s_6 - s_0))$
Absorber	$\dot{m}_1 = \dot{m}_{10} + \dot{m}_6$	$Q_A = \dot{m}_{10} h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1$	$E_{10} = \dot{m}_{10}(h_{10} - h_0 - T_0(s_{10} - s_0))$
Low Pressure Generator	$\dot{m}_3 + \dot{m}_{16} = \dot{m}_4 + \dot{m}_{11}$	$\dot{m}_3 h_3 + \dot{m}_{16} h_{16} + Q_C = \dot{m}_4 h_4 + \dot{m}_{11} h_{11}$	$E_{11} = \dot{m}_{11}(h_{11} - h_0 - T_0(s_{11} - s_0))$ $E_{16} = \dot{m}_{16}(h_{16} - h_0 - T_0(s_{16} - s_0))$
Pump2	$\dot{m}_{11} = \dot{m}_{12}$	$W_{P2} = \dot{m}_{11}(h_{12} - h_{11})$	$E_{12} = \dot{m}_{12}(h_{12} - h_0 - T_0(s_{12} - s_0))$
Heat exchanger1	$\dot{m}_{12} = \dot{m}_{13}$ $\dot{m}_{14} = \dot{m}_{15}$	$\dot{m}_{12} h_{12} + \dot{m}_{14} h_{14} = \dot{m}_{13} h_{13} + \dot{m}_{15} h_{15}$	$E_{13} = \dot{m}_{13}(h_{13} - h_0 - T_0(s_{13} - s_0))$ $E_{14} = \dot{m}_{14}(h_{14} - h_0 - T_0(s_{14} - s_0))$ $E_{15} = \dot{m}_{15}(h_{15} - h_0 - T_0(s_{15} - s_0))$
Expansion Valve2	$\dot{m}_{15} = \dot{m}_{16}$	$\dot{m}_{15} h_{15} = \dot{m}_{16} h_{16}$	
High pressure Generator	$\dot{m}_{13} = \dot{m}_{14} + \dot{m}_{17}$	$\dot{m}_{13} h_{13} + Q_G = \dot{m}_{14} h_{14} + \dot{m}_{17} h_{17}$	$E_{17} = \dot{m}_{17}(h_{17} - h_0 - T_0(s_{17} - s_0))$
Condanser1	$\dot{m}_{17} = \dot{m}_{18}$	$\dot{m}_{17} h_{17} = Q_{C1} + \dot{m}_{18} h_{18}$	$E_{18} = \dot{m}_{18}(h_{18} - h_0 - T_0(s_{18} - s_0))$
Expansion Valve3	$\dot{m}_{18} = \dot{m}_{19}$	$\dot{m}_{18} h_{18} = \dot{m}_{19} h_{19}$	$E_{19} = \dot{m}_{19}(h_{19} - h_0 - T_0(s_{19} - s_0))$
Condanser2	$\dot{m}_7 + \dot{m}_{19} = \dot{m}_8$	$\dot{m}_{19} h_{19} + \dot{m}_7 h_7 = Q_{C2} + \dot{m}_8 h_8$	$E_7 = \dot{m}_7(h_7 - h_0 - T_0(s_7 - s_0))$ $E_8 = \dot{m}_8(h_8 - h_0 - T_0(s_8 - s_0))$
Expansion Valve4	$\dot{m}_8 = \dot{m}_9$	$\dot{m}_8 h_8 = \dot{m}_9 h_9$	$E_9 = \dot{m}_9(h_9 - h_0 - T_0(s_9 - s_0))$
Evaporator	$\dot{m}_9 = \dot{m}_{10}$	$\dot{m}_9 h_9 + Q_E = \dot{m}_{10} h_{10}$	
Overall cycle	$(\dot{Q}_A + \dot{Q}_{C2})_{outletenergy} = (Q_{HPG} + W_P + \dot{Q}_E)_{inletenergy}$ $COP = \dot{Q}_E / (W_{P,tot} + \dot{Q}_{HPG})$ $W_P = \dot{m}_{in}(h_{in} - h_{out})$ $\dot{E}_{HPG} = \dot{E}_{17} + \dot{E}_{14} - \dot{E}_{13}$ $\dot{E}_E = \dot{E}_9 - \dot{E}_{10}$ $ECOP = \dot{E}_E / (W_{P,tot} + \dot{E}_{HPG})$ $\eta_{ex,component} = 1 - \frac{\dot{E}_D}{\dot{E}_{in}}$		$\dot{E}_{D,C1} = \dot{E}_{17} - \dot{E}_{18} - \dot{E}_{C1}$ $\dot{E}_{D,C2} = \dot{E}_{19} + \dot{E}_7 - \dot{E}_8$ $\dot{E}_{D,A} = \dot{E}_9 + \dot{E}_3 - \dot{E}_4$ $\dot{E}_{D,HE1} = \dot{E}_{12} + \dot{E}_{14} - \dot{E}_{13} - \dot{E}_{15}$ $\dot{E}_{D,HE2} = \dot{E}_2 + \dot{E}_4 - \dot{E}_3 - \dot{E}_5$ $\dot{E}_{D,E} = \dot{E}_9 + \dot{E}_E - \dot{E}_{10}$ $\dot{E}_{D,EXV} = \dot{E}_{in} + \dot{E}_{out}$ $\dot{E}_{D,LPG} = \dot{E}_{C1} + \dot{E}_3 + \dot{E}_{16} - \dot{E}_{11} - \dot{E}_4$

RESULTS AND DISCUSSION

The enthalpy and entropy values of the streams are calculated with a computer program written by the authors in FORTRAN codes. The equations used in the program to calculate the enthalpy and the entropy values of the streams are taken from the reference [18, 19]. However, for the mixture of H₂O/LiBr, the reference state values are taken at 25 °C temperature, 100 kPa pressure and for 50 % concentration kgNH₃/kgmix, as h₀=49.2 kJ/kg and s₀=0.1867 kJ/kgK.

Table 2. Thermodynamic properties of the streams of the double effect parallel flow absorption cycle.

Stream Nu.	Fluid	Pressure kPa	Temperature °C	Concentration kgNH ₃ /kgmix	Flow rate Kg/s	Enthalpy kJ/kg	Entropy kJ/kgK	Exergy kW
0	LiBr/H ₂ O	100	25	50	-	49.2	0.1867	-
1	Weak	0.88	42.4	59.5	9.551	117.7	0.2466	483.7

2	Weak LiBr/H ₂ O	8.36	42.4	59.5	9.551	117.7	0.2465	484
3	Weak LiBr/H ₂ O	8.36	75.6	59.5	9.551	182.3	0.4381	555.3
4	Strong LiBr/H ₂ O	8.36	97.8	64.6	8.797	247.3	0.5149	881.9
5	Strong LiBr/H ₂ O	8.36	58.8	64.6	8.797	177.2	0.3163	786.1
6	Strong LiBr/H ₂ O	0.88	53.2	64.6	8.797	177.2	0.3169	785.1
7	H ₂ O	8.36	85.6	0	0.32	2661.1	8.3693	55.1
8	H ₂ O	8.36	42.4	0	0.754	177.4	0.6164	0.1
9	H ₂ O	0.88	5	0	0.754	177.4	0.618	-0.3
10	H ₂ O	0.88	5	0	0.754	2510.8	9.007	-126.8
11	Weak LiBr/H ₂ O	8.36	85.6	59.5	5.498	201.8	0.4929	337.1
12	Weak LiBr/H ₂ O	111.8	85.6	59.5	5.498	201.8	0.4927	337.4
13	Weak LiBr/H ₂ O	111.8	136.7	59.5	5.498	301.2	0.754	455.6
14	Strong LiBr/H ₂ O	111.8	170.7	64.6	5.064	378.8	0.845	675.2
15	Strong LiBr/H ₂ O	111.8	110.9	64.6	5.064	270.9	0.5777	532.3
16	Strong LiBr/H ₂ O	8.36	99.1	64.6	5.064	270.9	0.5779	532
17	H ₂ O	111.8	155.7	0	0.434	2787.3	7.6355	224.5
18	H ₂ O	111.8	102.8	0	0.434	430.6	1.335	16.9
19	H ₂ O	8.36	42.4	0	0.434	430.6	1.34	16.1
Absorber heat energy- exergy destruction					Q _A = 2328 kW, E _A = E _{D,A} = 174 kW, η _{ex,A} = 0.74			
Pump1 work energy					W _{P1} = 0.043kW			
Pump1 work energy					W _{P2} =0.346 kW			
Condanser1 heat energy-exergy--exergy destruction					Q _{C1} =1023 kW, E _{C1} =207 kW, (E _{D,LPG} +E _{D,C1})=75.9 kW, η _{ex,C1+LPG} = 0.94			
Condanser2 heat energy-exergy--exergy destruction					Q _{C2} =905 kW, E _{C2} = E _{D,C2} =71kW, η _{ex,C2} = 0.02			
Expansion valve1,2,3,4 exergy destruction					E _{D,EXV1} =0.4 kW, E _{D,EXV2} =0.3 kW, E _{D,EXV3} =0.8 kW, E _{D,EXV4} =0.4 kW			
Heat Exchanger1--exergy destruction, Exergy efficiency					E _{D,HE1} =5.3 kW, η _{ex, HE1} = 0.83			
Heat Exchanger2--exergy destruction, Exergy efficiency					E _{D,HE2} =5.4 kW, η _{ex, HE2} = 0.74			
Evaporator heat energy- exergy					Q _E = 1760 kW, E _E = 126.5 kW, E _{D,E} = 253 kW			
High pressure generator heat energy					Q _{HPG} = 1472 kW, E _{HPG} = 445 kW			
Low pressure generator heat energy exergy destruction					Q _{LPG} =Q _{C1} =1023 kW, E _{LPG} = 133.9 kW, (E _{D,LPG} +E _{D,C1})=75 kW, η _{ex,C1+LPG} = 0.94			
COP					1.195			
ECOP					0.28			
Inlet Energy=Outlet Energy → (W _{P,TOT} + Q _{HPG} + Q _E =Q _A + Q _{C2}) → (0.3+1472+1760=2328+905)→3233≈3232.3								
Overall Cycle(inlet exergy (E _{OC} = E _{HPG} + W _{P,TOT} +E _E =445+0.4+126.5=572)								
Overall Cycle(inlet exergy=outlet exergy=Lost + Destroyed)								
(E _{D,OC} = (E _{D,LPG} +E _{D,C1})+ E _{D,A} + E _{D,C2} + E _{D,E} + E _{D,others} =75+174+71+253+12=585≈572= inlet exergy)								
Efficiency=(585-572)/572=0.02								

In Table 2 for each stream, the fluids, the pressures, the temperatures, the concentrations, the flow rates values, the enthalpies, the entropies, the exergy, and the energy, the heat exergy, the destructed exergy of each component, exergy efficiency, COP, ECOP, and the energy balance of the cycle are given.

As can be seen that for the overall cycle the total destructed and lost exergy is about 572 kW, and the 253 kW of it is destructed at the evaporator that means 44 % of the total destructed exergy. The destructed exergy in the absorber found as 174 kW, and that is 30 % of the total destructed exergy. In the evaporator and in the absorber 74 % of the total is destructed. That means the efficiency of the evaporator and the absorber is crucial for the cycle. Better design and improving of these two components will directly affect and improve the performance and the working conditions of the cycle. The condenser2 also has high destructed exergy that is about 71 kW and that is 12 % of the total destructed exergy. The destructed exergy in the evaporator, at the absorber and at the condenser2 is 86 % of the total destructed exergy. That means the efficiency of these three components is crucial for the cycle.

The destructed exergy in the heat exchangers are nearly the same that is about 5 kW and both of them destructed 2 % of the total irreversibility. The irreversibility in the heat exchangers, in the pumps and in the expansion valves is small, but they are taken into calculation. These components exergetic efficiency are high and the irreversibility is small. The coefficient performance (COP) of the cycle is 1.195 and the exergetic coefficient performance (ECOP) of the cycle is 0.28. These results are in good agreement with the literature. In all this calculation 2 % error is happened which can be ignored.

CONCLUSION

Absorption cooling cycles have the advantage of being environmental and to use solar or waste heat for cooling with very small electric power. The double effect parallel flow absorption systems have better advantages than the half and single effect absorption system. The coefficient performance (COP) and the exergetic coefficient performance (ECOP) are higher than the single and half cycles. This work presents exergy analysis of a double effect parallel flow absorption system. For the exergy analysis a computer program is developed for the thermodynamic properties of lithium bromide-water solutions by the author in FORTRAN codes. The exergy analysis of the cycle has showed that the exergetic coefficient performance (ECOP) is about 0.28 and that means the cycle is an efficient cycle. The exergy loss of each component is calculated. Most of the irreversibilities in the cycle occurred in the evaporator and in the absorber which about 74 % of the total irreversibility. That means the efficiency of the evaporator and the absorber is crucial for the double effect absorption cycles. To improve the performance and the working conditions of the cycle better design and improving of these two components is essential.

REFERENCES

- [1] ASHRAE Fundamentals Handbook 2001 Chapter 1. Thermodynamics and refrigeration cycles.
- [2] L.A. Domínguez-Inzunza, M. Sandoval-Reyes, J.A. Hernández-Magallanes, W. Rivera, "Comparison of the performance of single-effect, half-effect, double-effect in series and inverse absorption cooling systems operating with the mixture H₂O/LiBr", *Energy Procedia* 57, 2534-2543, 2014.
- [3] M. Izquierdo, A. González-Gil, E. Palacios, "Solar-powered single-and double-effect directly air-cooled LiBr-H₂O absorption prototype built as a single unit", *Applied Energy* 130, 7-19, 2014.
- [4] R. Ventas, A. Lecuona, C. Vereda, M. Legrand, "Two-stage double-effect ammonia/lithium nitrate absorption cycle", *Applied Thermal Engineering* 94, 228-237, 2016.
- [5] Catalina Vasilescu, Carlos Infante Ferreira, "Solar driven double-effect absorption cycles for sub-zero temperatures", *International journal of refrigeration* 39, 86-94, 2014.
- [6] Rabah Gomri, Riad Hakimi, "Second law analysis of double effect vapor absorption cooler system. *Energy Conversion and Management* 49, 3343-3348, 2008.
- [7] D. Colorado, W. Rivera, "Performance comparison between a conventional vapor compression and compression-absorption single-stage and double-stage systems used for refrigeration", *Applied Thermal Engineering* 87, 273-285, 2015.
- [8] Zeyu Li, Xiangyang Ye, Jinping Liu, "Performance analysis of solar air cooled double effect LiBr/H₂O absorption cooling system in subtropical city", *Energy Conversion and Management* 85, 302-312 2014.
- [9] K. Talukdar, 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* 108, 468-477, 2016.

- [10] L. Garousi Farshi, S.M.S. Mahmoudi, M.A. Rosen, M. Yari, M. Amidpour, "Exergoeconomic analysis of double effect absorption refrigeration systems", *Energy Conversion and Management* 65, 13–25, 2013.
- [11] T. Avanesianab, M. Ameri, "Energy, exergy, and economic analysis of single and double effect LiBr–H₂O absorption chillers", *Energy and Buildings* 73, 26–36, 2014.
- [12] Nahla Bouaziz, D. Lounissi, "Energy and exergy investigation of a novel double effect hybrid absorption refrigeration system for solar cooling" *International journal of hydrogen energy* 40, 3849- 3856, 2015.
- [13] Omer Kaynakli, Kenan Saka, Faruk Kaynakli, "Energy and exergy analysis of a double effect absorption refrigeration system based on different heat sources" *Energy Conversion and Management* 106, 21–30, 2015.
- [14] L.A. Domínguez-Inzunza, J.A. Hernández-Magallanes, M. Sandoval-Reyes, 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 NH₃-LiNO₃ mixture", *Applied Thermal Engineering* 66, 612-620, 2014.
- [15] Canan Cimsit, Ilhan Tekin Ozturk, Olcay Kincay,"Thermoeconomic optimization of LiBr/H₂O-R134a compressionabsorption cascade refrigeration cycle" *Applied Thermal Engineering* 76, 105-115, 2015. <http://dx.doi.org/10.1016/j.applthermaleng.2014.10.094>
- [16] K. Annamalai, I.K. Puri, *Advanced thermodynamics engineering*, CRC Press LLC, 2002.
- [17] I. Dincer, M.A. Rosen, "EXERGY, energy, environment and sustainable development", 1st ed., Elsevier Ltd., 2007.
- [18] H.T. Chua, H.K. Toh, A. Malek, K.C. Ng, K. Srinivasan, "Improved thermodynamic property fields of LiBr-H₂O solution" *International Journal of Refrigeration* 23, 412-429, 2000.
- [19] Y. Kaita, "Thermodynamic properties of lithium bromide-water solutions at high temperature" *International Journal of Refrigeration* 24, 374-390, 2001.