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EXERGY ANALYSIS OF HALF-EFFECT ABSORPTION COOLING CYCLES

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ABSTRACT

Absorption cooling cycles are environmental, clean and economically driven cycles. By consuming very small electric power, they can use waste heat or solar energy for cooling. This work presents exergy analysis of a half effect absorption system. A computer program is developed by the author in FORTRAN codes for the thermodynamic properties of lithium bromide-water solutions to use for the exergy analysis. The half effect absorption systems have the advantage of using low temperature heat energy for cooling. The coefficient performance (COP) and the exergetic coefficient performance (ECOP) of the half effect absorption systems are found as 0.45 and 0.24, respectively. For each component the exergy loss is calculated. Most of the irreversibilities are found in the evaporator and in the absorbers. It is concluded that the half effect absorption systems is the best for cooling driven by low temperature heat energy and the performance of the evaporator and the absorbers is very important for the cycle.

KEYWORDS: Cooling, Absorption, Half 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
TOT	total
0	environment conditions

INTRODUCTION
In the most of industrial process waste heat energy increases production cost. Using this waste heat to produce power or cooling can decrease the costs and increases the thermo-economic efficiency of the industrial systems. A growing need for refrigeration in industry and for building cooling all over the world has seen by the experts. The main causes of this growing are the demands of higher living standards, the increasing requirements for comfort, and the increasing thermal load of buildings. Using LiBr-H_2O solution offers very good efficiency for cooling at over 0 ^{0}C temperature than the other solutions. But the solution crystallization that means salt crystal formation is a real risk that happens when there is low ambient temperature or air leak into machine and high absorber temperature. To produce cold at temperatures below than 0 ^{0}C, the ammonia-water solution, the ammonia-lithium nitrate, or other appropriate solutions can be used better [1]. Three external temperatures affects the COP of an absorption cycle; ambient, generation (driving) and evaporation temperatures. Among the half effect, the single effect, and the double effect cycles the triple effect cycle has the best COP, however it needs heat energy at over 150 ^{0}C temperatures. The half effect (also called double lift) cycles have the lowest COP, however it needs heat energy at between 50-90 ^{0}C temperature. The double effect cycle has better COP than the single one, and the heat energy at between 110-150 ^{0}C temperatures is needed. The heat energy at between 90-120 ^{0}C temperatures is used for the single effect absorption cycles. The details of these cycles, their differences and their configuration can be found in literature. The double and the single effect absorption cycles have more commercial use than the half one and triple one [2, 3]. The air cooled double effect systems are better than the single one because they are more efficient, flexible, without cooling tower and independence upon water. However, for low temperature heat energy the half effect one is the best [2, 3].

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 [4]. Gomri and Hakimi have done the second law analysis of double effect vapor absorption cooler system and they obtained that increasing low pressure generator temperature increases the performance of the system and the highest exergy losses occurs in high pressure generator and absorber [5]. 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_2O/LiBr. They found that for the generation temperature between 100 ^{0}C and 110 ^{0}C, the COP of the single effect was up to 0.89, for the generation temperature of over 55 ^{0}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 [6]. 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_3-$LiNO_3$ mixture. They obtained that the COP values of H_2O/LiBr are higher than the COP values of NH_3-$LiNO_3$; however the evaporator temperature can be as low as -50 ^{0}C with NH_3-$LiNO_3$ refrigeration solution [7]. 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 [8]. They concluded that the compression power of the cascade cycles was 45 % lower than in compression cycles. Wang and Zheng have studied on the performance of one and half effect absorption cooling cycle of H_2O/LiBr system, they found that 1.5-effect cycles include the advantages of the single-effect and half-effect absorption cooling cycles [9]. Arivazhagan et al., have investigated on the performance of a HFC based two-stage half effect absorption cooling system and they concluded that the system is capable of producing evaporating temperature as low as -7 ^{0}C with generator temperatures ranging from 55 to 75 ^{0}C [10].

The goal of this study is to investigate and compare the irreversibility and the exergetic coefficient of performance (ECOP) of the half effect absorption system. The cycle and its working conditions are taken from the reference [3].

MATERIALS AND METHODS
The schematic diagram of a half effect absorption system is given in Figure 1.
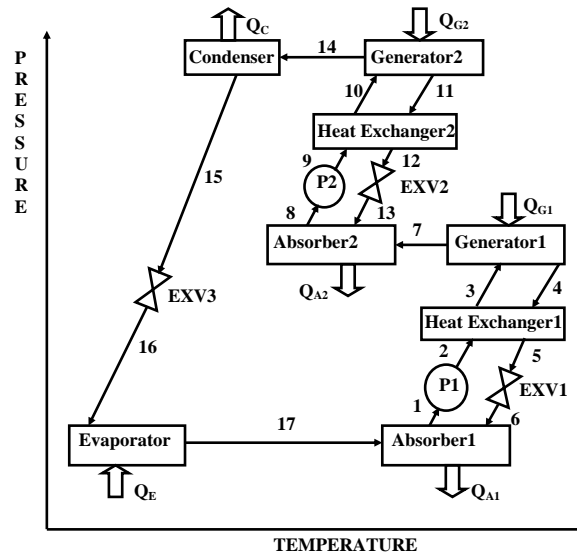
Figure 1. Schematic diagram of a half effect absorption cooling system.

A half effect absorption system consists of two absorbers, a condenser, two generators, an evaporator, two heat exchangers, two pumps and three expansion valves. The half effect absorption system has two H_2O-LiBr solution circuits.

The evaporated H_2O is produced in the generator2and then it is conducted to the condenser. The condensed H_2O in the condenser is expanded in the expansion valve3 and then evaporated in the evaporator. When the refrigerant absorbs heat from the environment the cooling process is obtained in the evaporator. The evaporated refrigerant is conducted to the absorber1. The vapor mixes with H_2O-LiBr solution coming from the generator1 and the low pressure absorber1 releases heat. After that the weak H_2O-LiBr solution is pumped to the low pressure generator1 by passing through the heat exchanger1 which increases solution temperature. Some part of H_2O evaporates in the low pressure generator1 and goes directly to high pressure absorber2. At the high pressure absorber2 low concentrated H_2O-LiBr is formed and pumped to the high pressure generator2 by passing through the heat exchanger2. The cycle starts once again in the generator2.

Table 1.The mass, energy, and exergy equations for each component and for overall cycle [11, 12].

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 exchanger1	$\dot{m}_2 = \dot{m}_3$ $\dot{m}_4 = \dot{m}_5$	$\dot{m}_2 h_2 + \dot{m}_4 h_4 = \dot{m}_3 h_3 + \dot{m}_5 h_5$	$E_3 = \dot{m}_3(h_3 - h_0 - T_0(s_3 - s_0))$ $E_4 = \dot{m}_4(h_4 - h_0 - T_0(s_4 - s_0))$ $E_5 = \dot{m}_5(h_5 - h_0 - T_0(s_5 - s_0))$
Generator1	$\dot{m}_3 = \dot{m}_4 + \dot{m}_7$	$\dot{m}_3 h_3 + Q_{G1}$ $= \dot{m}_4 h_4 + \dot{m}_7 h_7$	$E_7 = \dot{m}_7(h_7 - h_0 - T_0(s_7 - s_0))$
Expansion Valve1	$\dot{m}_5 = \dot{m}_6$	$\dot{m}_5 h_5 = \dot{m}_6 h_6$	$E_6 = \dot{m}_6(h_6 - h_0 - T_0(s_6 - s_0))$
Absorber1	$\dot{m}_6 + \dot{m}_{17} = \dot{m}_1$	$Q_{A1} = \dot{m}_{17} h_{17} + \dot{m}_6 h_6 - \dot{m}_1 h_1$	$E_{17} = \dot{m}_{17}(h_{17} - h_0 - T_0(s_{17} - s_0))$
Absorber2	$\dot{m}_7 + \dot{m}_{13} = \dot{m}_8$	$Q_{A2} = \dot{m}_{13} h_{13} + \dot{m}_7 h_7 - \dot{m}_8 h_8$	$E_{13} = \dot{m}_{13}(h_{13} - h_0 - T_0(s_{13} - s_0))$ $E_8 = \dot{m}_8(h_8 - h_0 - T_0(s_8 - s_0))$
Pump2	$\dot{m}_8 = \dot{m}_9$	$W_{P2} = \dot{m}_8(h_9 - h_8)$	$E_9 = \dot{m}_9(h_9 - h_0 - T_0(s_9 - s_0))$
Heat exchanger2	$\dot{m}_9 = \dot{m}_{10}$ $\dot{m}_{11} = \dot{m}_{12}$	$\dot{m}_9 h_9 + \dot{m}_{11} h_{11} = \dot{m}_{10} h_{10} + \dot{m}_{12} h_{12}$	$E_{10} = \dot{m}_{10}(h_{10} - h_0 - T_0(s_{10} - s_0))$ $E_{11} = \dot{m}_{11}(h_{11} - h_0 - T_0(s_{11} - s_0))$ $E_{12} = \dot{m}_{12}(h_{12} - h_0 - T_0(s_{12} - s_0))$

Generator2	$\dot{m}_{10} = \dot{m}_{14} + \dot{m}_{11}$	$\dot{m}_{10}h_{10} + Q_{G2} = \dot{m}_{14}h_{14} + \dot{m}_{11}h_{11}$	$E_{14} = \dot{m}_{14}(h_{14} - h_0 - T_0(s_{14} - s_0))$
Expansion Valve2	$\dot{m}_{12} = \dot{m}_{13}$	$\dot{m}_{12}h_{12} = \dot{m}_{13}h_{13}$	
Condenser	$\dot{m}_{14} = \dot{m}_{15}$	$\dot{m}_{14}h_{14} = \dot{m}_{15}h_{15} + Q_C$	$E_{15} = \dot{m}_{15}(h_{15} - h_0 - T_0(s_{15} - s_0))$
Expansion Valve3	$\dot{m}_{15} = \dot{m}_{16}$	$\dot{m}_{15}h_{15} = \dot{m}_{16}h_{16}$	$E_{16} = \dot{m}_{16}(h_{16} - h_0 - T_0(s_{16} - s_0))$
Evaporator	$\dot{m}_{16} = \dot{m}_{17}$	$\dot{m}_{16}h_{16} + Q_E = \dot{m}_{17}h_{17}$	$E_{17} = \dot{m}_{17}(h_{17} - h_0 - T_0(s_{17} - s_0))$
Overall cycle		$(\dot{Q}_{A,TOT} + \dot{Q}_C)_{outletenergy}$ $= (Q_{G,TOT} + W_{P,TOT} + \dot{Q}_E)_{inletenergy}$ $COP = \dot{Q}_E/(W_{P,TOT} + \dot{Q}_{G,TOT})$ $W_P = \dot{m}_{in}(h_{in} - h_{out}) = \Delta P/\rho$ $\dot{E} = Q\left(1 - \dfrac{T_0}{T}\right)$ $\dot{E}_{D,G1} = \dot{E}_3 + \dot{E}_{G1} - \dot{E}_7 - \dot{E}_4$	$\dot{E}_{D,C} = \dot{E}_{14} - \dot{E}_{15} - \dot{E}_C$ $\dot{E}_{D,HE1} = \dot{E}_2 + \dot{E}_4 - \dot{E}_3 - \dot{E}_5$ $\dot{E}_{D,HE2} = \dot{E}_9 + \dot{E}_{11} - \dot{E}_{10} - \dot{E}_{12}$ $\dot{E}_{D,E} = \dot{E}_{16} + \dot{E}_E - \dot{E}_{17}$ $\dot{E}_{D,EXV} = \dot{E}_{in} + \dot{E}_{out}$ $\dot{E}_{D,G2} = \dot{E}_{10} + \dot{E}_{G2} - \dot{E}_{14} - \dot{E}_{11}$ $ECOP = \dot{E}_E/(W_{P,tot} + \dot{E}_{G,TOT})$

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, 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, refrigerant leaving the evaporator is saturated vapor at evaporator pressure, direct heat transfer from the components to the surroundings is negligible.

In this study the thermodynamic analysis of the cycles which is introduced above is done and the thermodynamic and the mathematical modeling are explained as follows. The chemical exergy of the streams are not taken into calculation because, there is no mass inlet or outlet of the cycle. The physical exergy of the streams is taken as the total exergy. The equations of the calculation of the cycle are given in Table 1 for each component and for overall cycle.

RESULTS AND DISCUSSION
A computer program written by the author in FORTRAN codes is used to calculate the enthalpy and entropy values of the streams.

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

Stream Nu.	Fluid	Pressure kPa	Temperature ^{0}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 LiBr/H$_2$O	1.0	32	52.3	2.13	69.01	0.2207	20.6
2	Weak LiBr/H$_2$O	2.1	32	52.3	2.13	69.01	0.2205	20.7
3	Weak LiBr/H$_2$O	2.1	48.2	52.3	2.13	103.63	0.3291	25.5
4	Strong LiBr/H$_2$O	2.1	49.8	55	2.03	113.98	0.3191	51.4
5	Strong LiBr/H$_2$O	2.1	32.2	55	2.03	77.58	0.2058	46.1
6	Strong LiBr/H$_2$O	1.0	32.2	55	2.03	77.58	0.206	46

7	H_2O	2.1	49	0	0.1	2590	8.095	18.3
8	Weak LiBr/H_2O	2.1	32	43.5	2.13	61.61	0.2899	-39.1
9	Weak LiBr/H_2O	4.81	32	43.5	2.13	61.61	0. 2897	-39
10	Weak LiBr/H_2O	4.81	48.3	43.5	2.13	99.82	0.4117	-35.1
11	Strong LiBr/H_2O	4.81	49.8	45.7	2.03	102.12	0.3997	-21.5
12	Strong LiBr/H_2O	4.81	32.2	45.7	2.03	61.79	0.2719	-26
13	Strong LiBr/H_2O	2.1	32.2	45.7	2.03	61.79	0.2721	-26.1
14	H_2O	4.81	49.1	0	0.1	2591	8.51	6.1
15	H_2O	4.81	32.2	0	0.1	134.8	0.47	0.1
16	H_2O	1.0	7.0	0	0.1	134.8	0.4703	0.1
17	H_2O	1.0	7.0	0	0.1	2514.2	8.97	-15.4

Absorber1 heat energy- exergy destruction	Q_{A1}= 261.9 kW, $E_{A1}=E_{D,A1}+E_{A1,loss}$=10 kW
Heat Exchanger1--exergy destruction, Exergy efficiency	$E_{D,HE1}$=0.5 kW
Generator1 heat energy- exergy	Q_{G1}= 269.7 kW, E_{G1}= 44.2 kW
Absorber2 heat energy- exergy destruction	Q_{A2}= 253 kW, $E_{A2}=E_{D,A2}+E_{A2,loss}$=31.3 kW
Pump1-2 work energy	$W_{P1}=W_{P2}$= 0.1 kW
Heat Exchanger2--exergy destruction, Exergy efficiency	$E_{D,HE2}$=0.6 kW
Generator2 heat energy- exergy	Q_{G2}= 253.8 kW, E_{G2}= 19.7 kW
Condenser heat energy-exergy--exergy destruction	Q_C=246 kW, $E_C=E_{C,Lost}+E_{D,C}$ =6 kW
Expansion valve1,2,3 exergy destruction	$E_{D,EXV1}$=0.1 kW, $E_{D,EXV2}$=0.1 kW, $E_{D,EXV3}$=0.01 kW
Evaporator heat energy- exergy	Q_E= 238 kW, E_E= 15.3 kW, $E_{D,E}$= 31.6 kW
COP	0.45
ECOP	0.24
Inlet Energy=Outlet Energy → $(W_{P,TOT}+Q_{G,TOT}+Q_E=Q_{A,TOT}+Q_C)$ → (0.21+523.5+238=514.9+246)→761.6≈760.9	
Overall Cycle(inlet exergy $(E_{OC}=E_{G,TOT}+W_{P,TOT}+E_E$=63.9+0.21+15.3=79.4)	
Overall Cycle(inlet exergy=outlet exergy=Lost + Destructed)	
$(E_{D,OC}+E_{Loss,OC}=E_C+E_{A1}+E_{A2}+E_{D,HE,TOT}+E_{D,E}+E_{D,EXV,TOT}$=6+10+31.3+1.1+31.6+0.2=80.2)	
inlet exergy=outlet exergy →79.4≈80.2)	

To calculate the enthalpy and the entropy values of the streams the equations used in the program are taken from the reference [13, 14]. However, for 50 % concentration H_2O-LiBr, the reference state values are taken at 100 kPa pressure, 25 ^0C temperature, and as h_0=49.2 kJ/kg and s_0=0.1867 kJ/kgK for the mixture of H_2O/LiBr.

In Table 2 for each stream of the half effect cycle, 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, COP, ECOP, the exergy and the energy balance are given.

As can be seen that for the overall half effect cycle the total destructed and lost exergy is about 80 kW, and the 31.6 kW of it is destructed at the evaporator that means 40 % of the total destructed exergy. The total destructed and lost exergy in the two absorber of the half effect cycle is found as 41.3 kW, and that is 52 % of the total destructed exergy. For the half effect cycle in the evaporator and in the absorber 92 % of the total exergy is destructed and lost. That means the efficiency and the design of the absorber and the evaporator is very important for the cycle. Better design

of these two components will directly affect and improve the working conditions and the performance of the overall cycle.

The destructed and lost exergy of the condenser of the half effect cycle is about 6 kW and that is about 8 % of the total lost and destructed exergy. For the cycle the irreversibility in the heat exchangers, in the expansion valves and in the pumps are small, but they are taken into calculation.

For a refrigeration system, the coefficient performance (COP) and the exergetic coefficient performance (ECOP) are the most important thing for evaluation and consideration. The coefficient performance (COP) of the half effect cycle is 0.45 and the exergetic coefficient performance (ECOP) of the cycle is 0.24. The operating temperature of the half effect cycle is about 50 ^{0}C. These results showed that the half effect systems are very appropriate for using low temperature heat energy in cooling. 1 % Error is happened in all this calculation of the half effect cycle which can be ignored. These results are in good agreement with the literature.

CONCLUSION

Absorption cooling cycles can use solar or waste heat for cooling with very small electric power. These cycles are environmental and they can decrease the costs of cooling. This work presents exergy analysis of a half effect absorption system. For the evaluation of the thermodynamic properties of lithium bromide-water solutions, a computer program is developed by the author in FORTRAN codes for the exergy analysis. The main advantage of the half effect absorption systems is that low temperature heat energy or waste heat energy can be used. The coefficient performance (COP) and the exergetic coefficient performance (ECOP) of the half effect absorption system are found as 0.45 and 0.24, respectively. For each component the exergy loss is calculated. 92 % of the total irreversibilities are found in the absorber and in the evaporator. It is concluded that better design and efficiencies of the evaporator and the absorber is very important for the half effect absorption cycles. Improving the design of these two components will directly improve and affect the working conditions and the performance of the cycle.

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