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Thermoeconomic Analysis of a Single and Double-Effect LiBr/H_2O Absorption Refrigeration System*

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Abstract

The aim of this work is to carry out a thermoeconomic analysis of a single and double-effect LiBr/H_2O absorption refrigeration system. The methodology of functional analysis with negentropy is used. The exergetic cost of the main product, the cooling cost, was calculated as a function of the exergy of the heat source. Two cases were analyzed for each system: the first considers a direct-fired system while the second considers a hot-water driven system for the single-effect system and a steam-driven system for the double effect system as part of a cogeneration system. As expected, the resultant exergetic cost of the main product was higher for the direct-fired system.

Keywords: *Absorption refrigeration, exergy, thermoeconomic evaluation, lithium bromide.*

1. Introduction

Absorption refrigeration systems differ from compression systems by the use of a heat source as the energy input in order to operate; conversely, compression-based systems require mechanical energy to operate. Thus the main advantage of the absorption systems is that they can run burning a fuel or using waste heat recovered from other thermal systems. Moreover, these systems present other advantages, such as high reliability, low maintainability and a silent and vibration-free operation (New Buildings Institute, 1998). Another important aspect is the elimination of CFC and HCFC refrigerants.

Single-effect absorption refrigeration systems have only one heating level of the working fluid (dilute solution). The coefficient of performance (COP) of these systems, working with a LiBr/H_2O solution, is in the range of 0.6 to 0.7 (Dorgan et al. 1995).

The components of this system are mainly heat exchangers (see Figure 1): condenser, evaporator, generator (or desorber), absorber, and solution heat exchanger. The heat enters the system at the generator (energy input) and at the evaporator (where heat is removed from a water flow from point 17 to 18 in Figure 1, producing the cooling effect). In the condenser and absorber, heat is removed by water flowing from a cooling tower (water flow from point 15 to 16 at the condenser and from point 13 to 14 at the absorber in Figure 1).

Double-effect absorption refrigeration systems have two stages of vapor generation to separate the refrigerant from the absorbent. The heat transfer occurs at a higher temperature compared to the single-effect cycle.

The hot fluid temperature input for double-effect chiller in HVAC systems is usually around 180 ºC (ASHRAE, 2002), but it can be lower depending on evaporator and absorber/condenser temperatures. According Srikhirin et al. (2001) the temperature for the heat source in these systems is within the range of 120 – 150°C.

According to the literature, the COP value of the double-effect LiBr/H_2O system is within the range of 1.0 to 1.2 (Herold et al., 1996), so the double-effect units could be more competitive than the single-effect units, depending on the operational conditions.

The capacity of the absorption refrigeration system considered in this work is 316 kW (90 TR). This capacity was chosen in order to analyze the refrigeration system to be installed in a university hospital in Campinas, Brazil.

Thermoeconomic analysis is used because it is considered a useful tool, which combines thermodynamics (second law) with economic principles (Bejan et al. 1996).

The base of the thermoeconomic theory is the consideration of exergy as the objective measure of the thermodynamic and economic values of an energy carrier. Hence, exergy will herein be considered as the base for the process of cost attribution.

In exergetic analysis, techniques are developed with the aim of evaluating the thermodynamic inefficiencies in terms of exergy destructions and losses, and also in terms of variations in end-product costs (Bejan et al., 1996).

The exergetic cost analysis is based on accounting for the exergy destruction related to the system design and its operation and maintenance.

For the case of the absorption refrigeration system the thermoeconomic functional analysis was adopted (Frangopoulos, 1983; Frangopoulos, 1987) instead of the exergetic cost theory (Tsatsaronis, 1993; Lozano and Valero, 1993; Gonzales and Nebra, 2004), because the functional analysis considers the exergy variations (differences) while the exergetic cost theory considers the exergy of each flow. This choice was made due to the occurrence of negative exergy flows (as they usually appear in this kind of system), such that the use of the exergetic cost theory would result in a negative cost, which would not have any sense in this case.

2. Functional Analysis with Negentropy

Negentropy is a concept related to thermal systems. This concept was introduced by Frangopoulos (1983) to analyze a power generation system. This concept had the

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aim of taking into account the useful function of dissipative equipments. In this kind of cycle the condenser has the function of removing entropy from the system, which increases along the cycle, or in other words, the condenser supplies negentropy.

Negentropy is introduced in the analysis of productive structures in complex plants with the objective of redistributing the costs of the external irreversibilities. In the analysis of productive structures, negentropy is considered as a fuel for control volumes where entropy flows increase (into it) and it is a product for control volumes where the entropy flows are reduced.

The negentropy concept is used in this analysis due to the presence of dissipative components such as a condenser, absorber and cooling tower.

For the realization of this work, previous analyses of the authors were taken into consideration (Palacios-Bereche et al., 2007a and 2007b), as well as several studies in which functional analysis was applied were reviewed (Alves, 2007; Cerqueira, 1999; Gonzales, 2004; Modesto, 2004; even though they do not deal with refrigeration systems). Moreover, the works of Misra et al. (2002) and (2005) were considered, in which the exergetic cost theory is applied for absorption refrigeration systems (although the chemical exergy was not taken into consideration in their calculations). Also, the work of Accadia and Rossi (1998) presents a thermoeconomic optimization for a compression refrigeration plant. In this work, the exergetic cost theory with negentropy was applied.

3. Functional Analysis for Absorption Refrigeration Systems

Two cases were considered for the analysis of the single-effect system. The first case considers a direct-fired system whilst the second is a hot-water driven system as part of a cogeneration system.

Two cases were also considered for the analysis of the double-effect absorption refrigeration system. The first case considers a direct-fired system while the second case is a steam-driven system. The steam-driven system considers that the absorption refrigeration system uses steam supplied from a cogeneration system. The data used for these cases were from Palacios-Bereche et al. (2007b). The properties of the LiBr-water solutions were taken from Palacios-Bereche et al. (2007a). In the cooling tower, the conditions of the ambient air were adopted as a mean relative humidity of 70% and a temperature of 29 °C. In the thermoeconomic analysis the air exergy was neglected.

Figures 1 and 2 present, respectively, the single and double-effect absorption refrigeration systems. The control volumes adopted for the thermoeconomic analysis are indicated in these figures. Four control volumes were adopted for each case.

To facilitate the comparison with thermo-mechanical systems, this analysis introduces the concept of a “thermal compressor” (Wark, 1995). Thus, according to Figure 1, the thermal compressor is composed of the following units: generator, solution heat exchanger, absorber, expansion valve and solution pump.

In the case of the double-effect system the thermal compressor has in addition a low generator-high condenser and a second solution heat exchanger with a second solution expansion valve and a second solution pump. The numbers 1 and 2 in the figure captions indicate if the component is between the absorber and the low generator (1) or if the component is between the low generator and the high generator (2).

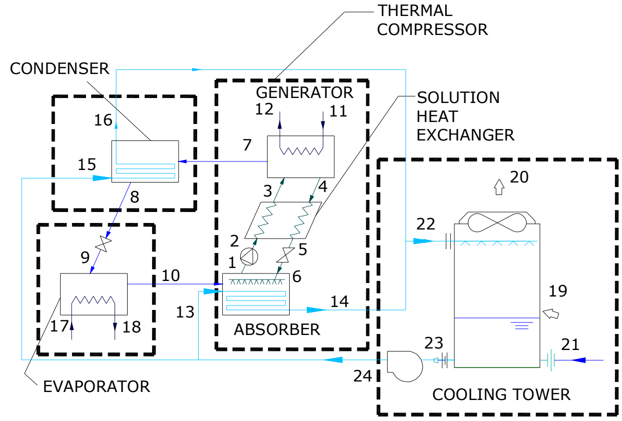


Figure 1. Control Volumes for the Thermoeconomic Functional Analysis – Single-Effect System.

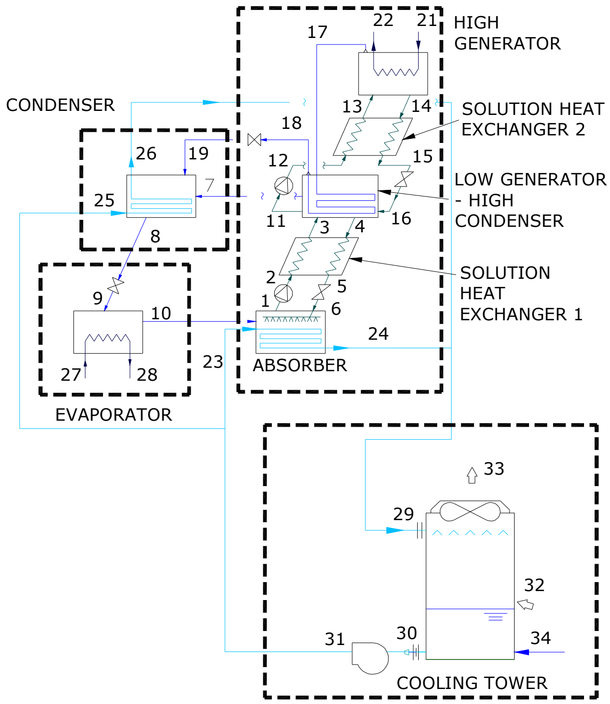


Figure 2. Control Volumes for the Thermoeconomic Functional Analysis – Double-Effect System.

The function of the thermal compressor is to carry the working fluid (the refrigerant) from the low pressure and temperature point (point 10) to the higher pressure and temperature conditions (point 7 in the single-effect and point 19 in the double-effect system).

Due to the thermoeconomic approach adopted, the consideration of the chemical exergy of LiBr – water is not necessary (it was considered in other work of the same authors, Palacios-Bereche et al., 2007b), but care must be taken with the negative exergies of the water flux with pressures lower than atmospheric.

Figure 3 shows the T-s diagram for the water cycle in the single-effect system. Here the function of the thermal compressor can be viewed (process from 10 to 7). Figure 3 shows also the negentropy generation of the system in the processes from 10 to 7 and 7 to 8 (shaded area).

Thus:

$$Total\ negentropy = T_0.(s_{10} - s_7) + T_0.(s_7 - s_8) \qquad (1)$$

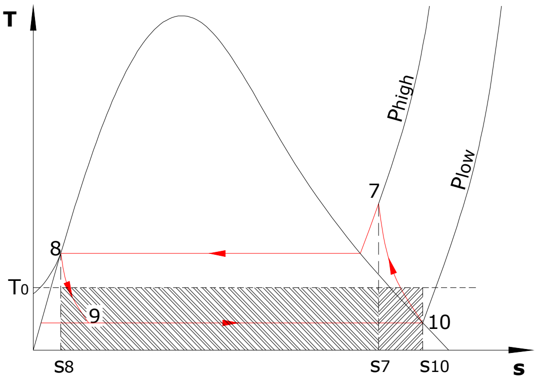


Figure 3. Diagram T-s for the Water Cycle (Single-Effect Cycle).

Table 1 presents the operational conditions for the absorption refrigeration systems according to Palacios-Bereche (2007b) whilst Table 2 presents the flows considered in the functional analysis of the single- and double-effect absorption refrigeration systems.

Table 3 presents the control volumes adopted for the functional analysis (as well as the respective fuels and products for each one).

Figures 4 and 5 show, respectively, the functional diagrams of the single and double-effect systems. Four control volumes and two virtual units were adopted (one exergy distributor and one negentropy distributor). The virtual units are considered as a combination of either an exergy junction and distributor (B) or a negentropy junction and distributor (S).

Control volume 2 is comprised of the evaporator and the expansion valve as a single unit because the latter is a dissipative element whose function is to reduce the pressure in order to close the cycle, and it is difficult to define a product in exergy or negentropy terms.

Control volume 3, namely the thermal compressor, is comprised of the absorber, the solution heat exchanger (1 and 2 in the double-effect cases), the generator, the solution pump (1 and 2) and the solution expansion valve (1 and 2) and is considered as a single unit.

The product of control volume 3 is defined as the difference of exergy between the states 10 and 7 (Figure 1) in the single-effect system. For the double-effect system this unit has as products the flows B710 and B1910 (see Figure 5). These flows are calculated according to Table 1, and represent the exergy gained by the refrigerant in the thermal compressor.

Additionally, the product of control volume 3 is the negentropy generation (S107 in the single-effect system and S1019 in the double-effect system). The fuels of this unit are the exergy supplied for the heat source (B25 or B35 for the direct-fired system and B1112 or B2122 for the hot water and steam-driven system respectively) and the negentropy supplied by the negentropy distributor (S1413 in the single effect system and S2423 in the double effect system). The cooling tower is a negentropy generator (S2224 or S2931) and it has as fuel the exergy supplied by the cooling water pump (BWBAR), the power of the fan (BWVENT) and the water reposition consumed (BARE) by the cooling tower.

Table 1. Operational Conditions in the Single-effect Absorption Refrigeration System.

Pt.	h	m	P	T	S	Ex_{ph}	Ex_q	Ex_{to}
	[kJ/kg]	[kg/s]	[kPa]	[ºC]	[kJ/kg-K]	[kW]	[kW]	[kW]
Single-effect system								
1	84.7	1.8	0.823	33.8	0.207	0.296	904.9	905.2
2	84.8	1.8	7.91	33.8	0.207	0.306	904.9	905.2
3	144.2	1.8	7.91	63.1	0.392	8.051	904.9	912.9
4	205.6	1.6647	7.91	85.7	0.486	17.36	939.9	957.3
5	141.4	1.6647	7.91	52.5	0.298	3.641	939.9	943.6
6	141.4	1.6647	0.823	43.3	0.303	0.971	939.9	940.9
7	2641.6	0.14	7.91	76	8.430	17.98	6.759	24.74
8	173	0.14	7.91	41.3	0.590	0.2307	6.759	6.989
9	173	0.14	0.823	4.2	0.624	-1.161	6.759	5.598
10	2508.2	0.14	0.823	4.2	9.045	-24.9	6.759	-18.14
11	419.3	13.6	424.7	100	1.307	466.4	679.4	1146
12	386.9	13.6	145.2	92.4	1.220	378.3	679.4	1058
13	121.9	23	424.7	29	0.423	10.02	1149	1159
14	140.2	23	145.2	33.4	0.484	12.32	1149	1161
15	121.9	23	424.7	29	0.423	10.02	1149	1159
16	136.4	23	145.2	32.5	0.471	10.01	1149	1159
17	50.8	13.46	424.7	12	0.180	20.8	672.4	693.2
18	27.3	13.46	145.2	6.5	0.098	34.48	672.4	706.9
19	74.4	32.1	101.3	29	5.87	--	--	2.91
20	99.6	32.4	101.3	30	5.95	--	--	22.39
21	121.6	0.2975	101.3	29	0.423	0.0331	14.86	14.9
22	138.3	46	101.3	33	0.478	20.29	2298	2318
24	122.1	46	424.7	29.1	0.423	20.19	2298	2318
25	8.3	0.0124	101.3	29	10.83	0.001	635.5	635.6
Double-effect system								
1	86.7	1.764	0.869	34.7	0.213	0.387	887.2	887.6
2	86.7	1.764	4.99	34.7	0.213	0.394	887.2	887.6
3	123.5	1.764	4.99	52.9	0.329	4.233	887.2	891.4
4	187	1.63	4.99	76.0	0.432	12.120	922	934.1
5	147.2	1.63	4.99	55.4	0.314	4.347	922	926.4
6	147.2	1.63	0.869	44.5	0.320	1.588	922	923.6
7	2623.9	0.06	4.99	66.4	8.59	4.032	3.003	7.035
8	137.5	0.13	4.99	32.83	0.475	0.043	6.687	6.73
9	137.5	0.13	0.87	4.95	0.495	-0.744	6.687	5.943
10	2509.6	0.13	0.87	4.95	9.03	-23.64	6.687	-16.95
11	151.1	0.972	4.985	66.39	0.4116	5.125	488.8	493.9
12	151.1	0.972	67.703	66.4	0.4116	5.185	488.8	494
13	219.8	0.972	67.703	99.43	0.6047	16.02	488.8	504.8
14	316	0.898	67.703	141.49	0.7703	31.95	508	539.9
15	241.7	0.898	67.703	103.95	0.5824	15.5	508	523.5
16	241.7	0.898	4.985	78.54	0.5949	12.15	508	520.1
17	2740	0.07	67.7	130.4	7.704	33	3.684	36.69
18	373.1	0.07	67.7	89.1	1.18	1.862	3.684	5.546
19	373.1	0.07	4.99	32.8	1.25	0.468	3.684	4.153
21	2746.4	0.13	475.7	150	6.84	91.71	6.434	98.15
22	632.3	0.13	475.7	150	1.84	11.29	6.434	17.72
23	121.7	14.5	290.7	29	0.423	4.367	724.4	728.8
24	150.7	14.5	123.4	35.9	0.518	12.22	724.4	736.6
25	121.7	18	290.7	29	0.423	5.119	899.2	904.4
26	130.8	18	123.4	31.21	0.453	5.197	899.2	904.4
27	50.6	13.5	290.7	12	0.180	18.99	672.4	691.4
28	26.9	13.5	123.4	6.38	0.097	34.48	672.4	706.9
29	139.7	32.5	101.3	33.3	0.482	15.52	1624	1639
30	121.6	32.5	101.3	29	0.423	3.616	1624	1627
31	121.9	32.5	290.7	29	0.423	9.85	1624	1633
32	74.4	23.0	101.3	29	5.87	--	--	2.129
33	101.6	23.3	101.3	30.4	5.96	--	--	18.07
34	121.6	0.23	101.3	29	0.423	0.025	11.41	11.43
35	8.3	0.008	101.3	29	10.83	0	393.3	393.3
36	74.4	0.149	101.3	29	5.87	0.0138	0	0.0138
37	366.9	0.156	101.3	610	8.44	48.68	10.22	58.9

Table 2. Flows for the Thermoeconomic Analysis in the Single and Double-effect Absorption Refrigeration Systems.

Nature	Single-Effect System	Double-Effect System
Exergy added to water flow in the evaporator	$B1817 = Ex_{18} - Ex_{17}$	$B2827 = Ex_{28} - Ex_{27}$
Negentropy generated in the condenser	$S78 = \dot{m}_7.T_0.(s_7 - s_8)$	$S78 = \dot{m}_7.T_0.(s_7 - s_8)$
Negentropy generated in the condenser	-	$S198 = \dot{m}_{19}.T_0.(s_{19} - s_8)$
Negentropy generated in the thermal compressor	$S107 = \dot{m}_{10}.T_0.(s_{10} - s_7)$	$S107 = \dot{m}_{10}.T_0.(s_{10} - s_7)$
Negentropy generated in the thermal compressor	-	$S1019 = \dot{m}_{19}.T_0.(s_{10} - s_{19})$
Negentropy produced in the cooling tower	$S2224 = \dot{m}_{22}.T_0.(s_{22} - s_{24})$	$S2931 = \dot{m}_{29}.T_0.(s_{29} - s_{31})$
Exergy consumed in the cooling tower	$B2224 = Ex_{22} - Ex_{24}$	$B2931 = Ex_{29} - Ex_{31}$
Exergy added to the refrigerant in the thermal compressor	$B710 = Ex_7 - Ex_{10}$	$B710 = \dot{m}_7.(ex_7 - s_{10})$
Exergy added to the refrigerant in the thermal compressor	-	$B1910 = \dot{m}_{19}.(ex_{19} - ex_{10})$
Exergy consumed in the condenser	$B78 = Ex_7 - Ex_8$	$B78 = \dot{m}_7.(ex_7 - ex_8)$
Exergy consumed in the condenser	-	$B198 = \dot{m}_{19}.(ex_{19} - ex_8)$
Exergy consumed in the evaporator	$B810 = Ex_8 - Ex_{10}$	$B810 = Ex_8 - Ex_{10}$
Negentropy consumed in the condenser	$S1615 = \dot{m}_{16}.T_0.(s_{16} - s_{15})$	$S2625 = \dot{m}_{16}.T_0(s_{26} - s_{25})$
Negentropy consumed in the thermal compressor	$S1413 = \dot{m}_{14}.T_0.(s_{14} - s_{13})$	$S2423 = \dot{m}_{14}.T_0(s_{24} - s_{23})$
Negentropy consumed in the evaporator	$S108 = \dot{m}_{10}.T_0.(s_{10} - s_8)$	$S108 = \dot{m}_{10}.T_0(s_{10} - s_8)$
Exergy added in the thermal compressor – Hot water driven system	$B1112 = Ex_{11} - Ex_{12}$	-
Exergy added in the thermal compressor– Steam driven system	-	$B2122 = Ex_{21} - Ex_{22}$
Exergy added in the thermal compressor – Direct fired system	$B25 = Ex_{25}$	$B35 = Ex_{35}$
Power consumed by the solution pump	$BWbs = \dot{W}_{b\_sol}$	$BWbs1 = \dot{W}_{b\_sol\ 1}$
Power consumed by the solution pump	-	$BWbs2 = \dot{W}_{b\_sol\ 2}$
Power consumed by the cooling tower fan	$BWVENT = \dot{W}_{vent}$	$BWVENT = \dot{W}_{vent}$
Power consumed by the water cooling pump	$BWBAR = \dot{W}_{b\_AR}$	$BWBAR = \dot{W}_{b-AR}$
Exergy of reposition water	$BARE = Ex_{21}$	$BARE = Ex_{34}$

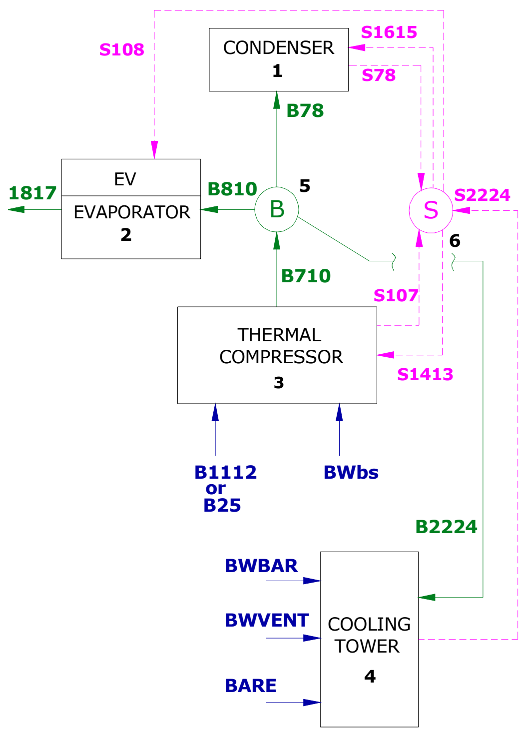


Figure 4. Productive Structure for the Functional Analysis (Single-effect Cycle).

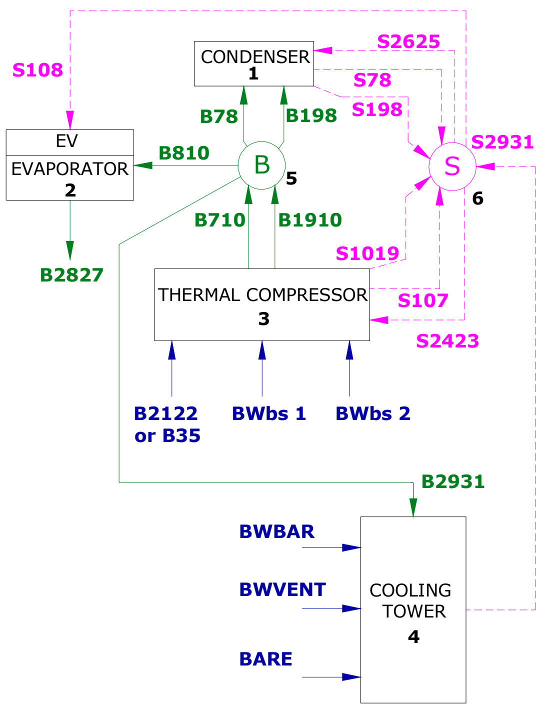


Figure 5. Productive Structure for the Functional Analysis (Double-effect Cycle).

Table 3. Table of Fuels and Products for the Functional Analysis.

Control volume	Products	Fuels
Single-effect system		
1. Condenser	*S78*	*B78 + S1615*
2. Evaporator + Expansion valve	*B1817*	*B810 + S108*
3.Thermal compressor-Hot water-driven system	*B710 + S107*	*B1112 + BWbs + S1413*
3.Thermal compressor–Direct-fired system	*B710 + S107*	*B25 + BWbs + S1413*
4.Cooling Tower	*S2224*	*B2224 + BWBAR + BWVENT+BARE*
Double-effect system		
1. Condenser	*S78 + S198*	*B78 + B198 + S2625*
2. Evaporator + Expansion valve	*B2827*	*B810 + S108*
3. Thermal compressor – Steam-driven system	*B710 + B1910 + S1019 + S107*	*B2122 + BWbs1 + BWbs1 + S2423*
3. Thermal compressor – Direct-fired system	*B710 + B1910 + S1019 + S107*	*B35 + BWbs1 + BWbs2 + S2423*
4. Cooling tower	*S2931*	*B2931 + BWBAR + BWVENT+BARE*

4. Functional Analysis for the Cogeneration System

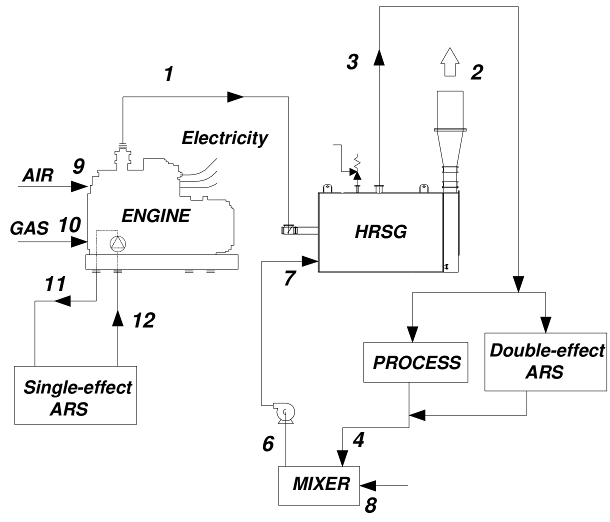
In order to analyze the cases of the absorption refrigeration systems driven by hot water and by steam, a thermoeconomic functional analysis for the cogeneration system should be done.

The cogeneration system considered in this study is comprised of an internal combustion engine, a heat recovery steam generator (HRSG), a mixer tank, and a process.

The internal combustion engine has a 425 kW nominal capacity and the fuel supplied is natural gas. The water used for cooling the jacket and auxiliary circuit is used later as a thermal energy source in the single-effect absorption refrigeration $LiBr/H_2O$ system. The engine exhaust gas has a high quality of thermal energy and will be used later in the HRSG. The steam produced in the HRSG will be used for driving the double-effect absorption refrigeration system which is considered within the process block in Figure 6. While Figure 6 shows the cogeneration system considered in this study, Table 4 shows the operational conditions.

Figure 7 shows the functional diagram for the cogeneration system. In this case five control volumes and three virtual units were adopted. These virtual units are respectively junction-distributors of electrical power (W), exergy (B) and negentropy (S). Table 5 shows the control volumes adopted for the functional analysis (as well as the respective fuels and products considered for each one). The results of the functional analysis are presented in Table 5.

From the thermoeconomic analysis the unit exergetic costs of the hot water (k_{TP1}) and the steam (k_{TP2}) can be obtained. In this way the analysis of the absorption refrigeration systems described in the previous items can be done.



ARS : Absorption refrigeration system

Figure 6. Cogeneration System Considered.

Table 4. Operational Conditions in the Cogeneration System

Pt.	Flow	m	T	P
		kg/s	°C	kPa
1	Exhaust gas	1,094	645	121,6
2	Exhaust gas	1,094	212	121,6
3	Steam	0,1348	179,1	981
4	Water	0,1078	70	200
6	Water	0,1348	61	200
7	Water	0,1348	65,2	981
8	Water	0,0270	29	101,3
9	Air	0,6532	29	101,3
10	Natural gas	0,0348	29	101,3
11	Water	13,6	100	424,7
12	Water	13,6	100	145,2

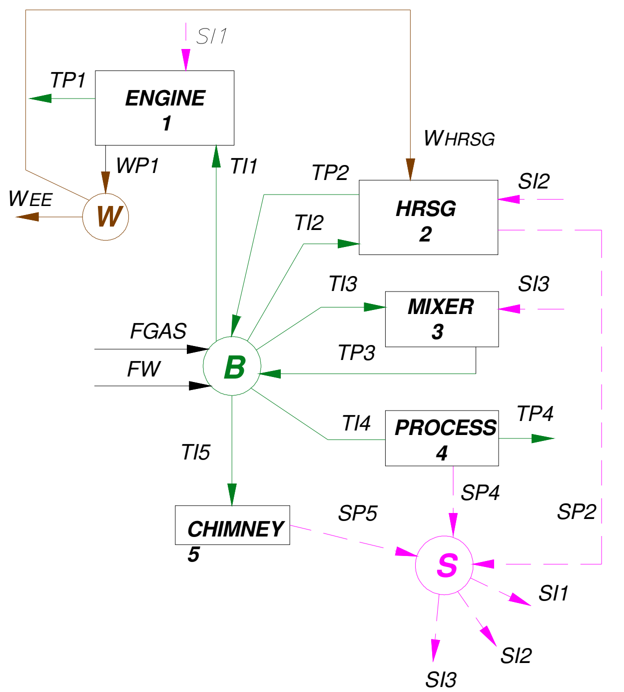


Figure 7. Productive Sstructure for the Functional Analysis of the Cogeneration System.

Table 5. Table of Fuels and Products for the Functional Analysis – Cogeneration System.

Control volume	Fuel	Product
Engine	TI1+ SI1	TP1 + WP1
HRSG	TI2 + SI2 + WHRSG	TP2 + SP2
Mixer	TI3 + SI3	TP3
Process	TI4	TP4 + SP4
Chimney	TI5	SP5
Exergy junction	FGAS + FW + TP2 + TP3	TI1 + TI2 + TI3 + TI4 + TI5
Electrical Power junction	WP1	WEE + WHRSG
Negentropy junction	SP2 + SP4 + SP5	SI1 + SI2 + SI3

Table 6. Unit Exergetic Cost in the Cogeneration System.

Flow	Equation	E [kW]	k
TI1	TI1=Ex10-Ex1	1275	1.012
TI2	TI2=Ex1-Ex2	351.3	1.012
TI3	TI3=Ex8+Ex4	19.96	1.012
TI4	TI4=Ex3-Ex4	116.8	1.012
TI5	TI5=Ex2	161.3	1.012
TP1	TP1=Ex11-Ex12	88.02	1.012
TP2	TP2=Ex3-Ex7	117.1	1.176
TP3	TP3=Ex7	18.02	1.182
TP4	TP4=η_C.Qproc	70.71	1.972
SI1	SI1=S1-S9- S10 + (S11-S12)	760.9	0.5758
SI2	SI2=S3-S7	49.64	0.5758
SI3	SI3=S7-S4-S8	1.889	0.5758
SP2	SP2=S1-S2	249.5	1.026
SP4	SP4=S3-S4	47.25	1.026
SP5	SP5=S2	159.2	1.026
W_{P1}	W_{ENGINE}	425	3.859
W_{EE}	W_{EE}	422.5	3.859
W_{HRSG}	W_{HRSG}	2.453	3.859
F_G	F_G=Ex10	1788	1
F_W	F_W=Ex8	1.557	1

5. Exergetic Cost Calculation

The unit exergetic cost of a flow indicates the number of exergetic units required to produce one exergetic unit of the flow taken into consideration. The unit exergetic cost can be calculated by the following equation:

$$k = \frac{E^*}{Ex} \tag{2}$$

From this definition it can be noticed that a value of $k>1$ is found in exergy flows that are produced from others in the productive structure. Due to irreversibilities, the exergy content in a product is lower than the exergy used to produce it (fuel). In the case of external fuels where additional exergy was not necessary to produce them, the exergetic cost is considered equal to the exergy content of the fuel. Consequently $k=1$. As negentropy is a concept used to redistribute costs, their unit exergetic cost can have values less or greater than one.

The equations considered to calculate the exergetic cost of each flow follow these considerations:

- Balance of exergetic cost in a control volume

$$\sum E^*_{in,j} = \sum E^*_{out,j} \tag{3}$$

- Applying optimality conditions Frangopoulos (1983) determined that the flows that exit the same junctions have the same unit exergetic cost. Thus for single–effect systems

$$k_{B810} = k_{B78} = k_{B2224} \tag{4}$$

$$k_{S1615} = k_{S1413} = k_{S108} \tag{5}$$

and for double-effect systems

$$k_{B78} = k_{B198} = k_{B810} = k_{B2931} \tag{6}$$

$$k_{S108} = k_{S2625} = k_{S2423} \tag{7}$$

Moreover, other assumptions were considered in relation to the negentropy flows in the condenser and in the thermal compressor. For the single-effect system

$$k_{S78} = k_{S107} \tag{8}$$

and for the double-effect system

$$\mathrm{k}_{S107} = \mathrm{k}_{S1019} = \mathrm{k}_{S198} = \mathrm{k}_{S78} \tag{9}$$

- And for the exergetic products of the thermal compressor:

$$k_{B710} = k_{B1910} \tag{10}$$

- Direct-fired systems: Fuels entering from the outside have a unit exergetic cost of 1 (k=1) due to the approach adopted in these cases which analyzes these systems in an isolated way.
- Hot water-driven system: The unit exergetic costs for hot water ($k_{B1112} = k_{TP1}$) and electricity (k_{EE}) were adopted from the cogeneration system analyzed in the prior item, as follows.

$$k_{B1112} = 1.012$$

$$k_{EE} = 3.859$$

- Steam-driven system: The unit exergetic costs for steam ($k_{B2122} = k_{TP2}$) and electricity (k_{EE}) were obtained from the prior item as follows.

$$k_{B2122} = 1.176$$

$$k_{EE} = 3.859$$

$$k_{EE} = k_{BWAR} = k_{BWVENT} = k_{BWbs1} = k_{BWbs2} \tag{11}$$

6. Results

The results of the thermoeconomic analysis are presented in Tables 7 and 8. The exergetic costs (E*) and unit exergetic costs are presented in these tables.

The results show that, as expected, the exergetic cost of the main product of the system (B1817 in the single-effect system and B2827 in the double-effect system) is higher in the direct-fired case. On the other hand, the unit exergetic costs of hot water and electricity in the second case are higher than unity because the exergy to produce them in the cogeneration plant is being considered here.

Concerning the single-effect system the results of this work can be compared with those of Gonzales and Nebra (2005), who applied functional analysis for a single-effect LiBr/H2O absorption refrigeration system in a cogeneration plant. However the unit exergetic cost of the cooling effect was lower for these authors (Gonzales and Nebra, 2005) (k_{Qcold} = 2.431) due to several differences in the calculations. For example:

Table 7. Exergetic Costs and Unit Exergetic Costs of the Single-effect Absorption Refrigeration System: Direct-fired and Hot Water-driven System.

		Direct-fired system		Hot-water driven system	
Flow	***E***		***E******		***E******
	kW	***k***	**kW**	***k***	**kW**
B1817	**13.68**	**50.33**	**688.5**	**18.37**	**251.3**
S78	316.2	1.709	540.4	0.666	210.7
S107	24.82	1.709	42.4	0.666	16.5
S2224	744.7	0.07126	53.1	0.219	162.7
B710	42.88	19.52	837	5.181	222.2
B78	17.75	19.52	346.5	5.169	91.7
B810	25.13	19.52	490.5	5.169	129.9
S1615	334	0.5809	194	0.356	119.0
S1413	419.8	0.5809	243.9	0.356	149.5
S108	341.1	0.5809	198.1	0.356	121.5
B1112	**88.02**	**--**	**--**	**1.012**	**89.1**
B25	**635.6**	**1**	**635.6**	**--**	**--**
BWbs	**0.01331**	**1**	**0.0133**	**3.859**	**0.1**
BWVENT	**13.28**	**1**	**13.3**	**3.859**	**51.2**
BWBAR	**24.89**	**1**	**24.9**	**3.859**	**96.1**
BARE	**14.9**	**1**	**14.9**	**1**	**14.9**

Table 8. Exergetic Costs and Unit Exergetic Costs in the Double-effect Absorption Refrigeration System.

		Direct-fired system		Steam-driven system	
Flow	***E***	***k***	***E******	***k***	***E******
	kW		**kW**		**kW**
B2827	**15.49**	**27.42**	**424.7**	**11.86**	**183.7**
S78	145.4	0.6643	96.6	0.371	54.0
S107	7.776	0.6643	5.2	0.371	2.9
S2931	572.1	0.0547	31.3	0.194	110.9
S198	16.93	0.6643	11.2	0.371	6.3
S1019	171.1	0.6643	113.7	0.371	63.5
B710	14.65	13.87	203.2	4.829	70.7
B1910	13.49	13.87	187.1	4.829	65.1
B78	4.013	13.87	55.7	4.019	16.1
B198	0.4448	13.87	6.2	4.019	1.8
B810	23.68	13.87	328.4	4.019	95.2
S2625	163.3	0.2817	46	0.260	42.4
S2423	411.2	0.2817	115.8	0.260	106.7
S108	341.2	0.2817	96.1	0.260	88.5
B2122	**81.05**	**--**	**--**	**1.176**	**95.3**
B35	**393.3**	**1**	**393.3**	**--**	**--**
BWbs1	**0.00757**	**1**	**0.0076**	**3.859**	**0.0**
BWbs2	**0.06424**	**1**	**0.1**	**3.859**	**0.2**
BWVENT	**9.574**	**1**	**9.6**	**3.859**	**36.9**
BWBAR	**10.3**	**1**	**10.3**	**3.859**	**39.7**
BARE	**11.43**	**1**	**11.4**	**1**	**11.4**

- The present work considers a forced draft cooling tower, and then considers the electric power of the fan (Gonzales and Nebra, 2005 did not consider this power consumption)

- On the other hand they calculated the exergy of the cooling flow using the Carnot factor while this work calculates the exergy of the chiller water flows (the limits of the control volume considered are different)

Moreover, in this work the temperature of the hot water is higher than the respective temperature adopted by Gonzales and Nebra (2005).

These differences make the system of Gonzales and Nebra (2005) more efficient than the system analyzed in this work, presenting lower exergetic costs.

The work of Accadia and Rossi (1998) also presented lower exergetic costs (k_{Qcold} = 3.826) than the present work. It is possibly due to better efficiency of the compression system and that those authors considered the unit exergetic cost of the electricity equal to unity.

Regarding the double-effect system, it can be observed that the unit exergetic cost of the negentropy flow produced in the cooling tower (S2931) in the direct-fired system is lower than the cost for the steam-driven system.

In the literature there are a few works about thermoeconomic analysis for double-effect LiBr/H_2O absorption refrigeration systems. One of these works was done by Misra et al. (2005). Their exergetic costs were lower (k_{Qcold} = 8.20 for the main product), However, they assumed the unit exergetic cost equal to 1 for both steam and electricity (fuels of the system).

7. Conclusions

In this work, new concepts are introduced to analyze an absorption system. To perform the division in control volumes, a thermal compressor was considered as a unique volume and a functional exergoeconomic analysis with negentropy was performed as well. These two new concept applications permit a simpler and complete analysis to provide easier comparisons with refrigeration systems based on mechanical compression.

The exergetic costs and the unit exergetic costs for the product of the system (single and double effect) is high due to the low exergetic efficiency of the system (Palacios-Bereche et al., 2007b).

The unit exergetic costs of the products is higher for the single-effect direct-fired system, according to the trends in the exergetic analysis, however the difference in costs between the direct-fired and the steam-driven systems (in double effect systems) is lower than the difference in costs between the direct-fired and hot-water driven systems (in single effect systems).

Similar to the case of the single-effect system, if the internal exergy flows of the double-effect system are observed, the analysis shows that the highest unit exergetic cost increases take place in the thermal compressor and in the evaporator + expansion valve control volume due to the high irreversibilities in these components.

Thus, from the standpoint of the exergetic analysis, single-effect absorption refrigeration systems are suitable to operate in cogeneration systems or using as fuel some waste heat at low temperature (higher than 80°C but lower than 120°C).

On the other hand, double effect systems have a better exergetic performance for either direct-fired or steam-driven, but they need higher temperatures to operate.

In the analysis of the direct-fired system (fueled by natural gas and electricity), unit exergetic costs equal to unity were adopted. The results would be different if the necessary exergy to produce the electricity consumed by the system was considered. With this last consideration the unit exergetic cost of the electricity would be higher than 1 and the unit exergetic cost of the product of the system (k_{2827}) would be higher too.

For the steam-driven system it was considered that the steam and electricity were produced by the cogeneration system. For this reason the unit exergetic costs of the external fuels were considered higher than 1. It contributes to an increase the exergetic cost of the main product.

Nomenclature

B	Exergy rate for the functional analysis (kW)
Ex	Exergy rate (kW)
E^*	Exergetic cost (kW)
h	Specific enthalpy (kJ/kg)
k	Unit exergetic cost
$\dot{m}$	Mass flow rate (kg/s)
P	Pressure (kPa)
s	Specific entropy (kJ/kg.K)
S	Negentropy flow (kW)
SI	Negentropy flow consumed (kW)
SP	Negentropy flow produced (kW)
T	Temperature (K)
TI	Exergy rate consumed
TP	Exergy rate produced
$\dot{W}$	Work flow (kW)
X	Mass fraction of lithium bromide (%)

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