THERMODYNAMIC SIMULATION OF A SOLAR ABSORPTION REFRIGERATION SYSTEM GENERATOR – HEAT EXCHANGER

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ABSTRACT

A thermodynamic simulation of a solar absorption refrigeration cycle has been carried out. The binary mixture considered in the present investigation was H_2O – NH_3 (Water – Ammonia). This simulation was performed in order to investigate the effect that the generator temperature and the heat exchanger efficiency have over the Coefficient of Performance (COP) and mass flux on a single absorption refrigeration system that uses solar energy as a primary source. It was found that the heat exchanger efficiency determines the maximum temperature that can be used at the generator in order to obtain the maximum COP out of the system. For a constant efficiency at the heat exchanger, there is an optimum temperature to be used at the generator, while a higher temperature will decrease the system COP.

KEYWORDS: Refrigeration, Absorption, Energy Systems, Thermodynamic Simulation.

SYMBOLS

COP = Coefficient of Performance, $\left[\dot{Q}_e \Big/ \left(\dot{Q}_g + \dot{W}_b\right)\right]$

$COP_{\max}$ = Maximum Coefficient of Performance

E = Efficiency
f = Mass flow ratio
h = Enthalpy (kJ/kg)
$\dot{m}$ = Mass flow (kg/s)
P = Pressure (kPa)
q = Heat per unit mass (kJ/kg)
Q = Heat (kJ)
$\dot{Q}$ = Heat rate (kW)
T = Temperature (^{0}C)
υ = Specific volume (m^3/kg)
$\dot{W}$ = Power (kW)
x = Concentration (kg/kg)

Subscripts

a = Absorber
b = Pump
c = Condenser
e = Evaporator
l = Liquid
g = Generator
ref = Refrigerant
sf = Strong solution
sd = Weak solution
v = Vapor

INTRODUCTION

The use of solar energy as an alternative source has been attracting a lot of interest in the last years due to the environmental considerations that apply all around the world. So many applications have been considered, including transportation and refrigeration systems, among others. Solar absorption refrigeration is one of the applications under review because of the advantages that present its use in sunny and warm regions, where solar power can be used as the main source for its operation. Unlike mechanical vapor compression refrigerators, these systems cause no ozone depletion and reduce demand on electricity supply. Besides, heat powered systems could be superior to electricity powered systems because of the use of inexpensive waste heat, solar, biomass or geothermal energy sources for which the supplying cost is negligible in many cases. Despite using an economic energy source, the system is characterized by its low COP, for that reason it is necessary to perform a study in order to find the most efficient operation range. One of the main factors that have helped to develop this kind of systems is the thermodynamic simulation that can be carried out in order to study the different variables affecting the performance of the equipment. Whitlow [1] gathered the memories of the Toronto 73rd ASHRAE Conference, where the absorption refrigeration cycle was studied from the thermodynamic point of view, the COP and $COP_{\max}$ were given as a function of the operation temperature at the generator, evaporator and condenser. The use of heat exchangers and some other binary mixtures were recommended. Van Passen [2] presented the work done by the International Health Organization in order to impulse a vaccination program to control child diseases through immunization where Delf University of Technology was involved in the thermodynamic simulation of a solar absorption refrigeration system. The simulated model was developed because of the excellent results obtained during the simulation process. In 1977 Shwarts and Shitzer [3] analyzed thermodynamically the possibility to operate the solar absorption refrigeration system for air conditioning. Their results showed that the system was suitable for domestic use. Sun [4] analyzed and performed an optimization of the water – ammonia cycle. As a result, he obtained a mathematical model that allowed the simulation of the process. Sun [5] presented a thermodynamic design and performed an optimization of the absorption refrigeration process in order to map the most common cycles for water – ammonia, and lithium bromide – water. The results can be used to select the operation conditions in order to obtain a maximum performance from the system. Sun [6] (1998) performed a thermodynamic analysis of different binary mixtures considered in the absorption refrigeration cycle.

The literature review on solar absorption refrigeration cycles shows that a thermodynamic simulation can be performed in order to study and analyze the system. A lot of work has been done in this area and the effect of the generator temperature has been considered extensively, but the effect of the heat exchange efficiency in a single absorption refrigeration system has not been considered.

MATHEMATICAL MODEL

Figure 1 illustrates the main components of the absorption refrigeration cycle. High-pressure liquid refrigerant (2) from the condenser passes into the evaporator (4) through an expansion valve (3) that reduces the pressure of the refrigerant to the low pressure existing in the evaporator. The liquid refrigerant (3) vaporizes in the evaporator by absorbing heat from the material being cooled and the resulting low-pressure vapor (4) passes to the absorber, where it is absorbed by the strong solution coming from the generator (8) through an expansion valve (10), and forms the weak solution (5).

The weak solution (5) is pumped to the generator pressure (7), and the refrigerant in it is boiled off in the generator. The remaining solution (8) flows back to the absorber and, thus, completes the cycle. By weak solution (strong solution) is meant that the ability of the solution to absorb the refrigerant vapor is weak (strong), according to the ASHRAE definition. In order to improve system performance, a solution heat exchanger is included in the cycle. An analyzer and a rectifier need to be added to remove water vapor from the refrigerant mixture leaving the generator before reaching the condenser. For the current study, it is assumed that the refrigerant vapor is 100% ammonia.

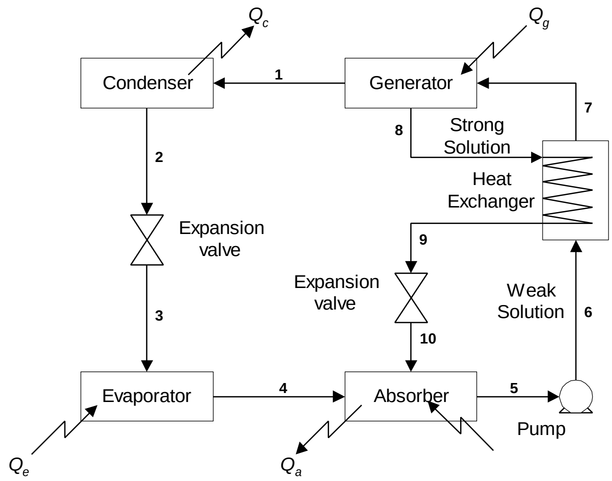

Figure 1. Schematic of an absorption refrigeration cycle.

In order to analyze the system, mass and energy balance must be performed at each component.

At the expansion valves,

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{ref} \qquad \text{(Total mass balance)} \tag{1}$$

$$\dot{m}_9 = \dot{m}_{10} \qquad \text{(Total mass balance)} \tag{2}$$

$$h_2 = h_3 \qquad \text{(Energy balance)} \tag{3}$$

$$h_9 = h_{10} \qquad \text{(Energy balance)} \tag{4}$$

At the evaporator,

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{ref} \qquad \text{(Total mass balance)} \tag{5}$$

$$\dot{Q}_e = \dot{m}_{ref}\,(h_4 - h_3) \qquad \text{(Energy balance)} \tag{6}$$

At the generator,

$$\dot{m}_7 = \dot{m}_1 + \dot{m}_8 \qquad \text{(Total mass balance)} \tag{7}$$

$$\dot{m}_7\, x_7 = \dot{m}_1\, x_1 + \dot{m}_8\, x_8 \quad \text{(NH}_3\text{ mass balance)} \tag{8}$$

$$\dot{Q}_g = \dot{m}_1\, h_1 + \dot{m}_8\, h_8 - \dot{m}_7\, h_7 \quad \text{(Energy balance)} \tag{9}$$

From equations (7) and (8), the strong and weak solution mass flow rate can be obtained

$$\dot{m}_8 = \frac{x_7 - x_1}{x_8 - x_7}\dot{m}_1 \tag{10}$$

$$\dot{m}_7 = \frac{x_8 - x_1}{x_8 - x_7}\dot{m}_1 \tag{11}$$

From equation (11), the circulation ratio can de derived:

$$f = \frac{\dot{m}_7}{\dot{m}_1} = \frac{x_8 - x_1}{x_8 - x_7} \tag{12}$$

At the absorber,

$$\dot{m}_4 + \dot{m}_{10} = \dot{m}_5 \quad \text{(Total mass balance)} \tag{13}$$

$$\dot{Q}_a = \dot{m}_4\, h_4 + \dot{m}_{10}\, h_{10} - \dot{m}_5\, h_5 \quad \text{(Energy balance)} \tag{14}$$

Dividing by $\dot{m}_4$

$$q_a = (h_4 - h_{10}) + f(h_{10} - h_5) \tag{15}$$

where $\dot{q}_a$ represents the heat dissipated per unit mass, and *f* the mass flow ratio. The first term of the right side represents the phase change, and the second the cooling of the mixture.

At the pump,

$$\dot{m}_5 = \dot{m}_6 \quad \text{(Total mass balance)} \tag{16}$$

$$\dot{W}_b = \dot{m}_5 (h_6 - h_5) \quad \text{(Energy balance)} \tag{17}$$

At the condenser,

$$\dot{m}_1 = \dot{m}_2 \quad \text{(Total mass balance)} \tag{18}$$

$$\dot{Q}_c = \dot{m}_{ref}\, (h_1 - h_2) \quad \text{(Energy balance)} \tag{19}$$

At the heat exchanger,

$$\dot{m}_8 + \dot{m}_6 = \dot{m}_7 + \dot{m}_9 \quad \text{(Total mass balance)} \tag{20}$$

$$h_7 = h_6 + \frac{\dot{m}_8}{\dot{m}_6}(h_8 - h_9) \quad \text{(Energy balance)} \tag{21}$$

COMPUTATIONAL MODEL

In order to analyze how the system reacts to different operating conditions, it is necessary to simulate the variables that affect its performance, with the intention of obtaining the maximum COP out of the system. The operating conditions choose were:

T_g = 70 – 90 ^{O}C

T_c = 30 ^{O}C

T_a = 25 ^{O}C

T_e = 5 ^{O}C

Refrigerant mass flow $\dot{m}_{ref}$ = 1.0 kg/s

Heat exchanger efficiency: 50 – 100%

High pressure: 1.16 MPa

Low pressure: 0.51 MPa.

From Sun [4], the pressure can be calculated according to equation (22). The liquid and gas enthalpies of the refrigerant (NH_3) can be calculated from equation (23) and (24), respectively.

$$P(T) = 10^3 \sum_{i=0}^{6} a_i (T - 27.15)^i \tag{22}$$

$$h_l(T) = \sum_{i=0}^{6} b_i (T - 27.15)^i \tag{23}$$

$$h_u(T) = \sum_{i=0}^{6} c_i (T - 27.15)^i \tag{24}$$

The coefficients for equations (22 – 24) are presented in Table 1.

Table 1. Coefficients for equations (22 – 24) (Da Wen Sun, 1997a)

i	a_i equation (22)	b_i equation (23)	c_i equation (24)
0	4.2871 x 10^{-1}	1.9879 x 10^{2}	1.4633 x 10^{3}
1	1.6001 x 10^{-2}	4.4644 x 10^{0}	1.2839 x 10^{0}
2	2.3652 x 10^{-4}	6.2790 x 10^{-3}	-1.1501 x 10^{-2}
3	1.6132 x 10^{-6}	1.4591 x 10^{-4}	-2.1523 x 10^{-4}
4	2.4303 x 10^{-9}	-1.5262 x 10^{-6}	1.9055 x 10^{-6}
5	-1.2494 x 10^{-11}	-1.8069 x 10^{-8}	2.5608 x 10^{-8}
6	1.2741 x 10^{-13}	1.9054 x 10^{-10}	-2.5964 x 10^{-10}
Standard error	1.6 x 10^{-1}	8.5626 x 10^{0}	1.059 x 10^{1}
Mean deviation	1.252 x 10^{-2}	5.566 x 10^{-3}	3.679 x 10^{-3}

For the mixture, the enthalpy was calculated according to equations (25 – 27).

$$h\left(60^{o}C, x\right) = 11101.5 - 148593x + 767227x^2 - 1911990x^3 + 2309150x^4 - 1084970x^5 \tag{25}$$

$$h\left(80^{o}C,x\right)=5708.1-76879.3x+413443x^{2}-1079610x^{3}+1367500x^{4}-672653x^{5} \quad (26)$$

$$h\left(100^{o}C,x\right)=33499.6-470584.x+2560630x^{2}-6740720x^{3}+8613990x^{4}-4282390x^{5} \quad (27)$$

Maximum standard error for equations (25 – 27) is 0.5%.

RESULTS

Table 2 presents a comparison between the results obtained in this simulation and Da Wen Sun (1997a) for COP and mass flow. It is noticed that the results in the actual investigation have a good agreement with those obtained by Da Wen Sun (1997a).

Table 2. COP and mass flow rate comparison for heat exchanger efficiency of 80 %

COP (Actual Investigation)	COP (Dae Wen Sun 1997a)	Error %	*f* (Actual Investigation)	*f* (Dae Wen Sun 1997a)	Error %
0,53	0,60	11,7	7,88	7,7	2,3
0,55	0,61	9,8	6,71	6,6	1,7
0,56	0,62	9,7	5,88	5,5	6,9
0,56	0,63	11,1	5,26	5	5,2
0,57	0,63	9,5	4,78	4,4	8,6
0,57	0,63	9,5	4,4	4,3	2,3
0,57	0,60	5,0	4,09	3,8	7,6
0,57	0,63	9,5	3,83	3,6	6,4
0,57	0,63	9,5	3,61	3,3	9,4
Media		9,13	Media		5,26
Deviation		1,69	Deviation		2,61

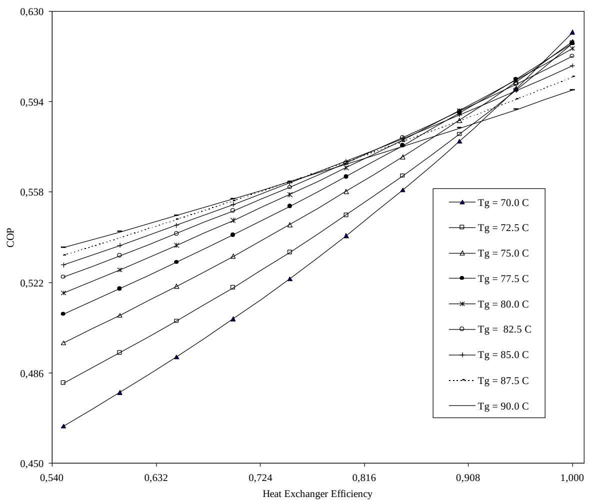

Figure 2. COP variation as a function of the heat exchanger efficiency for different values of temperature at the generator

Figure 2 presents the variation of the COP as a function of the heat exchanger efficiency for different generator temperatures. This figure can be described as follows:

- For a given generator temperature.
  - As the heat exchanger efficiency increases, the COP increases.
- For a given heat exchanger efficiency.
  - If E < 0.7, the system COP increases as the generator temperature rises.
  - If E > 0.7,
    - If T_g < 77.5 ^{O}C, the system COP increases as the efficiency augments.
    - If T_g > 77.5 ^{O}C, the system COP decreases as the efficiency increases.
- For a given system COP.
  - It can be attained with several combinations of generator temperature and heat exchanger efficiency.

The generator temperature effects over the system COP for different heat exchanger efficiencies can be observed in Figure 3. This figure can be described as follows:

- For a given generator temperature
  - The system COP increases as the efficiency augment.
- For a given heat exchanger efficiency
  - As the generator temperature is elevated, the COP of the system presents a maximum, after this point, as the temperature continues raising, the COP decreases.

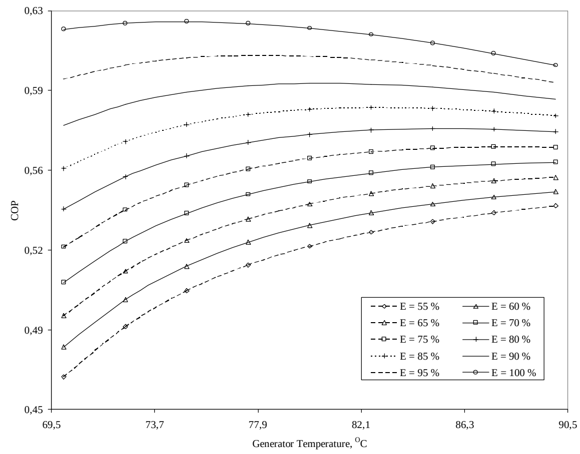

Figure 3. COP variation as a function of the generator temperature for different values of efficiency at the heat exchanger

Table 3. Heat, work and COP for different generator temperatures and heat exchanger efficiency of 50 %

Tg (°C)	Qc (KJ/s)	Qe (KJ/s)	Qa (KJ/s)	Qg (KJ/s)	Wb (KJ/s)	COP
70.0	1240.49	1115.84	2372.02	2488.67	8.0019	0.4469
72.5	1247.13	1115.84	2240.40	2364.87	6.8156	0.4705
75.0	1253.78	1115.84	2151.51	2283.47	5.9737	0.4874
77.5	1260.42	1115.84	2088.45	2227.68	5.3452	0.4997
80.0	1267.07	1115.84	2041.73	2188.09	4.8581	0.5088
82.5	1273.43	1115.84	2005.61	2158.73	4.4695	0.5158
85.0	1279.80	1115.84	1976.43	2136.23	4.1523	0.5213
87.5	1286.16	1115.84	1951.79	2118.22	3.8885	0.5258
90.0	1292.53	1115.84	1930.13	2103.15	3.6656	0.5296

Table 4. Heat, work and COP for different generator temperatures and heat exchanger efficiency of 75 %

Tg (°C)	Qc (KJ/s)	Qe (KJ/s)	Qa (KJ/s)	Qg (KJ/s)	Wb (KJ/s)	COP
70,0	1240,49	1128,63	2044,29	2149,73	6,4221	0,5234
72,5	1247,13	1128,63	1971,21	2084,24	5,4700	0,5401
75,0	1253,78	1128,63	1922,46	2042,81	4,7943	0,5512
77,5	1260,42	1128,63	1888,80	2016,30	4,2899	0,5586
80,0	1267,07	1128,63	1865,09	1999,62	3,8990	0,5633
82,5	1273,43	1128,63	1848,21	1989,42	3,5871	0,5663
85,0	1279,80	1128,63	1836,18	1984,02	3,3325	0,5679
87,5	1286,16	1128,63	1827,66	1982,07	3,1208	0,5685
90,0	1292,53	1128,63	1821,70	1982,66	2,9419	0,5684

Table 5. Heat, work and COP for different generator temperatures and heat exchanger efficiency of 100 %

Tg (°C)	Qc (KJ)	Qe (KJ)	Qa (KJ)	Qg (KJ)	Wb (KJ/s)	COP
70,0	1240,49	1128,63	1703,78	1809,21	6,4221	0,6216
72,5	1247,13	1128,63	1689,23	1802,26	5,4700	0,6243
75,0	1253,78	1128,63	1680,63	1800,98	4,7943	0,6250
77,5	1260,42	1128,63	1676,37	1803,87	4,2899	0,6242
80,0	1267,07	1128,63	1675,66	1810,19	3,8990	0,6221
82,5	1273,43	1128,63	1678,03	1819,24	3,5871	0,6192
85,0	1279,80	1128,63	1683,15	1830,98	3,3325	0,6153
87,5	1286,16	1128,63	1690,74	1845,15	3,1208	0,6106
90,0	1292,53	1128,63	1700,49	1861,45	2,9419	0,6054

Tables 3, 4 and 5 present the values of the heat at condenser, evaporator, absorber, and generator, as well as the work done by the pump. The COP of the system for these different conditions is presented too. The behavior presented in the previous plots can be confirmed here.

CONCLUSIONES

Ammonia water absorption refrigeration cycle was analyzed, with their thermodynamic properties expressed in polynomial equations [4]. The coefficient of performance (COP) of this cycle versus generator temperature and heat exchanger efficiency was analyzed and it was noticed that the heat exchanger efficiency is an important factor at the moment to consider the optimum temperature at which a solar absorption refrigeration cycle operates. The heat exchanger efficiency determines the maximum temperature that should be used at the generator in order to achieve the maximum COP out of the system. The simulation was carried out for specific temperatures and pressures at the evaporator and condenser and the study must continue in order to obtain operational maps that include the heart exchanger efficiency as a variable.

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