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ABSTRACT

A thermodynamic simulation of a charcoal heat driven absorption refrigeration cycle was carried out. The simulation was performed in order to investigate the performance of the cooler and the effect the generator temperature has over the coefficient of performance (COP) on a single absorption refrigeration system that uses charcoal heat energy as a primary source. A 10 kg of water was used as design load to cool from ambient temperature of 32°C to a desirable temperature of 10°C. The temperature of the evaporator is 10°C while the condenser temperature is 40°C. The high and low system pressures are 16 atm and 4 atm respectively. The refrigerant-absorbent mixture is Ammonia-water. The system requires two effective heating hours to generate the refrigeration needed by the cooler. It was found that as generator temperature increases, the performance decreases. The paper provides theoretical basis for designing a cooler with heat provided by burning charcoal in a stove.

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

NOMENCLATURE

COP = Coefficient of performance, $\left(\frac{Q_e}{Q_g} \right)$

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 (kW)

T = Temperature (°C)

x = Concentration (kg/kg)

Subscripts

a = Absorber

c = Condenser

e = Evaporator

g = Generator

v = Vapour

l = Liquid

ref = Refrigerant

1.0 INTRODUCTION

The use of charcoal as a source of energy and as an alternative source for cooking has been attracting a lot of interest in recent years due to the non availability of conventional energy, electricity and fossils fuel such as kerosene and cooking gas that apply all around Nigeria. Heat driven systems either waste heat and solar absorption refrigeration are among the applications under review because of their advantages that present its use in forest and

tropical areas, where charcoal heat can be used as the main source for its operation.

Unlike mechanical vapour compression refrigerators, these systems reduce demand on electricity supply. Besides, heat powered systems could be preferable to electricity powered systems because of the use of inexpensive waste heat, biomass or geothermal energy sources for which the supplying cost is negligible in many cases.

Despite the system is characterized by its low COP, using an economic energy source, for that reason it is necessary to perform a study in order to find the possibility of developing the system and 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. Antonio (1997) reported that ‘Whitlow’ 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 (1987) 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 Shirtzer analyzed, thermodynamically, the possibility for applying the solar absorption refrigeration system

for air conditioning. Their results showed that the system was suitable for domestic use. Sun (1997) 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 (1997) 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 (1997) performed a thermodynamic analysis of different binary mixtures considered in the absorption refrigeration cycle.

The literature review on absorption refrigeration cycles shows that a thermodynamic simulation can be performed in order to study and analyze the system. Several studies in this area have been carried out and the effect of the generator temperature has been considered extensively, but the use of charcoal as a source of energy has not been considered.

The cooler is designed to cool water from the ambient temperature of 32°C to 10°C using absorption refrigeration techniques, with charcoal as a source of heat to drive the system.

Mild steel, aluminum and pipes were used for the cooler materials, and they are sourced locally.

2.0 MATERIALS AND METHODS

2.1 Mathematical model

For the current study, it is assumed that the refrigerant vapour is 100% ammonia. To analyze the system mass and energy balance must be performed at each component. At the evaporator the total mass and energy balance may, respectively, be expressed as:

$$\dot{m}_3 = \dot{m}_4 = \dot{m}_{ref} \quad (1)$$

$$\dot{Q}_e = \dot{m}_{ref}(h_4 - h_3) \quad (2)$$

Where \dot{Q}_e is the total cooling load for the cooler.

At the expansion valve the total mass balance may be expressed using equations 3 and 4, while the energy balance is expressed using equations 5 and 6:

$$\dot{m}_2 = \dot{m}_3 = \dot{m}_{ref} \quad (3)$$

$$\dot{m}_6 = \dot{m}_7 \quad (4)$$

$$h_2 = h_3 \quad (5)$$

$$h_6 = h_7 \quad (6)$$

The mass flow rate at state 1 is given by

$$\dot{m}_{ref} = \dot{m}_2 = \dot{m}_3 = \dot{m}_4$$

At the generator the total mass balance, NH_3 and energy balance are respectively given by:

$$\dot{m}_5 = \dot{m}_1 + \dot{m}_6 \quad (7)$$

$$\dot{m}_5 x_5 = \dot{m}_1 x_1 + \dot{m}_6 x_6 \quad (8)$$

$$\dot{Q}_g = \dot{m}_1 h_1 + \dot{m}_6 h_6 - \dot{m}_5 h_5 \quad (9)$$

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

$$\dot{m}_6 = \frac{x_5 - x_1}{x_6 - x_5} \dot{m}_1 \quad (10)$$

$$\dot{m}_5 = \frac{x_6 - x_1}{x_6 - x_5} \dot{m}_1 \quad (11)$$

From equation (11), circulation ratio can be derived

$$f = \frac{\dot{m}_5}{\dot{m}_1} = \frac{x_6 - x_1}{x_6 - x_5} \quad (12)$$

At the absorber the total mass and energy balance are expressed as

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_7 \quad (13)$$

$$\dot{Q}_a = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_5 h_5 \quad (14)$$

Dividing by \dot{m}_4 , we get

$$q_a = (h_4 - h_7) + f(h_7 - h_5) \quad (15)$$

Where: q_a - represents the heat dissipated per unit mass; f - the mass flow ratio.

The first term of the right side represents the phase change and the second the cooling of the mixture. Thus, all the flow rates of the system are known. The position of the absorber is situated so that the flow to the generator is by gravity. At the condenser, the total mass and energy balance may, respectively, be expressed as:

$$\dot{m}_1 = \dot{m}_2 = \dot{m}_{ref} \quad (16)$$

$$\dot{Q}_c = \dot{m}_{ref}(h_1 - h_2) \quad (17)$$

The energy balance of the system can be determined as summarized in Table 1. The dimensions of the cooler are selected in accordance with Table 2. The selection is based on the standard sizes of the conventional coolers which can be adequate for a family size of ten persons. The configurations of a typical wall are shown in Figure 1 below. The cooler door is made of aluminum mild steel sheet and polystyrene insulation.

Table 1: Energy Balance of the system

Component	Calculations
Heat gain by the evaporator \dot{Q}_e	Total cooling load as calculated
Heat loss by the condenser \dot{Q}_c	$\dot{m}_{ref}(h_1 - h_2)$
Heat loss by the absorber \dot{Q}_a	$\dot{m}_4 h_4 + \dot{m}_6 h_6 - \dot{m}_5 h_5$
Heat gain by the generator \dot{Q}_g	$\dot{Q}_g = \dot{m}_1 h_1 + \dot{m}_6 h_6 - \dot{m}_5 h_5$

Table 2: Dimension of the cooler

Ref	Outside dimensions (m)	Internal dimensions (m)
Length	0.977	0.85
Width	0.627	0.50
Height	0.7135	0.65

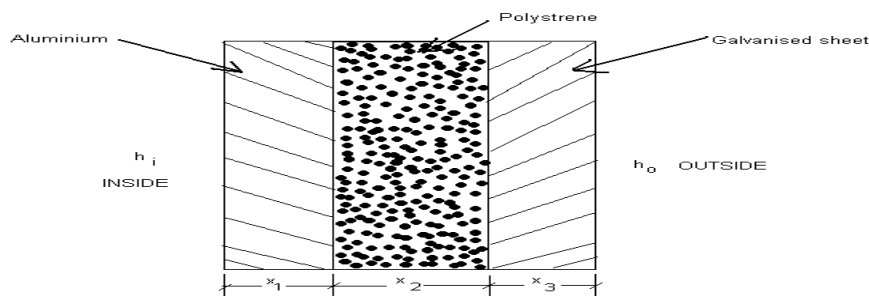


Fig 4.1 Configuration of the cooler wall

Figure 1: Configuration of cooler wall

2.2 Component design

Based on the initial design data, design calculations were carried out where many parameters were obtained such as total cooling load as 171.60 W, the length of the evaporator pipe 13.9 m. Heat released by the condenser as 176.30 W and its length as 3.2 m with fins that were used to increase the surface area of heat transfer. The heat released by the absorber was found to be 624.09 W and its length as 12.1 m.

For generator, the thermal design was carried out the mass of charcoal required to run the system for two hours was determined as 0.826 kg using power factor ϕ of 19.77% (Sazawa, 1997) was adapted for the design.

$$M_{\text{Charcoal}} = \frac{Q_g}{C_v} t\phi \quad (18)$$

Where: C_v - is the calorific value of the fuel (Charcoal = 29MJ/kg) (Dalhatu, 2004); t - is the time taken to burn that particular mass of fuel

2.3 Computational model

To analyze how the system reacts to the operating conditions, it is necessary to simulate the variables that affect its performance, with the intention of obtaining the coefficient of performance (COP) of the system.

The operating conditions from the design data are: $T_g = 70 - 120^\circ\text{C}$; $T_c = 40^\circ\text{C}$; $T_a = 40^\circ\text{C}$; $T_e = 10^\circ\text{C}$; Refrigerant mass flow = 1.61×10^{-4} kg/s; High pressure = 16×10^5 Pa; and Low pressure = 6×10^5 Pa. The liquid and gas enthalpies of the refrigerant (NH_3) can be calculated using the equations $h_l(T) = \sum_{i=0}^6 b_i (T - 27.15)$ and $h_g(T) = \sum_{i=0}^6 c_i (T - 27.15)$ as suggested by Sun (1997). The coefficients for these equations are presented in Table 3.

Table 3: Coefficients b_i and c_i (Sun, 1997)

i	b_i	c_i
0	198.79	1463.3
1	4.4644	1.2839
2	6.2790×10^{-3}	-1.1501×10^{-2}
3	1.4591×10^{-4}	-2.1523×10^{-4}
4	-1.5262×10^{-6}	1.9055×10^{-6}
5	-1.8069×10^{-8}	2.5608×10^{-8}
6	1.9054×10^{-10}	-2.5964×10^{-10}

For the mixture, the enthalpy was calculated using these equations (Sun, 1997):

$$h(60^\circ\text{C}, x) = 11101.5 - 148593x + 767227x^2 - 1911990x^3 + 2309150x^4 - 1084970x^5$$

$$h(80^\circ\text{C}, x) = 5708.1 - 76879.3x + 413443x^2 - 1079610x^3 + 1367500x^4 - 672653x^5$$

$$h(100^\circ\text{C}, x) = 33499.6 - 470584x + 2560630x^2 - 6740720x^3 + 8613990x^4 - 4282390x^5$$

Maximum standard error for equations is 0.5%

3.0 RESULTS AND DISCUSSION

3.1 Results

The results obtained from the simulation are presented in Tables 4 - 9.

Table 4: COP of the system for 40% aqua-ammonia and 60°C

(T _g)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	736.73	198.24	728.73	0.2582
72.5	190.25	755.51	198.20	747.56	0.2518
75	190.25	774.33	198.14	766.43	0.2457
77.5	190.25	793.15	198.06	785.34	0.2399
80	190.25	811.96	197.96	804.25	0.2343
82.5	190.25	830.76	197.85	823.15	0.2290
85	190.25	849.51	197.73	842.03	0.2240
87.5	190.25	868.23	197.61	860.87	0.2191
90	190.25	886.89	197.47	879.66	0.2145
92.5	190.25	905.49	197.32	898.41	0.2101
95	190.25	924.04	197.17	917.11	0.2059
97.5	190.25	942.54	197.01	935.78	0.2018
100	190.25	961.01	196.83	954.43	0.1980
102.5	190.25	979.47	196.63	973.08	0.1942
105	190.25	997.94	196.41	991.77	0.1906
107.5	190.25	1016.46	196.17	1010.54	0.1872
110	190.25	1035.09	195.88	1029.46	0.1838
112.5	190.25	1053.89	195.54	1048.59	0.1805
115	190.25	1072.91	195.15	1068.01	0.1773
117.5	190.25	1092.25	194.67	1087.83	0.1742
120	190.25	1112.02	194.10	1108.16	0.1711

Table 5: COP of the system for 40% aqua-ammonia and 80°C

(T _g)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	616.22	198.24	608.23	0.3087
72.5	190.25	635.01	198.20	627.06	0.2996
75	190.25	653.82	198.14	645.93	0.2910
77.5	190.25	672.65	198.06	664.84	0.2828
80	190.25	691.46	197.96	683.75	0.2751
82.5	190.25	710.25	197.85	702.65	0.2679
85	190.25	729.01	197.73	721.52	0.2610
87.5	190.25	747.72	197.61	740.36	0.2544
90	190.25	766.38	197.47	759.16	0.2482
92.5	190.25	784.99	197.32	777.91	0.2424
95	190.25	803.54	197.17	796.61	0.2368
97.5	190.25	822.04	197.01	815.28	0.2314
100	190.25	840.51	196.83	833.93	0.2264
102.5	190.25	858.96	196.63	852.58	0.2215
105	190.25	877.44	196.41	871.27	0.2168
107.5	190.25	895.96	196.17	890.04	0.2123
110	190.25	914.59	195.88	908.96	0.2080
112.5	190.25	933.38	195.54	928.08	0.2038
115	190.25	952.41	195.15	947.51	0.1998
117.5	190.25	971.75	194.67	967.33	0.1958
120	190.25	991.52	194.10	987.66	0.1919

Table 6: COP of the system for 40% aqua-ammonia and 100°C

(T _e)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	414.65	198.24	406.65	0.4588
72.5	190.25	433.43	198.20	425.48	0.4389
75	190.25	452.25	198.14	444.35	0.4207
77.5	190.25	471.07	198.06	463.26	0.4039
80	190.25	489.88	197.96	482.17	0.3884
82.5	190.25	508.68	197.85	501.07	0.3740
85	190.25	527.43	197.73	519.95	0.3607
87.5	190.25	546.15	197.61	538.79	0.3483
90	190.25	564.81	197.47	557.58	0.3368
92.5	190.25	583.41	197.32	576.33	0.3261
95	190.25	601.96	197.17	595.03	0.3161
97.5	190.25	620.46	197.01	613.7	0.3066
100	190.25	638.93	196.83	632.35	0.2978
102.5	190.25	657.39	196.63	651	0.2894
105	190.25	675.86	196.41	669.69	0.2815
107.5	190.25	694.38	196.17	688.46	0.2740
110	190.25	713.01	195.88	707.38	0.2668
112.5	190.25	731.81	195.54	726.51	0.2600
115	190.25	750.83	195.15	745.93	0.2534
117.5	190.25	770.17	194.67	765.75	0.2470
120	190.25	789.94	194.10	786.08	0.2408

Table 7: COP of the system for 46% aqua-ammonia and 60°C

(T _e)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	775.53	198.24	767.53	0.2453
72.5	190.25	794.32	198.20	786.36	0.2395
75	190.25	813.13	198.14	805.24	0.234
77.5	190.25	831.95	198.06	824.14	0.2287
80	190.25	850.77	197.96	843.05	0.2236
82.5	190.25	869.56	197.85	861.95	0.2188
85	190.25	888.32	197.73	880.83	0.2142
87.5	190.25	907.03	197.61	899.67	0.2097
90	190.25	925.69	197.47	918.47	0.2055
92.5	190.25	944.29	197.32	937.22	0.2015
95	190.25	962.84	197.17	955.92	0.1976
97.5	190.25	981.35	197.01	974.59	0.1939
100	190.25	999.81	196.83	993.23	0.1903
102.5	190.25	1018.27	196.63	1011.88	0.1868
105	190.25	1036.74	196.41	1030.57	0.1835
107.5	190.25	1055.27	196.17	1049.35	0.1803
110	190.25	1073.90	195.88	1068.26	0.1772
112.5	190.25	1092.69	195.54	1087.39	0.1741
115	190.25	1111.72	195.15	1106.81	0.1711
117.5	190.25	1131.06	194.67	1126.63	0.1682
120	190.25	1150.82	194.10	1146.97	0.1653

Table 8: COP of the system for 46% aqua-ammonia and 80°C

(T _e)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	638.70	198.24	422.18	0.2979
72.5	190.25	657.49	198.20	441.01	0.2894
75	190.25	676.30	198.14	459.89	0.2813
77.5	190.25	695.13	198.06	478.79	0.2737
80	190.25	713.94	197.96	497.70	0.2665
82.5	190.25	732.73	197.85	516.60	0.2596
85	190.25	751.49	197.73	535.48	0.2532
87.5	190.25	770.20	197.61	554.32	0.2470
90	190.25	788.86	197.47	573.11	0.2412
92.5	190.25	807.47	197.32	591.86	0.2356
95	190.25	826.01	197.17	610.56	0.2303
97.5	190.25	844.52	197.01	629.23	0.2253
100	190.25	862.99	196.83	647.88	0.2205
102.5	190.25	881.44	196.63	666.53	0.2158
105	190.25	899.91	196.41	685.22	0.2114
107.5	190.25	918.44	196.17	703.99	0.2071
110	190.25	937.07	195.88	722.91	0.2030
112.5	190.25	955.86	195.54	742.04	0.1990
115	190.25	974.89	195.15	761.46	0.1952
117.5	190.25	994.23	194.67	781.28	0.1914
120	190.25	1013.99	194.10	801.61	0.1876

Table 9: COP of the system for 46% aqua-ammonia and 100°C

(T _e)	\dot{Q}_e (J/s)	\dot{Q}_g (J/s)	\dot{Q}_c (J/s)	\dot{Q}_a (J/s)	COP
70	190.25	247.65	198.24	455.38	0.7682
72.5	190.25	266.48	198.20	474.21	0.7139
75	190.25	285.35	198.14	493.08	0.6667
77.5	190.25	304.24	198.06	511.99	0.6253
80	190.25	323.14	197.96	530.90	0.5888
82.5	190.25	342.03	197.85	549.80	0.5562
85	190.25	360.90	197.73	568.67	0.5272
87.5	190.25	379.72	197.61	587.51	0.5010
90	190.25	398.51	197.47	606.31	0.4774
92.5	190.25	417.24	197.32	625.06	0.4560
95	190.25	435.93	197.17	643.76	0.4364
97.5	190.25	454.58	197.01	662.43	0.4185
100	190.25	473.21	196.83	681.08	0.4020
102.5	190.25	491.84	196.63	699.73	0.3868
105	190.25	510.51	196.41	718.42	0.3727
107.5	190.25	529.26	196.17	737.19	0.3595
110	190.25	548.15	195.88	756.11	0.3471
112.5	190.25	567.24	195.54	775.24	0.3354
115	190.25	586.62	195.15	794.66	0.3243
117.5	190.25	606.39	194.67	814.48	0.3137
120	190.25	626.67	194.10	834.81	0.3036

Tables 4 - 9 present the values of the heat and COP at evaporator, generator, condenser and absorber.

The behavior of the plots is presented in Figures 2 and 3.

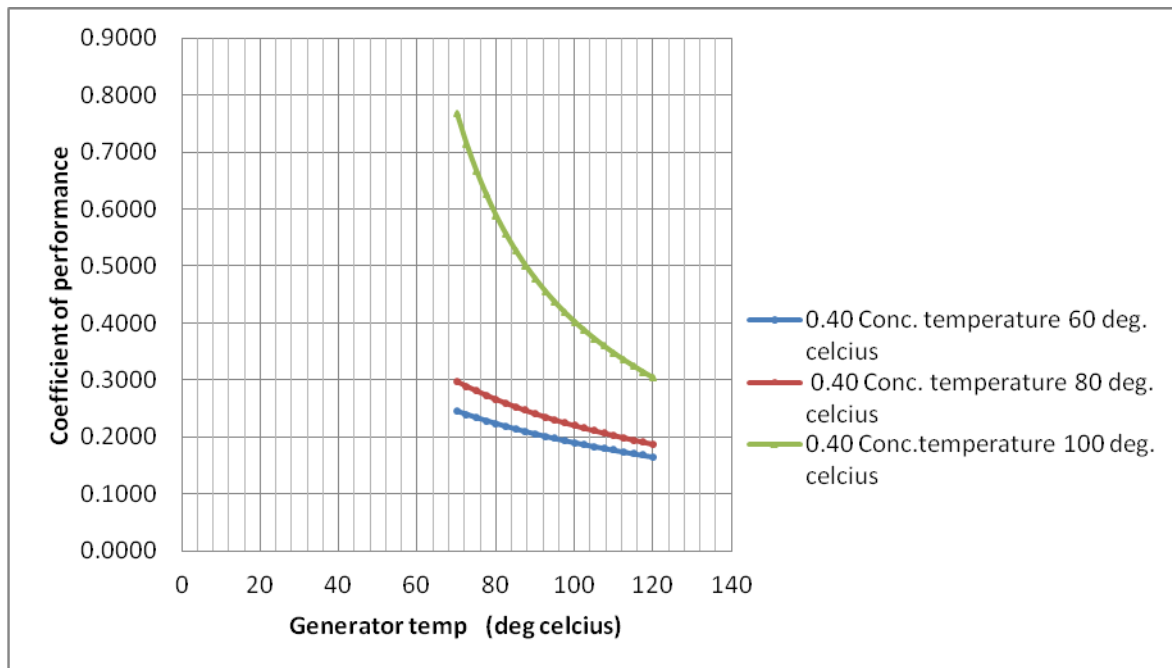


Figure 2: COP versus generator temperature at 0.40 aqua-ammonia concentration

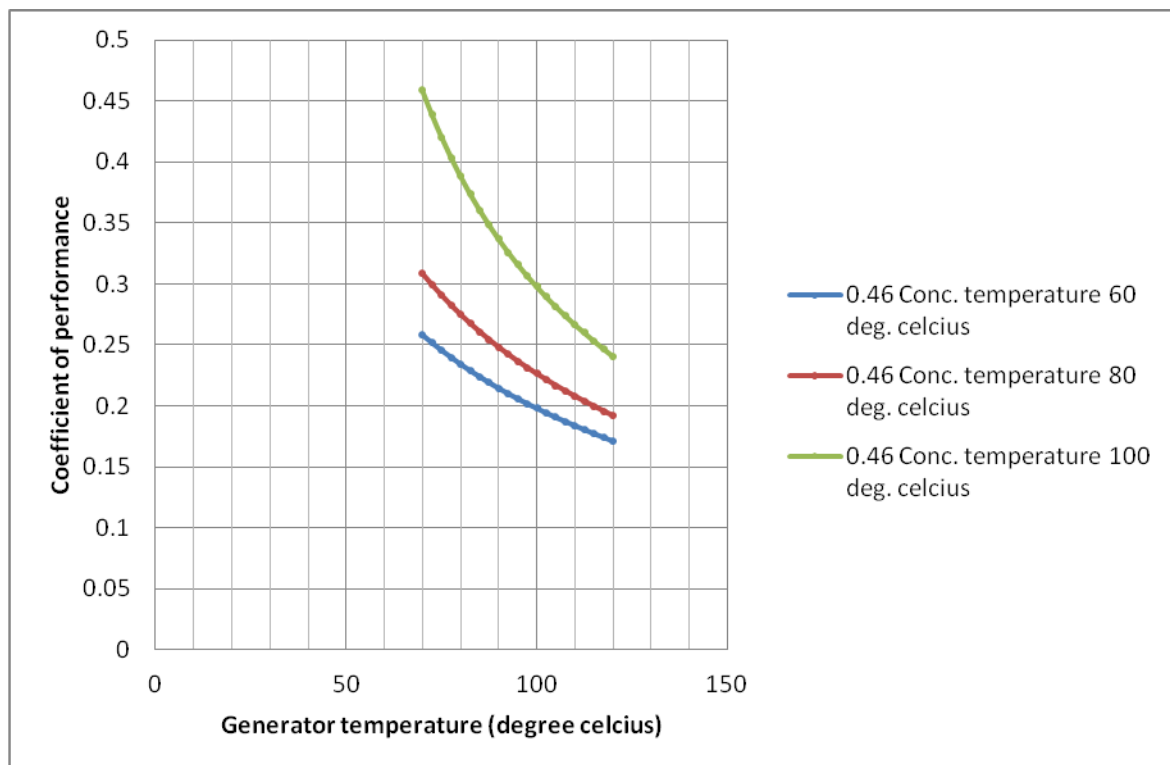


Figure 3: COP versus generator temperature at 0.46 aqua-ammonia concentration

3.2 Discussion of results

The results obtained from simulation for the absorption refrigeration cycle shown in Figures 2 and 3, were based on the thermodynamics model having as initial parameters those indicated earlier. In Figures 2 and 3 the COP variation was presented versus generator temperature from 70°C to 120°C, for different values of temperatures 60°C, 80°C and 100°C of aqua-ammonia mixture with concentration 0.40 and 0.46 respectively. The COP of the system developed in the current study (0.165-0.768) is in agreement with that reported by Sun (1997).

Analyzing the figures it can be noticed that the COP decreases with the increase in the generator temperature. In other words, the dependence of COP on the generator input at fixed concentration, evaporator temperature and pressure of the system shows that the COP decreases when the generator heat input increases. In figure 2, it was observed that while the strong solution temperature increases the coefficient of performance increases and highest COP for the strong solution concentration of 0.40 was 0.459. A similar behavior was observed in figure 3 with highest COP of 0.768 for the strong solution concentration of 0.40.

4.0 CONCLUSION

A design and simulation of charcoal fired heat driven cooler, for domestic use was done and the system performance was analyzed parametrically by computer simulation. A simulation of the designed parameters of temperatures and pressures at the evaporator and condenser was also carried out. The design was analyzed with thermodynamic properties of ammonia – water expressed in polynomial equation. The coefficient of performance (COP) of this system versus generator temperature with different concentrations of aqua-ammonia mixture was analyzed which shows that

that the COP of the system is higher when the generator temperature and concentration of stronger solution are high.

The figures plotted presents the various performances obtained in this simulation of the system which decreases as the temperature of the generator increases. The results obtained have a good agreement with those obtained by Sun (1997). The comparison of the COP, for validation of the simulation under particular operating conditions suggests that the predictions agree with the results obtained by other researchers.

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