

Development of a Model for Design of Double - Effect Vapor Absorption Refrigeration Systems

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Abstract- Increasing cost of electricity and environmental degradation, emanates from the usage CFC refrigerants have made the heat from renewable energy sources for absorption refrigeration systems attractive for application in residential, commercial and industrial sectors. In this research work, mathematical model for a double effect reverse – parallel vapour absorption refrigeration system was developed. Software was developed for the system using MATLAB Programming Language. The software was simulated and design parameters needed for the analysis of the system were generated. The model was validated by comparing its result with those from literature and showed good agreement with an average deviation of 20%. The model was used to estimate both refrigeration capacity and the coefficient of performance (COP). Highest COP was obtained when the concentration of the diluted solution in the evaporator was 62.2%. Generally the COP increases with increasing concentration of the solution in the absorber. The software developed can be used to predict design parameters for double-effect reverse-parallel VARS over a wide range of heat input.

Indexed Terms- design, double-effect, reverse-parallel, software, VARS.

I. INTRODUCTION

Absorption refrigeration systems used various combinations of two chemical compounds, known as refrigerant and absorbent for operations. The absorbent absorbs the refrigerant vapour, without any chemical reaction between them at all working conditions; and releases the refrigerant when heated. The combination of refrigerant and absorbent is call fluid pair (Renjith and Joshy, 1999). Absorption systems use heat energy to produce refrigeration as

well as heating if required. In LiBr/H₂O system, water is the refrigerant and aqueous Lithium Bromide (LiBr) is widely used as the carrier to absorb the refrigerant and provide a higher coefficient of performance (Wang and Lavan, 1999).

Vapour absorption refrigeration system (VARS) belong to the class of vapour cycles similar to vapour compression systems. However, unlike vapour compression refrigeration system, the required input to absorption system is in the form of heat. Similar to vapour compression refrigeration systems, VARS have in various refrigeration and air conditioning applications. Since these systems run low-grade thermal energy, they are preferred when low-grade energy, such as waste heat or solar energy is available. Conventional absorption systems use natural refrigerants such as water or ammonia that are environment friendly. Absorption refrigerator typically operates with relatively low quality heat energy. Therefore they can be powered using heat rejected or wasted by many industrial processes. Thus the waste heat energy can be converted to useful energy to power refrigerating systems. The use of waste heat energy can therefore cut global CO₂ emissions and reduce the global warming problem. According to Vliet et al.; (1982), the performance of single-effect absorption cycle can be further improved by accommodating a second generator, as in the case of double-effect. A VARS having two generators is known as double-effect absorption system. Double-effect LiBr – H₂O systems have proved to have significant higher COP than single effect but they need high driving temperature (Grossman, 2002). Many different thermodynamic cycles have been proposed, but the LiBr – H₂O pair is one of the most interesting. The refrigerant absorbent pair H₂O – LiBr, according to Asdrubali and Grignaffini, (2005) has numerous advantages such as high enthalpy of vaporization, no

need for rectification; it is neither toxic nor dangerous to health.

Manohar et al; (2006) developed a steady state model of a double-effect absorption chiller using steam as heat input. The model was based on the Artificial Neural Network (ANN) and predicts the chiller performance based on chilled water inlet and outlet temperatures and steam pressure. The network was trained with one year of experimental data and predicts the performance within $\pm 1.2\%$ of the actual values. Bereche et al; (2009) carried out a thermo-economic analysis of single and double effect LiBr-H₂O absorption refrigeration system. The methodology of functional analysis with negentropy was used. Negentropy is a concept related to thermal system for analyzing a power generation system with the aim of taking into account the useful function of dissipative equipment. The exergetic cost of the main product and the cooling cost, were calculated as a function of the energy of the heat source. Two cases were considered for the analysis of the single-effect system. The first case considered a direct-fired system whilst the second was 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 considered a direct-fired system, while the second was a steam-driven system. The steam-driven system considered that the absorption refrigeration system uses steam supplied from a cogeneration system. The data used for these cases and the properties of the LiBr-H₂O solutions were taken from previous works.

Oberweis and Al-shemmeri, (2012), developed a computer model that evaluates the performance of a Thermo-Chemical Accumulator (TCA). The model was based on operational data such as temperature and flow rate. Their ultimate goal was for the model to estimate the coefficient of performance (COP) of the unit when run on hot water from biomass combustion as the heat source. The input data consisted of temperatures for the heating load, heat sink and cooling circuit as well as respective flow rates. In a second instance the model was used to estimate the COP of a TCA unit when run on hot water from biomass combustion as the heat source. Although the system was developed to run on solar heating compared to the lower heat source temperature

available from domestic hot water boilers run on biomass. Thermodynamic analysis of LiCl absorption cooling was carried out and a computer model was developed. The model used a fixed number of internal time steps for every output time step in order to iterate to a reasonable solution or result. The simulations were all ran for heat load temperature varying between 80 °C and 100 °C. The outputs of the model were verified by comparing the simulation of the actual machine with the published experimental data. The computed results for cooling COP were within 10% of the measured data. Also, the results showed an increase in COP for increase heat source temperature as well as for decreased heat sink temperature while keeping all other variables constant.

Mahendru et al; (2012) analyzed steady state vapour absorption refrigeration systems; adopting solar energy chiller application. Mathematical models of various component of the system were developed, using visual basic programming and its performance was evaluated. The simulation provided good tools for identifying the parameter or components of H₂O – LiBr solution. The maximum COP and generator temperature were evaluated. It was observed that COP increased with generator temperature, though the highest temperature that could be reached at the generator was limited because solar energy was used as the heat generator. They concluded that, the main constraint of the system is the economic factor since the conventional energy is by far cheaper than renewable energy.

Micallef and Micallef (2012) developed a mathematical model for a vapour absorption refrigeration system. The system responses to varying absorber, generator and condenser temperature. A simulation of single-effect absorption, which can be modeled for LiBr-H₂O and H₂O – NH₃ pairs, was performed. The model was used to compute output parameters that are of fundamental importance for the analysis of the system. The model was implemented using Microsoft Visual Basic NET. The software, SOLSBSORB, gives useful numerical data for a single effect absorption refrigeration system, which uses LiBr-H₂O or H₂O-NH₃ absorbent-refrigerant pairs. The software enables system designers to input certain design conditions from which information can be

obtained regarding the operating conditions of the system including COP.

Further work can be seen in the works of Viswanathan et al., (2012); Kohlenbach and Ziegler (2008); Wang and GAO (2013); Azhar and Siddiqui (2013); Akintunde (2004) Akintunde et al., (2006); Azhar and Siddiqui (2013); Wang and Lavan, (1999).

The objectives of this work are to develop mathematical models for a double-effect reverse-parallel vapour absorption system; and validate the computer-aided application developed.

II. MATERIALS AND METHOD

In this work a double-effect direct fired LiBr-H₂O VARS with reverse parallel flow was modeled. In order to achieve the stated objectives, mass and energy balance analysis of the components were carried out.

Fig. 1 shows the schematic diagram of the double-effect direct fired LiBr-H₂O absorption chiller with reverse parallel flow modeled in this work. In the High-Temperature Generator (TG), diluted LiBr solution was heated and water vaporized from the solution. As water vapour boiled off, LiBr solution becomes concentrated in the HTG. The concentrated solution leaves HTG and enters the High-Temperature Heat Exchanger (HTG_x) at temperature t_7 with concentration x_7 (state point 7). The concentrated solution cools to temperature t_8 as it leaves the HTG_x. The water vapour that boils off from the LiBr solution leaves the HTG at temperature t_9 and flows through the tubes submerged in the diluted solution in the Low-Temperature Generator (LTE), the water vapour condenses to liquid. Latent heat of condensation released from the liquid refrigerant heats the diluted solution in LTG from temperature t_2 to temperature t_3 and boils off the water vapour from the diluted solution. In the LTG, the diluted solution becomes partly concentrated and splits into two streams with same concentrations x_3 and x_{14} when it leaves at the outlet. Stream of partly concentrated LiBr solution from LTG with concentration x_{14} combines with the concentrated solution from the HTG_x and the mixture enters the LTG_x at state point 4 with temperature t_4

and concentration x_4 . It then flows through LTG_x, where the solution temperature drops to t_5 .

The solution then enters the Absorber, sprays over the cooling water tube bundle and absorbs water vapour from the evaporator. The Absorber by absorbing the refrigerant vapour (in this case water vapour) produces low pressure that required for the operation of the evaporator. The condenser liquid refrigerant flow inside the tubes in the LTG enters the trough of the condenser at temperature t_{15} .

Water vapour boiled off in the LTG at temperature t_{10} is extracted by the condenser's low saturated pressure through the top passage and condenses to liquid as the cooling water removes the latent heat of condensation and drops the temperature to t_{11} . In the condenser, cooling water enters at low temperature and leaves at slightly higher temperature. The condensed liquid refrigerant from the tubes is cooled to a temperature t_{11} and combines with the liquid refrigerant from LTG condensed in the condenser. Liquid refrigerant at temperature t_{11} passes through an expansion valve where it expands to very low pressure and temperature t_{12} .

A vacuum of 6 to 8 mmHG abs was maintained inside the evaporator so as to attain low pressure and temperature range between 5 °C and 15 °C. Chilled water enters the evaporator inside the tube bundle leaves at a lower temperature. Heat was transferred from the chilled water to the refrigerant at an amount equal to the latent heat of vaporization and refrigerant becomes saturated vapour at temperature t_{13} .

Vaporized water in the evaporator was then extracted to the absorber because of its lower vapour pressure and is absorbed by the concentrated LiBr solution. In the Absorber, concentrated solution was supplied and sprayed at temperature t_5 with a concentration x_5 . As the water vapour from the evaporator was absorbed, the solution becomes diluted to a concentration of x_1 and its temperature drops to t_1 . In the LTG_x, diluted solution enters at temperature t_1 and leaves at t_2 complete description of the system can be found in the work of Adedoyin (2016).

III. ASSUMPTIONS FOR THE MODEL

In order to simplify the calculations and equations guiding the model, the following assumptions were made:

Steady state conditions were assumed, there is no loss of heat during each thermodynamic process; the refrigerant is pure water; pressure loss in the lines is negligible; the states at points 1, 4, 11, 3, 7 and 15 are saturated liquid; the state at point 13 is saturated vapour; LiBr solution is assumed to be in equilibrium with the reference pressure and temperature; mass flow rates of the refrigerant vapour at HTG and LTG are equal; mass flow rates of partially concentrated solution at state points 3 and 4 leaving the LTG are equal; the works input required for the refrigerant pump (Ep) and solution pumps (Ap, LTGp, HTGp) are negligible compared to heat input in HTG; heat loss across all the pumps is negligible; the generator temperature depends on the temperature range of the heat source (solar, biogas, fossil fuel, waste heat); this temperature should be above the minimum generator temperature; the temperature difference between high temperature condenser and LTG is 5 K; the generator and condenser operate at a high pressure level which corresponds to the saturation temperature of the refrigerant at condensation temperature; and the evaporator and absorber, operate at a lower pressure level which corresponds to the saturation temperature of the refrigerant at evaporator temperature.

The input parameters to the software developed are: Qe (refrigerating effect), t13 (evaporator temperature); t7 (generator solution exit temperature); t1 = t11 (condenser/absorber low temperature); x5 and x1 (solution concentrations at states points 5 and 1).

IV. MATHEMATICAL MODEL RELATIONSHIPS

Some the Mathematical relationships used to develop the software are presented as follows:

Mass equation:

$$\dot{m} = f(m_g, m_c, m_g, \dots \dots \dots) \quad (1)$$

Where

$$m = f(h, t, p, s, \dots \dots \dots) \quad (2)$$

Mass balance equation

$$\sum_{i=1}^n m_i = 0 \quad (3)$$

Concentration equations:

$$x = f(w_g, w_c, w_g, \dots \dots \dots) \quad (4)$$

Where

$$\omega = f(h, t, s, p, \dots \dots \dots) \quad (5)$$

Concentration balance equation

$$\sum_{i=1}^n x_i = 0 \quad (6)$$

Energy equation:

$$E = f(e_e, e_c, e_g \dots \dots \dots) \quad (7)$$

Where

$$e = f(h, t, s, u \dots \dots \dots) \quad (8)$$

Energy balance

$$\sum_{i=1}^n E_i = 0 \quad (9)$$

Also energy balance equation can be expanded to be as shown in equation

$$E = (m_{in} h_{in})_i + \sum \dot{Q} = \sum_{i=1}^n (m_{out} h_{out})_i \quad (10)$$

The governing equations for other parameters areas follow:

Enthalpy (Indrancel, 2003 and McNeely, 1979, Affandi et al, 2013)

$$h = \sum_{n=0}^4 (a_n x^n + b_n x^n T + C_n x^n T^2) \quad (11)$$

Saturation Temperature (T)

$$T^1 = \frac{-2e}{d + [d^2 - 4e(c - \log p)]^{0.5}} \quad (12)$$

$$T = \sum_{n=0}^3 (b_n x^n + a_n x^n T') \quad (13)$$

Saturation Pressure

$$T^1 = \frac{T - \sum b_n x^n}{\sum a_n x^n} b_n x^n \quad (14)$$

$$\log P = C + \frac{D}{T'} + \frac{E}{(T')^2} \quad (15)$$

Solution specific heat

$$\Delta T = (T + 0.00005) - (T - 0.00005) \quad (16)$$

$$c_p = \frac{h_{(t+0.00005)} - h_{(t-0.00005)}}{\Delta T} \quad (17)$$

Mathematical Models for the components of the Double-Effect Reverse parallel VARS.

Each components of the system were developed using mathematical equation as presented below.

V. EVAPORATOR

Mass balance for evaporator:

$$\dot{m}_{12} = \dot{m}_{13} \quad (18)$$

Thermal balance

$$\dot{m}_{13} h_{13} - \dot{m}_{12} h_{12} = Q_E \quad (19)$$

$$Q_E = m_{12} (h_{13} - h_{12}) \quad (20)$$

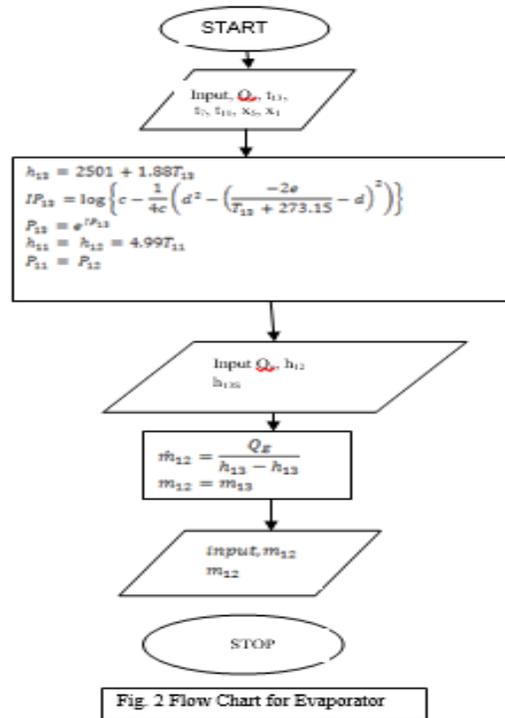
$$m_{12} = \frac{Q_E}{h_{13} - h_{12}} \quad (21)$$

$$h_{13} = 2501 + 1.88T_{13} \quad (22)$$

$$IP_{13} = \log_{10} \left(C - \frac{1}{4c} \left(d^2 - \frac{-2e}{T_{13} + 273.15} - d \right)^2 \right) \quad (23)$$

$$P_{13} = e^{IP_{13}} \quad (24)$$

The flow chart for the modeling of evaporator is shown in Fig 2. Flow chart for Low Temperature Heat exchanger is shown in Fig 3.



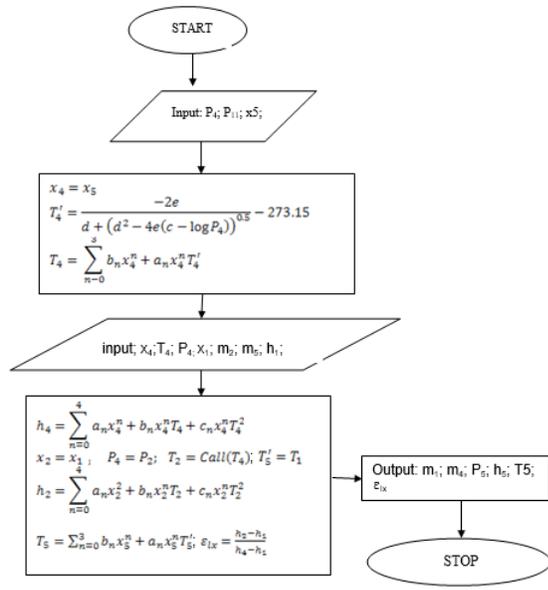


Fig. 3 Flow Chart for low Temperature Heat Exchanger

Charts and modeling equations for other components can be found in the work of Adedoyin (2016).

VI. RESULTS AND DISCUSSIONS

The results of the software developed is as shown in Tables 1 and 2

Table 1: Output parameters from the software

States	h (kJ/kg)	m (kg/s)	P (kPa)	T(°C)	x(%LiBr)
1	117.95	9.557	0.87	42.4	59.5
2	203.85	9.557	8.38	86.47	59.5
3	202.31	4.59	8.38	85.69	59.5
4	247.3	8.8025	8.38	97.77	64.6
5	154.03	8.8025	0.87	53.7	64.6
6	323.3	4.59	111.73	155.68	59.5
7	379.77	4.213	117.73	170.7	64.6
8	247.95	4.213	8.38	97.83	64.6
9	2694.12	0.3773	117.73	102.7	0

10	2672.96	0.3773	8.38	91.47	0
11	177.82	0.7545	8.38	42.4	0
12	177.82	0.7545	8.38	5.1	0
13	2510.4	0.7545	0.87	5.1	0
14	247.8	4.59	8.38	97.83	64.6
15	177.62	0.3773	8.38	42.3	0

Table 2: Estimated values of other parameters

Parameter	COP	ε _{h_x}	ε _{l_x}	Q _a (kW)	Q _c (kW)	Q _{htg} (kW)	Q _e (kW)	Q _{itg} (kW)
Value	1.55	0.68	0.66	2122.7	94.2	113.23	1760	1126.15

The corresponding ASHRAE (1997) values compared with the results in Tables 1 and 2 are shown in Tables 3 and 4.

Table 3: ASHRAE parameter compared with Table 1

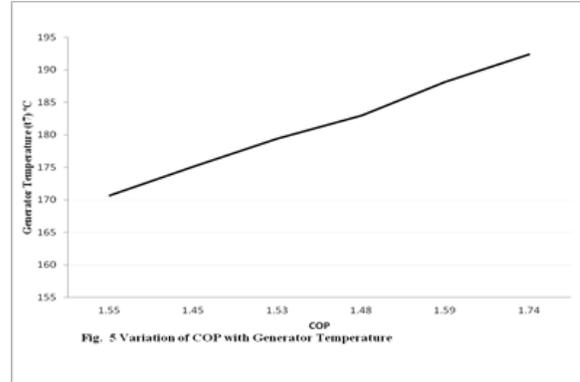
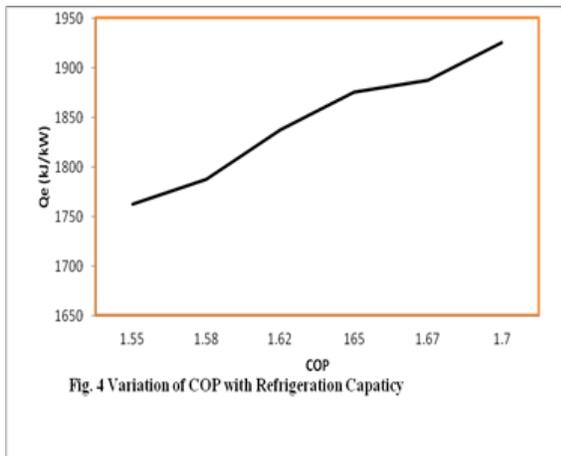
State	H (kJ/kg)	m (kg/s)	P (kPa)	t(°C)	x(%LiBr)
1	117.7	9.551	0.88	42.4	59.5
2	182.3	9.551	8.36	75.6	59.5
3	201.8	5.498	8.36	85.6	59.5
4	247.3	8.797	8.36	97.8	64.6
5	177.2	8.797	0.88	53.2	64.6
6	301.2	5.498	111.8	136.7	59.5
7	378.8	5.064	111.8	170.7	64.6
8	270.9	5.064	8.36	99.1	64.6
9	2787.3	0.434	111.8	155.7	0
10	2661.1	0.32	8.36	85.6	0
11	177.4	0.754	8.36	42.4	0
12	177.4	0.754	8.36	5.1	0

13	2510.8	0.754	0.88	5.1	0
14	247.3	8.797	8.36	97.8	64.6
15	430.6	0.434	8.36	42.4	0

Table 4 ASHRAE parameter compared with Table 2

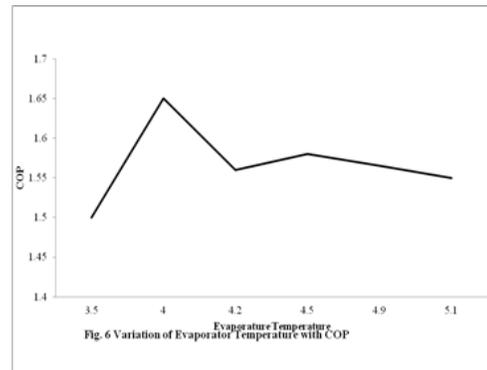
Parameter	Value
COP	1.3
ϵh_x	0.6
ϵl_x	0.6
Q_a	2328
Q_c	905
Q_{htg}	1472
Q_e	1760
Q_{ltg}	1023

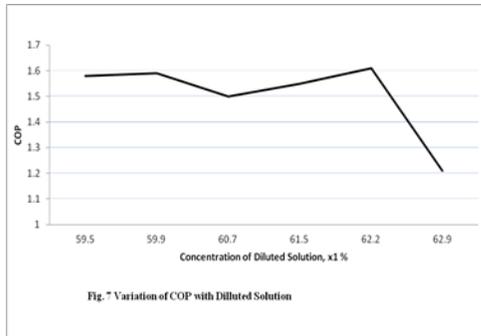
The variation of refrigeration capacity with COP is shown in Fig. 4. From the figure, the COP increases with the refrigeration capacity. During the testing, the capacity of the system was varied between 1760 kW and 1890 kW while keeping other parameters constant. The relationship between the generator temperature and COP is shown in Fig.5.



The value of generator temperature (T_7) was varied between 170.7 °C and 192.4 °C while other parameters were kept constant. It was observed that as the temperature increased to 183 °C, the COP fluctuated but beyond 183 °C, the COP started increasing gradually. The higher the generator temperature the higher the COP of the system.

The variation of COP with evaporator temperature (T_{13}) and concentration of diluted solution (x_1) is shown in Figs 6 and 7. The concentration of the diluted solution in the evaporator (x_1) was varied between 59.5% and 62.9% while keeping other parameters constant. It was observed that as the concentration increases, the COP increases, gradually but decreased as the concentration was increased above 62.2%. COP was maximum when x_1 was 62.2%, also as the solution becomes less diluted, the COP drops. So, for efficiently performance 62.2% is recommended.



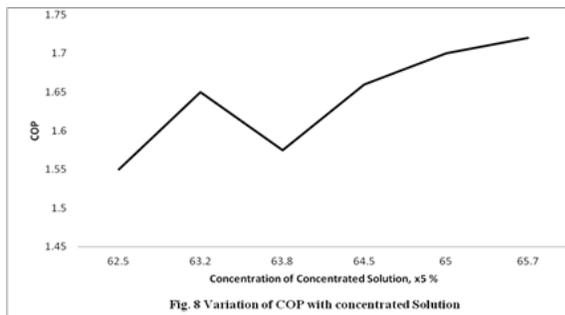


The temperature of the evaporator (T13) was varied between 3.5°C and 5.1°C, all other parameters kept constant. As the evaporator temperature increased to 4.0 °C, the COP was at its highest value of 1.654 and then decreased for higher evaporator temperatures.

The best evaporator temperature for optimum COP is 4.0 °C.

The concentration of the concentrated solution in the absorber (x5) was varied between 62.6% and 65.7%.

It was observed that as the concentration increased from 62.6% to 63.8%, the COP fluctuated between 1.55 and 1.66, but increased gradually as the concentration increased above 63.8%. Hence the higher the value of x5 the higher the COP, (Fig. 8).



CONCLUSION

Mathematical models and computer software for double-effect reverse parallel vapour absorption refrigeration system was developed and provides data for future system design. Available data was used to validate the model.

The software application was evaluated for its performance and various design data could be

generated using the software. The result from the software was comparable with the published data in Fundamentals Handbook (ASHRAE, 1997) and an average of 20% variation was observed. It was also noted that the COP increased with increase refrigeration effect. Though the COP gives initial fluctuation with increase in generator temperature, but at generator temperature of 183°C and above the COP steadily increased with the temperature.

As the diluted concentration in the evaporator (x1) increased, the COP increased gradually but decreased as the concentration reaches 62.2%. As the concentration of the concentrated solution in the absorber increased from 62.6% to 63.8%, the COP fluctuated between 1.55 and 1.66 but increased gradually as the concentration increased above 63.8%. Here the higher the solution concentration, the higher the COP.

REFERENCES

- [1] Adedoyin, F. H. (2016) Development of a Computer-Aided Application for Double-Effect Vapour Absorption Refrigeration System. M. Eng. Thesis at the Federal University of Technology, Akure (unpublished).
- [2] Affandi, M; Mamat N; Kanafiah, S. N. and Khalid, N. S, (2013) simplified Equations for Saturated Steam Properties for Simulation Purposes. Procedia Engineering. Vol. 5 pp 722-726.
- [3] Akintunde, M. A. (2004) Development of Design Model for Vapour Compression Refrigeration Systems. Ph.D Thesis, at the Federal University of Technology, Akure.
- [4] Akintunde, M. A.; Adegoke, C. O. and Fapetu, O. P. (2006) Experimental Investigation of the Performance of a Design Model Indies Journal of Engineering, Vol. 28 No 2, pp 80-87.
- [5] Asdrubali F. and Grignaffini S, (2005) Experimental Evaluation of the Performance of a H₂O/LiBr Absorption Refrigerator under different service conditions. International Journal of Refrigeration, Vol. 28, pp 489-497.
- [6] ASHRAE (1997) HVAC Fundamental Handbook, American Society of Heating, Refrigeration and Air-Conditioning Engineers Inc., Atlanta, GA.

- [7] Azhar M., and Siddiqui M. A., (2013) Thermodynamic Analysis of a Gas Operated Triple Effect Absorption Cycle. International Journal of Innovative Research in Science, Engineering and Technology, Vol. 2, No.5, pp 1610-1616.
- [8] Mohendru, P., Agrawal,, S. K.; and Pachovkar, P. (2012) Steady State Analysis of Vapour Absorption Refrigeration Systems using LiBr-H₂O. International Journal of Engineering Technology and Advanced Engineering. Vol. 2 No.9, pp 328-332.
- [9] McNeedy L. A., (1997) Thermodynamic Properties of Aqueous Solution of Lithium Bromide. ASHRAE Transaction, Vol. 85 No.1, pp 413-434.
- [10] Micallet, D. and Micallet, C. (2010) Mathematical Model of a Vapour Absorption Refrigeration Unit. International Journal of Simulation Modeling, Vol. 9 No. 2, pp 86-97.
- [11] Oberweis, S. and Al-Shemmeri, T. (2012) Performance Evaluation of a Lithium Chloride Absorption Refrigeration and an Assessment of its Suitability for Biomass Waste Heat. Applied Sciences Journal. Vol. 2, No. 4, pp 709-725.
- [12] Sawalhah S. (2016) Solar Assisted Automobile A/C Based on a Water-Lithium Bromide Absorption Refrigeration System. International Journal of Research in Engineering and Advanced Technology, Vol. 3, No 6, pp 49-61.
- [13] Wang R. and GAO H. (2013) Nemerical Study on Vapour Absorption into Aqueous LiBr Solution in a Liquid-Gas Ejector Using Convergent Nozzle. Research Journal of Applied Science, Engineering and Technology, Vol. 5, No 14, pp 3727-3731.