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Experimental Investigation on an Intermittent Ammonia Absorption Refrigeration System

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Abstract- *This work describes the experimental investigation on an intermittent ammonia absorption refrigeration system which consists of a generator/ absorber, a condenser, an evaporator and an expansion valve. The system used 2.5 kg of ammonia as refrigerant and 1.5 kg of calcium chloride as absorbent initially in generator and this NH_3-$CaCl_2$ refrigerant-absorbent pair is widely used in the intermittent refrigeration system. There are two modes where in desorption mode, an oil bath arrangement heat source provides heat to the generator and the generator drives of the vapor form refrigerant around the system through a condenser to evaporator. The evaporator pipes were placed outside of the evaporator container which was insulated and 3 kg of water was filled to be the load for cooling into the evaporator container. During the absorption mode, the liquid refrigerant absorbed heat from the surrounding water and performed cooling. The average refrigeration temperature was found to be 13.25^{0}C with a minimum value of 11.5^{0}C for a period of time up to 2.5 hours' absorption. The coefficient of performance was obtained on the average of 0.154 with a maximum value of 0.192.*

Keywords: Intermittent absorption system, NH_3-$CaCl_2$ refrigerant-absorbent pair, Absorption and desorption mode, Coefficient of performance, Refrigeration effect

1. INTRODUCTION

The vapor absorption refrigeration system gained renewed interest due to environmental problems of commonly used refrigerants in the vapor compression refrigeration system. Being a heat operated system; it is gaining major focus on solar energy based refrigeration system. Various types of vapor absorption system are used in solar energy based refrigeration system due to energy crises is a serious handicap for the socioeconomic development of the rural and urban population [1]. In absorption refrigeration system the vapor is drawn from the evaporator by absorption into a liquid having high affinity for the refrigerant. The refrigerant is expelled from the solution by the application of heat and its temperature is also increased. This refrigerant in the form of vapor passes to the condenser where heat is rejected and the refrigerant gets liquefied [2]. This liquid again flows to the evaporator at reduced pressure. The intermittent refrigeration system actually is not a true refrigeration system because the mass flow rate is not constant throughout the system. The mass flow rate increases when the temperature of the generator increases and the rate of flow of vapor ammonia also increase [3].

The simplest design of this intermittent system consists of three major parts such as a generator/ an absorber for desorption/absorption the salt-ammonia mixture, a condenser, and an evaporator [3-5]. Ammonia flows back and forth between the generator and evaporator. During the step-1 in desorption mode, the container 'P' acts as the generator producing high pressure and high temperature refrigerant vapor by heat from a source. The refrigerant vapor separating from the absorbent sent through the condenser, turned into liquid phase and stored inside the container 'Q' [2].

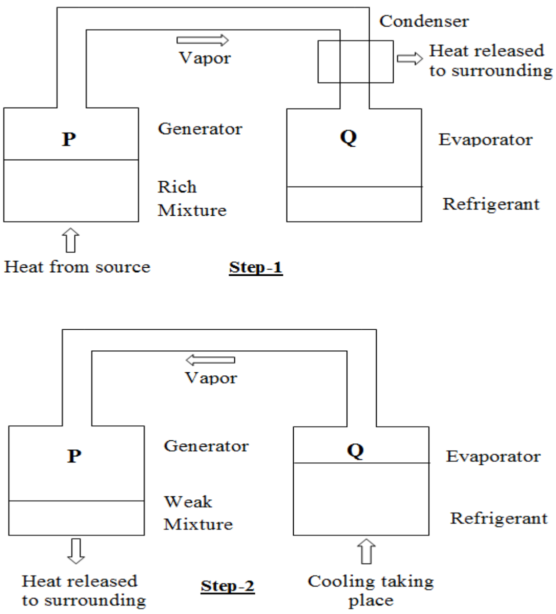

Fig.1: Intermittent absorption cooling system

Again step-2 in absorption mode, the container 'Q' acts as the evaporator absorbing heat from the surroundings. If it is surrounded by water, the cold water or ice can be produced. The refrigerant will be evaporated into the vapor and flow back into the container 'P' acting as the absorber where the concentrated absorbent awaits and both working substances are mixed [3-4].

The performance of the intermittent vapor absorption system depends on generator's temperature and heat addition. However, investigations are required to evaluate the performance of the intermittent vapor absorption with a constant temperature heat addition in the generator. Also the amount of heat addition in the generator is a key parameter to design solar collector for the intermittent vapor absorption system [12-14].

2. DESIGN CONSIDERATIONS

2.1 Selection of Refrigerant and Absorbent Pair

For this work ammonia acts as a refrigerant and **c**alcium chloride acts as an absorbent which is a salt of calcium & chlorine and behaves as a typical ionic halide that white solid at room temperature because the difference between the vaporization temperatures is large [8]. The range of pressure is relatively low compared to other pairs and the system can work with a wide variety of temperatures. Because of hygroscopic nature, anhydrous calcium chloride must be kept in tightly-sealed airtight containers [9].

2.2 Generator Design and Mass Calculation

At the generator, ammonia and calcium chloride refrigerant-absorbent pair was used and heat is provided from the heat source of the oil bath. The reaction between ammonia and calcium chloride are as follows-

$$CaCl_2 + 2NH_3 \leftrightarrow CaCl_2.2NH_3$$

$$CaCl_2 + 4NH_3 \leftrightarrow CaCl_2.4NH_3$$

$$CaCl_2 + 8NH_3 \leftrightarrow CaCl_2.8NH_3 \qquad \Delta H = -78 kJ/mol$$

Thus, 8 moles of NH_3 may be absorbed per mole of $CaCl_2$, during initial charging, but only 6 moles will be available for refrigeration. It is assumed that no heat and pressure losses in components and lines. Assume that 60% of the volume of generator is used for the purpose of well vaporization. So the required ammonia (NH_3) and the calcium chloride ($CaCl_2$) are 2.305 kg and 1.881 kg respectively, which were measured from the design parameters that are detailed on D. Mondal project [11].

2.3 Design of Condenser and Evaporator Size

The design of a refrigerator means designing the length, size, type and material of heat exchanger such as an evaporator and condenser. In single-pass heat exchangers, the temperature difference ΔT between the hot and the cold fluids is not constant, but it varies with distance along the heat exchanger [13]. For the design of condenser tube size, the condenser inlet and outlet temperatures are to be assumed and the condenser water inlet temperature depends exclusively on the available source of water.

Table 1: Considering Parameters for Condenser

Temperature	Value
Condenser input ammonia	90^0C
Condenser output ammonia	32^0C
Condenser input water	25^0C
Condenser output water	35^0C

The mass flow rate of water at 60 liter/hr in the condenser and specific heat of water is 4200 J/kg-C. The fouling factors (F_i , F_o) at the inside and outside surfaces of the tube can be taken as 0.00009 m^2C/W and thermal conductivity (k) for stainless steel is as 17 (W/m-C). The heat transfer coefficients (h_i, ho) for the inside and outside flow can be taken as 7000 W/m^2-C. Again the correction factor [10] of the heat exchanger can be chosen as, F=0.96.

The heat gain by water from the condenser coil, $Q = mC_p \Delta T_m$. In the heat transfer analysis, it is convenient to establish a mean temperature difference between the hot and cold fluids such that the total heat transfer rate Q between the fluids can be determined from the following expression, $Q = AU\Delta T_m$ in where, A (m^2) is the total heat transfer area and U (W/m^2-C) is the average overall heat transfer coefficient [10] based on the outside surface of the tube and mean temperature difference are defined as,

$$U = \frac{1}{(\frac{D_o}{D_i})(\frac{1}{h_i}) + (\frac{D_o}{D_i})F_i + (\frac{1}{2k})\, D_o \ln(\frac{D_o}{D_i}) + F_o + \frac{1}{h_o}} \qquad (1)$$

$$\Delta T_m = F\Delta T_{\ln}(\frac{\Delta T_O - \Delta T_L}{\ln(\Delta T_O / \Delta T_L)}) \qquad (2)$$

Again, for the design of evaporator tube size, it is taken into consideration during the absorption period and the cooling capacity or refrigerating effect of that evaporator. The entering and leaving temperatures of refrigerant are assumed with respect to the pressure of evaporator side. The evaporator temperature is also to be assumed as required on the basis of room temperature [15]. Now, the overall thermal resistance as,

$$R = \frac{1}{Ah_i} + \frac{t}{kA} + \frac{1}{Ah_o} \qquad (3)$$

The overall thermal resistance can be obtained where convective heat transfer coefficient of ammonia is 7000 W/m^2-C and thermal conductivity of stainless steel pipe is 17 W/m-C. Again, the required parameters are found from the ammonia properties table or pressure enthalpy diagrams of R717 and the evaporator heat transfer relation of $Q_e = m_{ref}(h_4 - h_3)$ with respect to corresponding pressure and temperature [10].

Table 2: Considering parameters of evaporator

Parameters	Value
Room temperature	28^0C
Evaporator temperature	10^0C
Evaporator entering temperature	7^0C
Evaporator leaving temperature	20^0C
Pressure at evaporator	5.5 bar
Absorption period	2.50 hrs
Mass of water in the evaporator	3 kg
Cooling capacity	34 W

Table 3: Selection parameter of generator [11] design

Parameters	Generator analysis
Tube outside diameter, D_o	0.0762 m
Tube inside diameter, D_i	0.0732 m
Calculated tube length	2.34 m
Used tube length	2.34 m
Material of the tube used	Stainless steel

Table 4: Selection of heat exchanger [11] size design

Parameters	Heat exchanger size	
	Condenser size	Evaporator size
Outside tube diameter, D_o	12.7 mm	9.525 mm
Inside tube diameter, D_i	10.7 mm	7.525 mm
Calculated tube length	2.13 m	4.22 m
Used tube length	5.5 m	6 m
Material of the tube used	Stainless steel	Stainless steel

Assume that 1.8 kg ammonia out of 2.305 kg will be vaporized and stored in the evaporator after passing through the condenser coil. Hence, neglecting the losses and for the best performance from this system work ammonia is used as of 2.5 kg instead of 2.305 kg and calcium chloride is used as of 1.5 kg instead of 1.881 kg. Again, both of condenser and evaporator heat exchanger sizing, it will be used more length for more heat transfer of length 5.5 m. The length of the condenser is 5.5 m which is higher than desired length and the used length of evaporator tube is 6 m for safety and more heat transfer. The above all of selection parameters are considered from the design parameters that detail on D. Mondal project [11].

3. WORKING PRINCIPLE AND CONSTRUCTION

In this project work the intermittent vapor absorption system was designed and constructed in which ammonia and calcium chloride refrigerant-absorbent pair was used. The system consists of a generator/ an absorber which is charged with ammonia and Calcium Chloride as refrigerant-absorber, a condenser, a storage tank and an expansion valve and evaporator. The heat source was an oil bath where oil was heated up by the electric heater [3]. At first, the generator was constructed and based upon the capacity of the generator than condenser and evaporator was designed and constructed. For constructing the whole system, the stainless steel tube is used because it is more sustainable on high temperature and pressure.

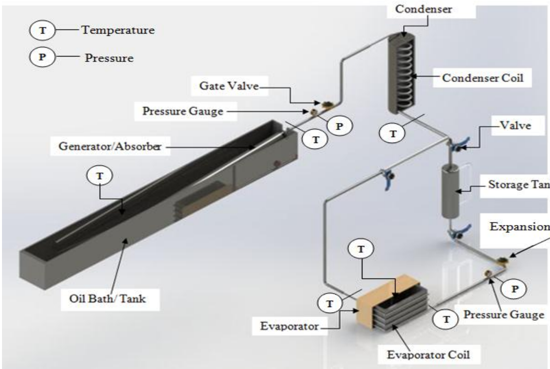

Fig.2: Experimental setup of ammonia absorption refrigeration system

To carry out various measurements and performance tests pressure gauge, thermocouple is installed in various locations on the system [6-8]. In the earlier, the condensed ammonia entering the storage tank during the desorption mode was cool, but not as cool as the temperatures one might hope to maintain in an icebox. The advantages of having a separate storage tank and evaporator is that are able to concentrate on specializing the evaporator to its purpose, that of absorbing heat from the water. During desorption mode, the heat source heated up the generator and the generator drives of the vapor form refrigerant around the system through a condenser to a storage tank when the valve 1 was only open and valve 2-3 remain closed in Fig.3. Before the absorption mode of the cycle fully starts the valve 3 may slightly open which will give some refrigerant effect to the evaporator [5].

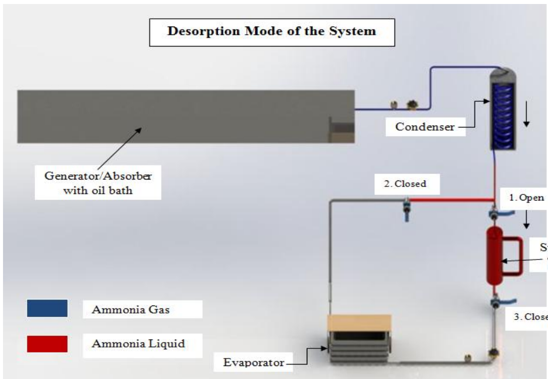

Fig.3: Desorption mode of ammonia absorption refrigeration system

On the other hand, after starting the absorption mode, the valves 2 and 3 remain open and valve 1 remains closed and hence the heat source was removed in Fig.4. The liquid refrigerant from the storage tank flowed through an expansion valve into the evaporator and absorbed heat from the surrounding water and performed cooling. Then the evaporated refrigerant vapor flowed back into the generator which acting as absorber where the concentrated absorbent awaits and both working substances are mixed [6-7].

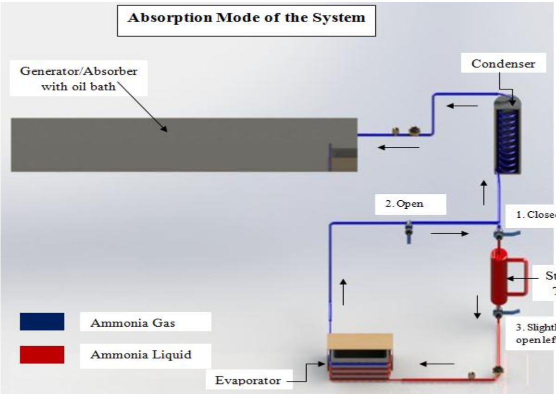

Fig.4: Absorption mode of ammonia absorption refrigeration system

4. RESULTS AND DISCUSSION

In order to experimentally evaluate the COP of the system the ammonia and calcium chloride mixture at the generator/absorber, the evaluating parameters such as temperatures, pressures and heat input from the heat source everything were recorded time to time. A summary of the operation and testing parameters was presented below.

Table 5: Operating and testing parameters of the system during the desorption mode

No. of obs.	Desorption Mode				
	Temp. of heat source (^{0}C)	Temp. of Gen. (^{0}C)	Temp. of Cond. Outlet (^{0}C)	Heat loss (kJ)	Gen. heat gain (kJ)
1	90	76.7	25.3	1312.5	1050.0
2	100	83.7	26.5	1312.5	1443.7
3	105	86.5	27.0	1312.5	1640.6
4	110	91.4	28.3	1312.5	1837.5
Avg.	101.25	84.58	26.78		1492.97

Table 6: Operating and testing parameters of the system during the absorption mode

No. of obs.	Absorption Mode				COP
	Temp. of Evap. (^{0}C)	Product load (kJ)	Heat loss (kJ)	Refrigerating Effect (kJ)	
1	15	151.2	50.4	201.6	0.192
2	14	163.8	50.4	214.2	0.148
3	12.5	182.7	50.4	233.1	0.142
4	11.5	195.3	50.4	245.7	0.134
Avg.	13.25			223.65	0.154

Under the project, the generator pressure was varied from 8.5 bar to 10.5 bar and then the system generated 0.164 kg (164ml) of ammonia [11] liquid in the storage tank during desorption mode when heat supplied by the heat source. The average heat gained by the generator was about 1492.97 kJ during desorption and also keep constant heat loss by the heat source. It could be seen that the generator pressure varies with the heat input from the heat source of the oil bath arrangement and also the generator, heat gain was fairly related to the heat input by considering the heat losses from the heat source of the oil bath.

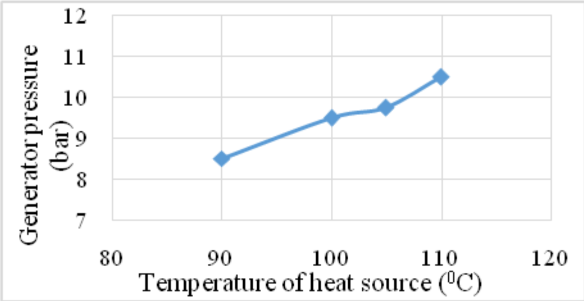

Fig. 4: Variation of generator pressure with the temperature of heat source

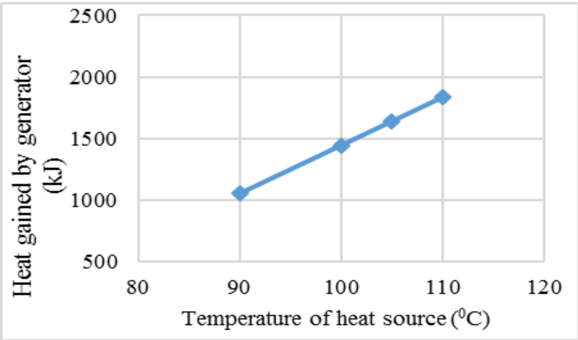

Fig. 5: Variation of heat gained by the generator with the temperature of heat source

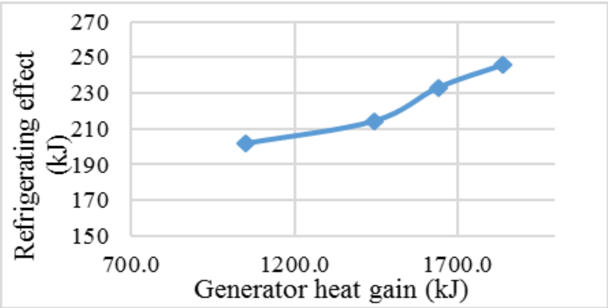

Fig. 6: Variation of refrigerating effect with the generator heat gain

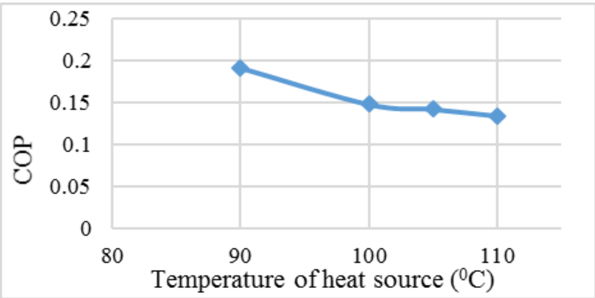

Fig.7: Variation of COP of the system with the temperature of heat source.

It could also be seen that the average refrigeration time varies with time and during the absorption mode, the lowest evaporator (i.e, refrigeration compartment) temperature was found as 11.5^{0}C whereas average of 13.25^{0}C. From Fig.4 and Fig.5, it could be seen that the generator pressure varies and also the generator, heat gain was fairly related to the heat input by considering the heat losses from the heat source. Again, from Fig.6, it could be seen that the refrigerating effect increased by the heat gained by the generator which as similar as the Katejanekarn [2].

The coefficient of performance (COP) of the system was obtained as the average of 0.154 with a maximum value of 0.192. As shown in Fig.7, the coefficient of performance also decreased with the increase of temperature of the heat source. The result of the COP was higher than the Vanek [1], Katejanekarn [2] developed an intermittent cooling system having the COP of 0.029 and Moreno-Quintanar [3] developed an intermittent system for ice production having the COP as 0.098. But lower in the Tangka [4] developed system having the COP of 0.487, Srinivasa Rao [5] absorption system, obtaining the COP of 0.69 and also the Aphornratana [6] & Bell et al [7] developed system. During the evaporation stage, the ammonia pressure descends until 5 to 6 bars of varying temperature from 10.5^{0}C to 13.5^{0}C and obtained 164 ml ammonia [11] cools 300 ml water from 27^{0}C to 11.5^{0}C.

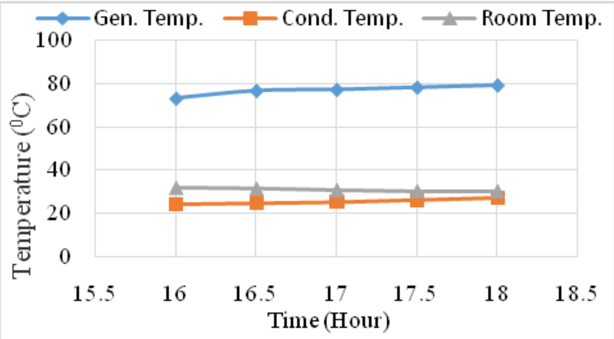

Fig. 8: Temperatures profile (sample-1) during the desorption mode

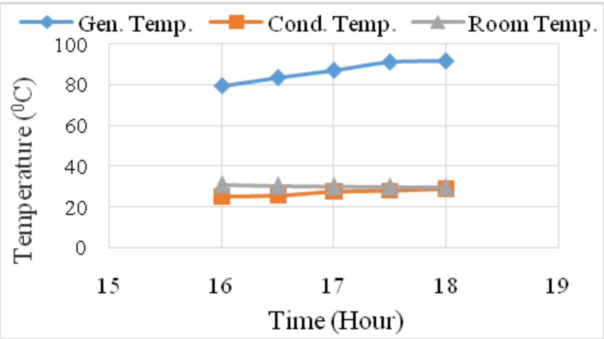

Fig. 9: Temperatures profile (sample-2) during the desorption mode

From Fig.8 to Fig.10, it was said that the temperature of condenser increased by the increase of the heat inputs of the generator to the system by almost constant room temperature with respect to time. Again from Fig.11 to Fig.13, it was seen that the temperature of the evaporator (i.e, evaporator compartment water) almost decreased wherewith the evaporator inlet temperature was same on average but the evaporator outlet temperature increased with the certain limit.

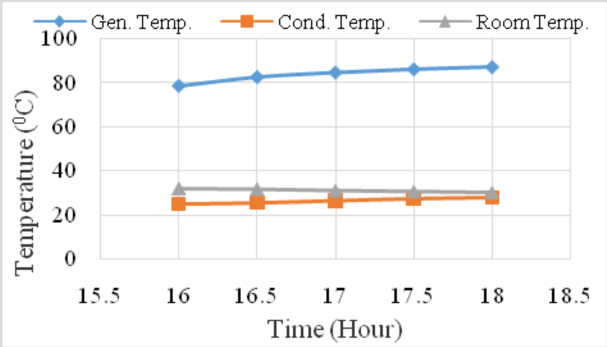

Fig.10: Temperatures profile (sample-3) during the desorption mode

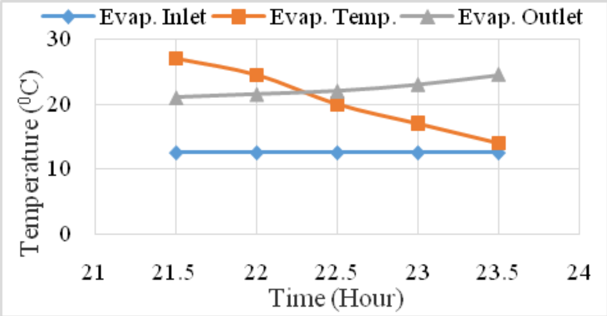

Fig. 11: Temperatures profile (sample-1) during absorption mode

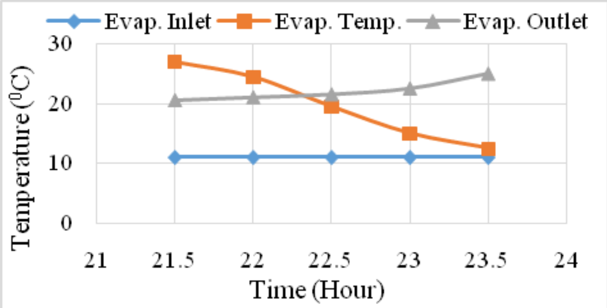

Fig. 12: Temperatures profile (sample-2) during absorption mode

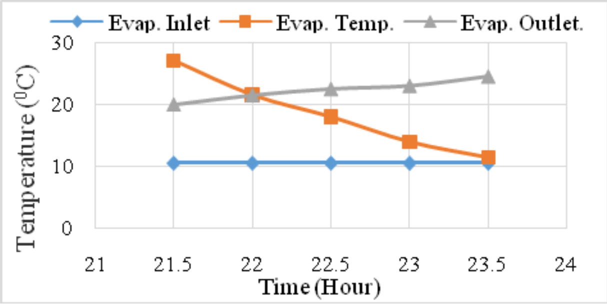

Fig. 13: Temperatures profile (sample-3) during absorption mode

5. CONCLUSION

Performances of the system with various generator temperature and heat addition were investigated. From the experimental investigations the followings could be concluded.

The average refrigeration temperature was found to be 13.25^0C with a minimum value of 11.5^0C and a maximum of 15^0C for a period of time up to 2.5 hours' absorption.

The generator temperature was found in 91.40^0C corresponding to the oil bath temperature up to 110^0C during 2.5 hours' desorption and the pressure was to be found 10.5 bars. It could be drawn that the refrigerating effect increased by the generator heat gained. The coefficient of performance of the system was obtained on the average of 0.154 with a maximum value of 0.192.

On an average the system could provide the heat gained by the generator of 1492.96 kJ and the refrigeration effect of 223.65kJ.

6. ACKOWLEDGEMENT

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7. REFERENCES

[1] J. Vanek, M. Green and S. Vanek, *A Solar Ammonia Absorption Icemaker*, Home Power#53, http://homepower.com/files/solarice.pdf, 1996.

[2] Katejanekarn, T. and Hudakorn, T., "An Intermittent Solar Absorption Cooling System Using a Parabolic Trough", *J SciTechnol MSU*, Vol 31, No 5, pp.496-502, September-October 2012

[3] G. Moreno-Quintanar, W. Rivera*, R. Best, "Development of a solar intermittent refrigeration system for ice production", *World Renewable Energy Congress 2011-Sweden, Solar Thermal Applications (STH)*, 8-13 May 2011, Linkoping, Sweden

[4] J. K. Tangka and N. E. Kamnang, "Development of a simple intermittent absorption solar refrigeration system", *International Journal of Low Carbon Technologies1/2*,(http://ijlct.oxfordjournals.org/content/1/2/127.full.pdf)

[5] K.V.N. Srinivasa Rao, B.J.M. Rao, "Low Cost Solar Cooling System", *International Journal of Engineering and Innovative Technology (IJEIT)*, Volume 3, Issue 4, October 2013, ISSN: 2277-3754, ISO 9001:2008 Certified

[6] Aphornratana, S., Eames, I.W., Thermodynamic analysis of absorption refrigeration cycles using second law of thermodynamics method, *International Journal of Refrigeration*, Vol. 18(4), (1995), pp. 244-252

[7] Bell, I.A., Al-Daini, A.J., Al-Ali, Habib., Abdel-Gayed, R.G., and Duckers, l., The design of an evaporator/absorber and thermodynamic analysis of a vapour absorption chiller driven by solar energy, *World Renewable Energy Congress*, (1996), pp. 657-660

[8] http://en.wikipedia.org/wiki/Refrigeration

[9] J. P. Holman, 2004 "*Heat Transfer*", Ninth Edition, Tata McGraw-Hill publishing company limited, New Delhi 110 095, ISBN 0-07-058874-0.

[10] M. N. Ozisik, 1985. "*Heat Transfer A Basic Approach*", International Edition 1985, McGraw-Hill Book Company. ISBN 0-07-066460-9.

[11] D. Mondal, "Experimental Investigation on an Intermittent Ammonia Absorption Refrigeration System", *A project report of M.Sc. Eng. (ME) degree*, Department of Mechanical Engineering, Khulna University of Engineering & Technology, Khulna-9203, Bangladesh, September 2014.

[12] Aphornratana S., "Theoretical and experimental investigation of a combined ejector-absorption refrigerator." *PhD thesis*, University of Sheffield, UK, 1995.

[13] Grossman G., 1991, "Absorption heat transformer for process heat generation from solar ponds." *ASHRAE Trans; 97:420–7.*

[14] Kaushik SC, Kumar R., 1987, "A comparative study of an absorber heat recovery cycle for solar refrigeration using NH3-refrigerant with liquid/solid absorbents." *Energy Res;* 11:123–32.

[15] Carlos Rivera, Elizabeth Méndez, Isaac Pilatowsky, Wilfrido Rivera, "Experimental Evaluation of Barium Chloride-Ammonia in an Absorption Solar Refrigeration Prototype", *RIO 3 - World Climate & Energy Event*, 1-5 December 2003, Rio de Janeiro, Brazil, pp. 183-187

8. NOMENCLATURE

Symbol	Meaning	Unit
U	Overall heat transfer coefficient	$(W/m^2.^0C)$
COP	Coefficient of performance	Dimensionless
ΔT_{ln}	Logarithmic mean temperature difference	(^0C)
k	Thermal conductivity of the material	$W/m\ .^0C$
Q	Heat transfer	kJ
C_P	Specific heat of fluid	kJ/Kg-C
D	Diameter of tube material	m
A	Heat transfer surface area	m^2
L	Length of heat exchanger	m
P	Saturation pressure	bar
h	Enthalpy	kJ/ kg
Gen.	Generator	-
Evap.	Evaporator	-
Cond.	Condenser	-
Ref.	Refrigerator	-