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EXPERIMENTAL INVESTIGATION ON AN INTERMITTENT AMMONIA ABSORPTION REFRIGERATION SYSTEM

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Abstract: The principal goal of this work is to investigate the performance of an intermittent ammonia absorption refrigeration system. In this work, the widely used ammonia- calcium chloride (NH_3-$CaCl_2$) refrigerant-absorbent pair is chosen where NH_3 as refrigerant and $CaCl_2$ as absorbent. This intermittent absorption refrigeration system consists of a generator/ absorber, a condenser, an evaporator and an expansion valve, which basically works in two modes or stages. In desorption mode, heat source provided heat to the generator and the generator driven of the vapor form refrigerant around the system through a condenser and then sent into the storage tank. An electrically heated constant temperature oil bath is used to provide heat to the generator. In absorption mode, the liquid refrigerant flowed through evaporator coil that absorbed heat from the surrounding water and performed cooling. The evaporator coil is placed in a water bath and temperature reduction of a fixed amount of water is measured to get the cooling performance. The average heat gain of the generator was 1492.97 kJ during 2.5 hours of desorption operation at a maximum temperature of 91.40°C and a pressure of 10.5 bar. The average refrigeration effect was 223.65 kJ at an average temperature of 13.25°C during 2.5 hours of absorption operation. The average coefficient of performance (COP) was obtained 0.154 with a maximum value of 0.192.

Keywords: *Intermittent absorption refrigeration, NH_3-$CaCl_2$ refrigerant-absorbent pair, Absorption and desorption mode, Refrigerating effect, Coefficient of performance.*

NOMENCLATURE

A = heat transfer surface area, m^2
C_p = specific heat of fluid, kJ/kg-C
COP = coefficient of performance
D = diameter of the tube, m
h = enthalpy, kJ/kg
k = thermal conductivity of the material, W/m -C
L = length of heat exchanger, m
P = saturation pressure, bar
Q = heat transfer, kJ
U= overall heat transfer coefficient, W/m^2 -C
ΔT_{ln} = logarithmic mean temperature difference, °C
Temp. = temperature
Gen. = generator
Cond. = condenser
Evap. = evaporator

1. INTRODUCTION

The vapor absorption refrigeration system gains renewed attention due to the environmental impacts 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 systems are developed [1–3] to use in solar energy-based refrigeration system as energy crises are a serious handicap for the socioeconomic development of the rural and urban population. Moreno-Quintanar *et al.* [3] developed an intermittent refrigeration system where the refrigerant is expelled from the solution by the application of heat and its temperature is also increased. The refrigerant in the form of vapor passes to the condenser where heat is rejected, and the refrigerant gets liquefied. This liquid again flows to the evaporator at reduced pressure. Katejanekarn and Hudakorn [2] developed a cooling system where the vapor is drawn from the

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evaporator by absorption into a liquid having high affinity for the refrigerant. The intermittent absorption is not a true refrigeration system because the mass flow rate is not constant throughout the system. The mass flow rate of the refrigerant increases when the temperature of the generator increases and the rate of flow of vapor ammonia also increase.

The simplest design of the intermittent absorption system [3–5] consists of three major parts such as a generator/ an absorber for the desorption/absorption of the salt-ammonia mixture, a condenser, and an evaporator. In this work, the ammonia flows back and forth between the generator and evaporator.

During the desorption mode as shown in Fig. 1 at step-1, the container '**P**' acts the generator producing high pressure and high temperature refrigerant vapor having the rich mixture of refrigerant-absorbent pair. The refrigerant vapor is separated from the absorbent by heat from a source and then sent through the condenser, turned into liquid phase and stored inside the container '**Q**' as per Katejanekarn and Hudakorn [2].

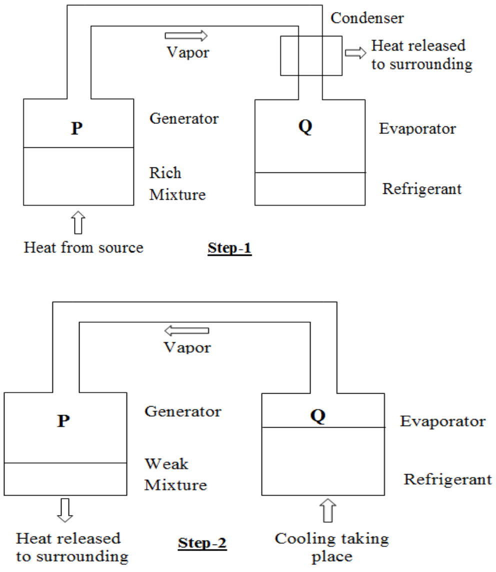

Fig.1 Intermittent absorption cooling system.

Again, from the Fig. 1 in absorption mode at step-2, the container '**Q**' acts as evaporator [3–4] and absorbed heat from the surroundings. If it is surrounded by the water, the cold water or ice is to be produced. Then the refrigerant is evaporated into the vapor and flowed back into the container '**P**' acting as the absorber where the concentrated absorbent awaits and both working substances are mixed.

The performance of the intermittent absorption system depends on generator's temperature and heat addition. However, the investigations are required to evaluate the performance of the intermittent absorption system with a constant temperature heat addition in the generator. Also, the amount of heat addition in the generator is a key parameter to design the intermittent vapor absorption system [6–8]. In this work, the performance of the system is investigated where the cooling performance is measured at evaporator in the absorption mode and the heat addition is measured at generator in the desorption mode.

2. DESIGN

2.1 Selection of Refrigerant and Absorbent Pair

In 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 [9-10]. The range of pressure is relatively low compared to other pairs and the system works with a wide variety of temperatures. Because of the hygroscopic nature, anhydrous calcium chloride must keep in tightly-sealed airtight containers. However, the necessary chemicals used are listed below shown in Table 1.

Table 1 Chemicals used in this work

Calcium chloride ($CaCl_2$)	Assay of 94% to 97%
Ammonia (NH_3)	99.995 % pure

2.2 Design of the Generator

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

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

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

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

Thus, 8 moles of NH_3 are 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 are calculated from the design parameters and details are explained elsewhere [11]. The selecting parameters for the generator are shown in Table 2 and details are given elsewhere [11].

Table 2 Selection parameters of the generator design

Parameters	Dimensions
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

2.3 Design of the Condenser and Evaporator

The design of a refrigerator means designing the length, size, type and material of heat exchanger such as an evaporator and condenser. Heat exchangers are one of the most critical components in any liquefaction/refrigeration system [12], because its effectiveness governs the efficiency of the whole system. 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 [7, 12, 13]. For the design of condenser coil as shown in Table 3 and Table 5, the condenser inlet and outlet temperatures are assumed, and the condenser water inlet temperature depends exclusively on the available source of water.

Table 3 Parameters considered of the condenser coil design

Temperature	Value
Condenser inlet ammonia	90ºC
Condenser outlet ammonia	32ºC
Condenser inlet water	25ºC
Condenser outlet water	35ºC

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 are taken as 0.00009 m^{2}C/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 are taken as 7000 W/m^2-C. Again, the correction factor by Ozisik [10] of the heat exchanger is chosen as, F=0.96.

The heat gain by water from the condenser coil, $Q = mC_p \Delta \mathrm{T}_m$. 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 is to determine from the following expression, $Q = AU\Delta \mathrm{T}_m$ in where, *A* (m^2) is the total heat transfer area and *U* (W/m^2-C) is the average overall heat transfer coefficient measurement by Ozisik [10], Mondal *et al.* [17] and Ahmed *et al.* [18] based on the outside surface of the tube and mean temperature difference are defined as,

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

$$\Delta \mathrm{T}_m = F \Delta \mathrm{T}_{\ln} (\frac{\Delta \mathrm{T}_O - \Delta \mathrm{T}_L}{\ln(\Delta \mathrm{T}_O / \Delta \mathrm{T}_L)}) \tag{2}$$

Again, for the design of evaporator coil shown in Table 4 and Table 5, it is taken into consideration during the absorption period for 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 based on room temperature as per Rivera *et al.* [14]. Now, the overall thermal resistance by Holman [9] and Ozisik [10] as,

$$R = \frac{1}{Ah_i} + \frac{t}{kA} + \frac{1}{Ah_o} \tag{3}$$

The overall thermal resistance is 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 4 Parameters considered for the evaporator coil design

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

Table 5 Selection of heat exchanger coil sizing

Parameters	Heat exchanger size	
	Condenser	Evaporator
Outside tube diameter, D_o	0.127 m	0.09525 m
Inside tube diameter, D_i	0.107 m	0.07525 m
Calculated tube length	2.13 m	4.22 m
Used tube length	5.5 m	6 m

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, the 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 is used more length than the calculated length for safety and for more heat transfer rate. The lengths of the condenser and evaporator are 5.5 m and 6 m respectively, which are higher than calculated lengths that are shown in Table 5. In this work, the condenser-evaporator coil and other tube material are used at stainless steel because ammonia has a high affinity to react with the copper tube. The above all of selection parameters are considered from the design parameters that are detailed in Mondal [11].

3. WORKING PRINCIPLE AND CONSTRUCTION

An intermittent vapor absorption system was designed and constructed in this work as shown in Fig. 2, where ammonia and calcium chloride refrigerant-absorbent pair were used. The system consists of a generator/ an absorber which was charged with a refrigerant-absorber pair, a condenser, a storage tank and an expansion valve and evaporator. The heat source was an oil bath arrangement where oil was heated up by the electric heater as per Moreno-Quintanar *et al.* [3]. At first, the generator was constructed and then condenser and evaporator were designed and constructed based upon the capacity of the generator. For constructing the whole system, the stainless-steel tubes were used because of high sustainability on high temperature and pressure. To carry out the various measurement as per Aphornratana and Eames [15] and Bell *et al.* [16], pressure gauges and thermocouples were installed in various locations on this experimental setup.

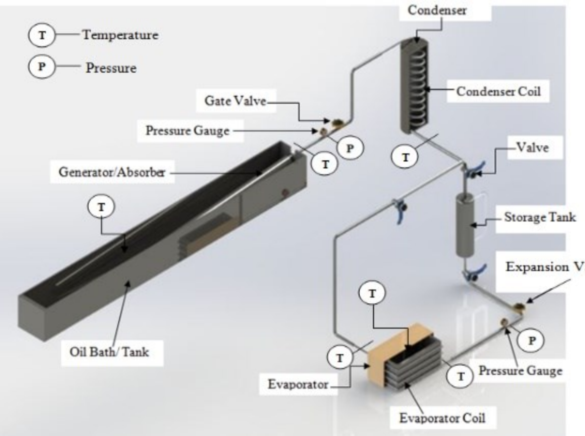

Fig. 2 Experimental setup of ammonia absorption refrigeration system.

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 an evaporator as per Srinivasa Rao and Rao [5], which can concentrate on specializing the evaporator to its purpose, that of absorbing heat from the water. During desorption mode shown in Fig.3, a constant temperature oil bath-based heat source heated up the generator and the generator driven of the vapor form refrigerant around the system through a condenser to a storage tank when the valve 1 was only opened and valve 2-3 remain closed. Before the absorption mode of the cycle fully starts the valve 3 may slightly open which will give some refrigerant effect to the evaporator.

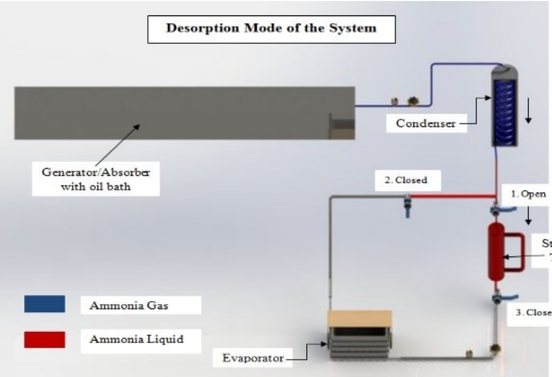

Fig. 3 Desorption mode of ammonia absorption refrigeration system.

On the other hand, after starting the absorption mode shown in Fig.4, the valves 2 and 3 were opened and valve 1 remains closed and hence the heat source was removed. As per Aphornratana and Eames [15] and Bell *et al.* [16], 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.

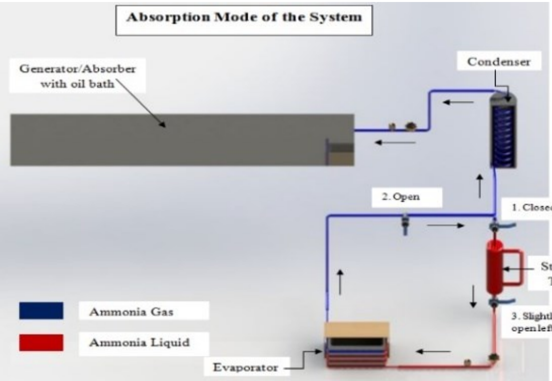

Fig. 4 Absorption mode of ammonia absorption refrigeration system.

4. RESULTS AND DISCUSSION

In order to experimentally evaluate the COP of the system the ammonia and calcium chloride mixture were at the generator/absorber and the operating parameters, such as temperatures, pressures and heat input from the heat source were recorded time to time. A summary of the operating and testing parameters for the system was presented below in the Table 6 & Table 7 that are given on Mondal [11].

Table 6 Operating and testing parameters of the system during the desorption mode

No. of obs.	Desorption Mode				
	Temp. of heat source (ºC)	Temp. of Gen. (ºC)	Temp. of Cond. Outlet (ºC)	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 7 Operating and testing parameters of the system during the absorption mode

No. of obs.	Absorption Mode				COP
	Temp. of Evap. (ºC)	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 shown in Fig. 5 and then the system generated 0.164 kg (164ml) of ammonia liquid by Mondal [11] developed system and store into the storage tank during desorption mode when the heat was supplied by the heat source. It was seen that the generator pressure varies with the heat input from the heat source of the oil bath arrangement. The average heat gained by the generator was about 1492.97 kJ during desorption and kept constant heat loss by the heat source. It was noticed from the Fig. 6 that the generators' heat gain was fairly related to the heat input by considering the heat losses of the heat source.

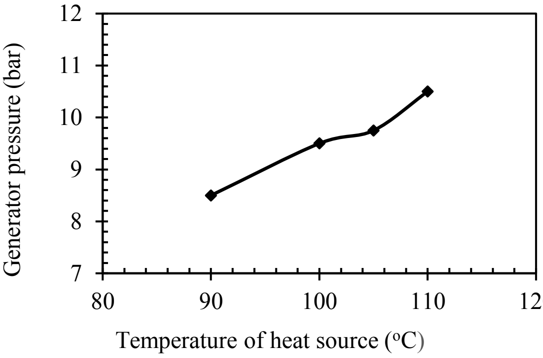

Fig. 5 Variation of generator pressure with the heat source temperature.

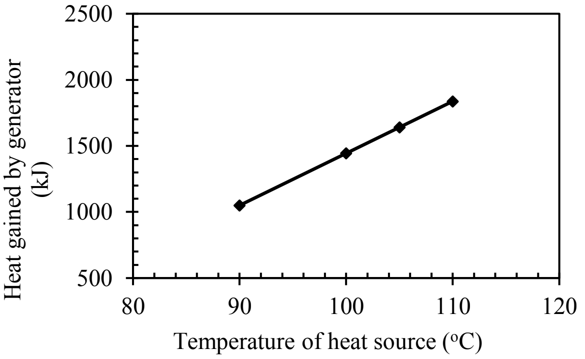

Fig. 6 Variation of heat gained by the generator with the heat source temperature.

It was observed that the average refrigeration means the evaporator temperature varies with time and during the absorption mode, the lowest evaporator (i.e, refrigeration compartment) temperature was found as 11.5°C whereas average of 13.25°C. Again, from the Fig. 7, it was said that the refrigerating effect increased with respect to the heat gained by the generator and was measured as the average of 223.65 kJ. The nature of this refrigerating effect was almost as similar as the Katejanekarn and Hudakorn [2] developed system.

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 the Fig. 8, the COP also decreased with the increase of the temperature of the heat source. This result of the COP was higher than the Vanek [1], Katejanekarn and Hudakorn [2] and Moreno-Quintanar *et al.* [3] developed intermittent absorption system. But, lower than the Tangka and Kamnang [4], Srinivasa Rao and Rao [5], Aphornratana and Eames [15], Bell et al. [16] and Mondal *et al.* [17] developed absorption system.

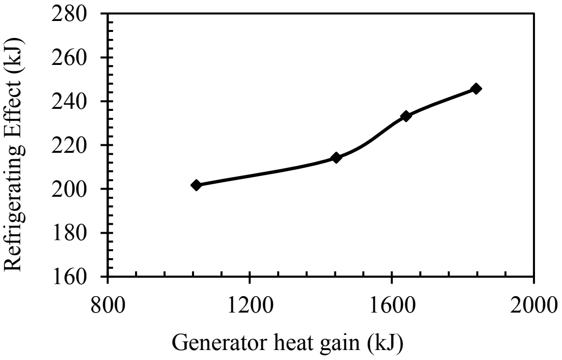

Fig. 7 Refrigerating effect as a function of generator heat gain.

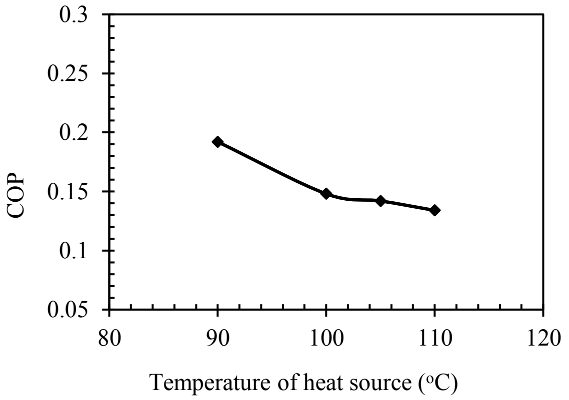

Fig. 8 COP of the system as the function of the heat source temperature.

From the Figs. 9 to 11, it was said that the temperatures of the condenser and generator were gradually increased, respectively, by the increase of the heat inputs of the generator to the system and almost kept constant room temperature with respect to time. Again, from the Fig. 12 to Fig. 14, it was noticed 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 with time.

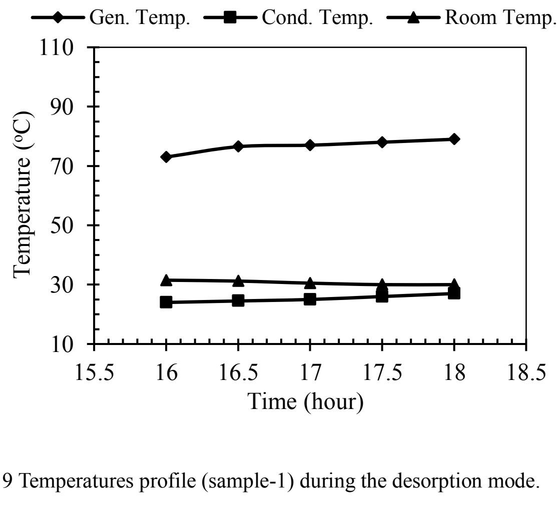

Fig. 9 Temperatures profile (sample-1) during the desorption mode.

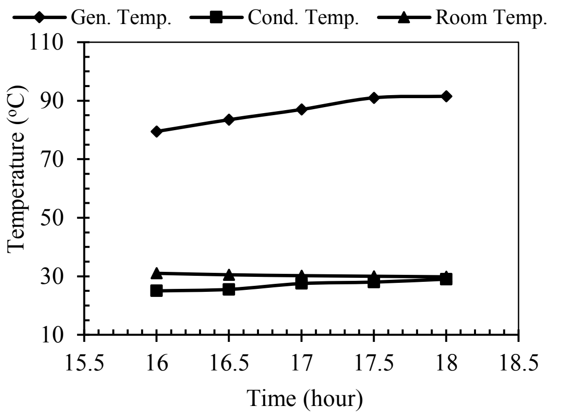

Fig. 10 Temperatures profile (sample-2) during the desorption mode.

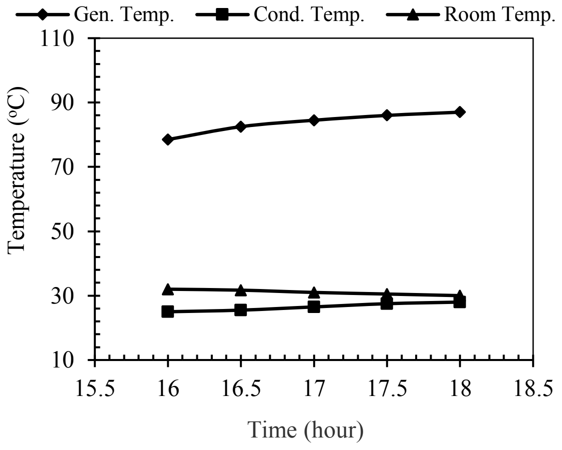

Fig. 11 Temperatures profile (sample-3) during the desorption mode.

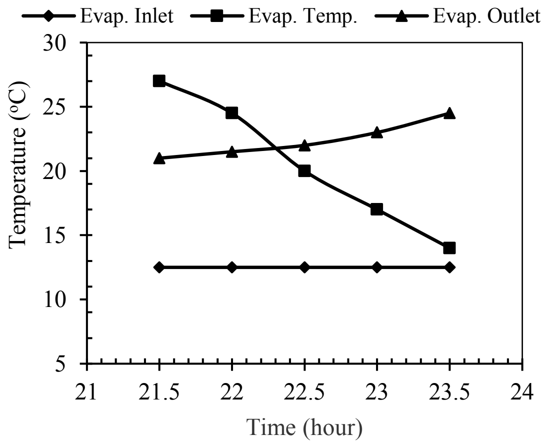

Fig. 12 Temperatures profile (sample-1) during absorption mode.

At last, in the absorption or evaporation stage, the ammonia pressure descended until 5 to 6 bars of varying temperature from 10.5 °C to 13.5 °C and the obtaining 164 ml ammonia cooled 300ml water from 27 °C to 11.5 °C that are detailed on Mondal [11] developed system.

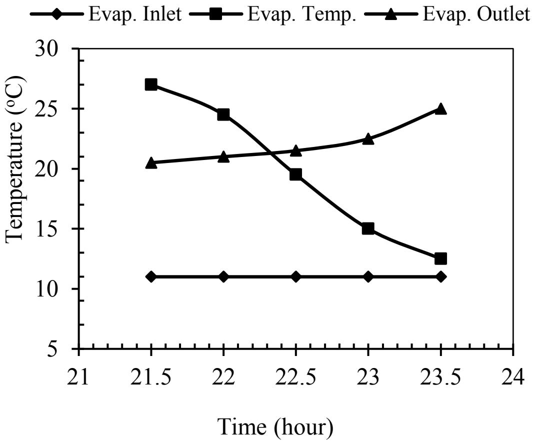

Fig. 13 Temperatures profile (sample-2) during absorption mode.

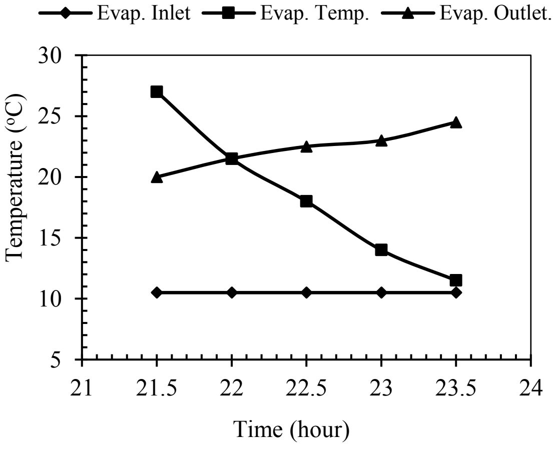

Fig. 14 Temperatures profile (sample-3) during absorption mode.

5. CONCLUSIONS

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

(i) The maximum generator temperature was found in 91.40°C corresponding to the oil bath temperature up to 110°C during 2.5 hours' desorption and the pressure was found to 10.5 bar. On an average heat gained by the generator was found to 1492.97 kJ.

(ii) The average evaporator temperature was found in 13.25°C with a minimum value of 11.5°C and a maximum of 15°C for a period up to 2.5 hours' absorption.

(iii) On an average the system provided the refrigeration effect of 223.65 kJ. It was drawn that the refrigerating effect increased by the generator heat gained. The coefficient of performance of the system was measured on the average of 0.154 with a maximum value of 0.192.

6. ACKNOWLEDGMENTS

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REFERENCES

[1] J. Vanek, M. M. Green and S. Vanek, “A solar ammonia absorption icemaker”, Home Power#53, pp. 20-23, 1996. (https://www.scribd.com/document/13138922/Solar-Icemaker).

[2] T. Katejanekarn and T. Hudakorn, “An intermittent solar absorption cooling system using a parabolic trough”, Journal of Science and Technology MSU, Vol. 31, Issue 5, pp. 496–502, 2012.

[3] G. Moreno-Quintanar, W. Rivera and R. Best, “Development of a solar intermittent refrigeration system for ice production”, World Renewable Energy Congress 2011-Sweden, Solar Thermal Applications (STH), Linkoping, Sweden, pp. 4033–4040, 2011.

[4] J. K. Tangka and N. E. Kamnang, “Development of a simple intermittent absorption solar refrigeration system”, International Journal of Low Carbon Technologies, Vol. 1, Issue 2, pp. 127–138, 2006.

[5] K. V. N. Srinivasa Rao and B. J. M. Rao, “Low cost solar cooling system”, International Journal of Engineering and Innovative Technology (IJEIT), Vol. 3, Issue 4, pp. 58–61, 2013.

[6] S. Aphornratana, “Theoretical and experimental investigation of a combined ejector-absorption refrigerator”, Ph. D. Thesis, The University of Sheffield, UK, 1994.

[7] G. Grossman, “Absorption heat transformer for process heat generation from solar ponds”, ASHRAE Transactions, Vol. 97, pp. 420–427, 1991.

[8] S. C. Kaushik and R. Kumar, “A comparative study of an absorber heat recovery cycle for solar refrigeration using NH_3-refrigerant with liquid/solid absorbents”, International Journal of Energy Research, Vol. 11, Issue-1, pp. 123–132, 1987.

[9] J. P. Holman, Heat Transfer, 9th ed., Tata McGraw-Hill Publishing Company Limited, New Delhi, 110 095, ISBN 0-07-058874-0, 2004.

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

[11] D. Mondal, “Experimental investigation on an intermittent ammonia absorption refrigeration system”, A project report of M. Sc. Eng. (ME), Khulna University of Engineering & Technology, Khulna-9203, Bangladesh, 2014.

[12] D. Mondal, M. O. Ikram, M. F. Rabbi and M. N. A Moral, “Experimental investigation and comparison of bend tube parallel flow & counter flow and cross flow water to air heat exchanger”, International Journal of Scientific & Engineering Research (IJSER), ISSN 2229-5518, Vol. 5, Issue 7, pp. 686–695, 2014.

[13] D. Mondal, M. O. Ikram, M. F. Rabbi and M. N. A Moral, “Experimental analysis of bend tube parallel flow and counter flow water to air heat exchanger”, Proceedings of the 7th IMEC & 16th Annual Paper Meet 2014, Mechanical Engineering Division (MED), IEB, Dhaka, Bangladesh, January 02–03, 2015.

[14] C. Rivera, E. Méndez, I. Pilatowsky and W. Rivera, “Experimental evaluation of barium chloride-ammonia in an absorption solar refrigeration prototype”, RIO 3 - World Climate & Energy Event, Rio de Janeiro, Brazil, pp. 183–187, 2003.

[15] S. Aphornratana and I. W. Eames, “Thermodynamic analysis of absorption refrigeration cycles using second law of thermodynamics method”, International Journal of Refrigeration, Vol. 18, Issue 4, pp. 244–252, 1995.

[16] I. A. Bell, A. J. Al-Daini, H. Al-Ali, R.G. Abdel-Gayed and L. Duckers, “The design of an evaporator/absorber and thermodynamic analysis of a vapor absorption chiller driven by solar energy”, Renewable Energy, Vol. 9, Issues 1-4, pp. 657–660, 1996.

[17] D. Mondal, Al-Imran, M. A. Alam and M. A. Islam, “Experimental observation of a small capacity vapor absorption cooling system”, International Journal of Scientific & Engineering Research (IJSER), ISSN 2229-5518, Vol. 5, Issue 8, pp. 456–467, 2014.

[18] F. Ahmed, D. Mondal and M. A. Islam, “Design, construction and performance test of generator and condenser for a small capacity vapor absorption cooling system”, International Conference on Mechanical, Industrial and Energy Engineering (ICMIEE 2014), KUET, Khulna, Bangladesh, December 26-27, 2014.