ISSN 2229-5518

Experimental Observation of a Small Capacity Vapor Absorption Cooling System

Dipayan Mondal, Al-Imran, Md. Alamgir Alam, Mohammad Ariful Islam

Abstract—This paper work indicates the experimental investigation and performance evaluation of a small capacity of vapor absorption cooling system. With the development in the field of refrigeration, cooling and heat transforming systems, the vapor absorption cycle has gained renewed interest due to environmental and electricity availability problems of commonly used refrigerants in vapor compression system. The most common refrigerant-absorbent pair is water-LiBr. It is the most popular choice in absorption cooling. A small capacity vapor absorption system is first analyzed and characteristics at various points are measured. Components like absorber, evaporator, condenser and generator are designed based on capacity 2 kW. Heat exchanger sizing are made based on type of heat exchanger used. The necessary heat and mass transfer equations and appropriate equations describing the working properties are specified. The difference between absorber LiBr inlet outlet percentage ratio, the COP of the unit, variation of absorption rate with LiBr weight percentage, variation of evaporator outlet temperature are examined. The experimentally obtained cop is 0.32 which is less than the design COP 0.58. Information on designing the heat exchangers of the unit is also presented. The calculated theoretical values are then compared to experimental results derived for a small unit with nominal capacity of 2 kW. Absorber heat rejection increases with increasing absorber temperature. Evaporator heat exchanger exit temperature decreases with increasing pressure at evaporator.

Index Terms— Vapor absorption, intermittent cooling, water/Li-Br refrigerant, Performance evaluation

1 INTRODUCTION

AN absorption refrigeration system uses heat source to provide the energy needed to drive the cooling system. Absorption refrigeration system is a popular alternative to regular compressor refrigeration system where electricity is unreliable, costly or unavailable, where noise from the compressor is problematic or where surplus heat is available. In hot climates, the heating and cooling demand of domestic dwellings can be reduced substantially with various measures such as good insulation, double glazing, use of thermal mass and ventilation. As the temperature in summer is very high, the cooling demand cannot be reduced to thermal comfort level by low energy cooling techniques. So an active cooling system is required. Such system is preferable because it is not powered by electricity. Its production totally depends on fuel. The overall performance of an absorption refrigeration system is greatly affected by the characteristics of heat and mass transfer in the absorber where the refrigerant vapor is absorbed into the absorbent. The refrigeration capacity of an absorption refrigeration engine is limited by the absorption ability of the absorbent solution. In other words, for a given amount of absorbent solution in the machine, its duty is directly determined by the amount of refrigerant that it can absorb.

• Author: *Dipayan Mondal,* Department of Mechanical Engineering, Khulna University of Engineering & Technology (KUET), Khulna-9203, Bangladesh, dip.kuet@gmail.com

• *Co-* Author*: Md. Al-Imran ,Md. Alamgir Alam , Mohammad Ariful Islam,* Department of Mechanical Engineering, Khulna University of Engineering & Technology (KUET), Khulna-9203, Bangladesh

In water-lithium bromide vapor absorption refrigeration system, water is used as the refrigerant while lithium bromide (Li-Br) is used as the absorbent. Lithium bromide (Li-Br) water absorption units offer good thermodynamic performance and they are environmentally benign. Lithium bromide was chosen as absorbent because it meets the following desired solution characteristics for the absorption refrigeration systems: stability in aqueous solution and low vapor pressure at absorber conditions. The drawbacks of many working fluids, such as lithium bromide, are that they crystallize at high concentrations, are corrosive to metals, and are very expensive. One of the more serious limitations of lithium bromide solutions as absorbent solutions for refrigeration engines is the high temperature necessary in the generator to reach the boiling point. The conventional operation conditions in the generator vessel of these machines used to be 373.15 K at 10 kPa for a LiBr concentration of 60%.[1]. Lithium bromide-water chillers are available in two types, the single and the double effect. The single effect absorption chiller is mainly used for building cooling loads, where chilled water is required at 6-7^{0}C. The coefficient of performance (COP) varies to a small extent (0.65-0.75) with the heat source and the cooling water temperatures. Single effect chillers can operate with hot water temperature ranging from about 80^{0}C to 120^{0}C when water is pressurized.

The double effect absorption chiller has two stages of generation to separate the refrigerant from the absorbent. Thus the temperature of the heat source needed to drive the high-stage generator is essentially higher than that needed for the single-effect machine and is in the range of 150 to 200 ^{0}C. Double effect chillers have a higher COP of about 0.9-1.2 [2]. Although, double effect chillers are more efficient than the single-effect machines they are obviously more expensive to purchase. However, every individual application must be considered on its merits since the resulting savings in capital cost of the single-effect units can largely offset the extra capital cost of the double effect chiller.

On the other hand solar cooling has recently become one of the most discussed applications in absorption cooling. Solar cooling takes the majority of driving energy from solar radiation at no cost. Since the cooling demands of buildings usually behave according to the course of the sun, solar cooling uses this heat as the driving energy with which to generate comfortable room conditions. The amount of solar energy available in summer is not sufficient to run a big capacity absorption system. However a small capacity system can be run using solar thermal energy. Most of the absorption systems available in market are manufactured for big capacity industrial applications. It is essential to design a small capacity absorption system to run it using solar thermal energy.

2.0 RELATED WORKS:

The following experimental work was done:

Design and Construction of a Lithium Bromide Water Absorption Refrigerator [20],

3 WORKING & DESIGN CONSIDERATIONS

3.1 Working Procedure of a Single Effect Lithium Bromide (LiBr) - Water Cooling System

The single effect absorption technology provides a peak coefficient of performance (COP) of approximately 0.75 and operates with heat input temperatures in the range of 75°C to120°C. It should be noted that the refrigerant in the water-lithium bromide system is water and the LiBr acts as the absorbent, which absorbs the water vapor thus making pumping from the absorber to the generator easier and economic. A single-effect, two shell, LiBr -water chiller is illustrated in Fig 3.1.With reference to the numbering system shown in Fig 3.1, at point (1) the solution is rich in refrigerant and a pump forces the liquid through a heat exchanger to the generator (3). The temperature of the solution in the heat exchanger is increased.

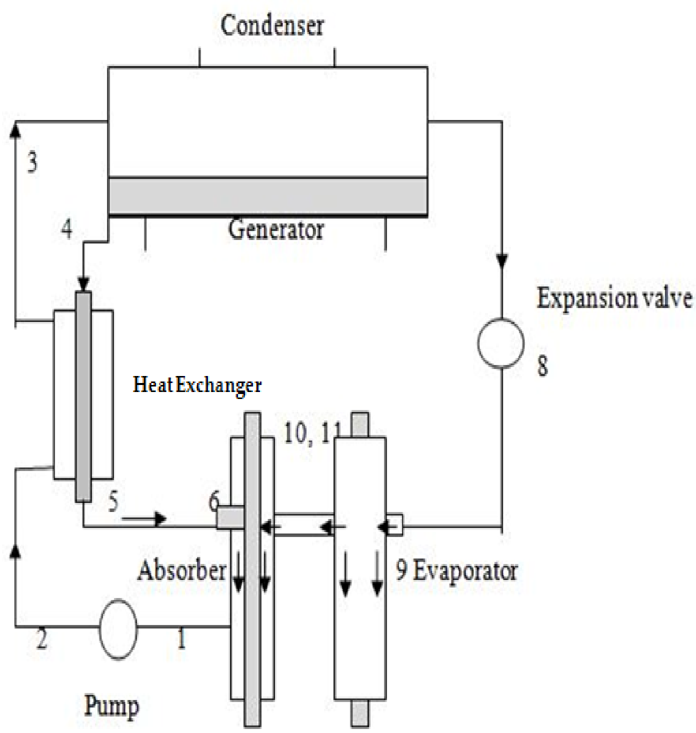

Fig.1: a) Single effect, LiBr-water absorption cooling cycle

ISSN 2229-5518

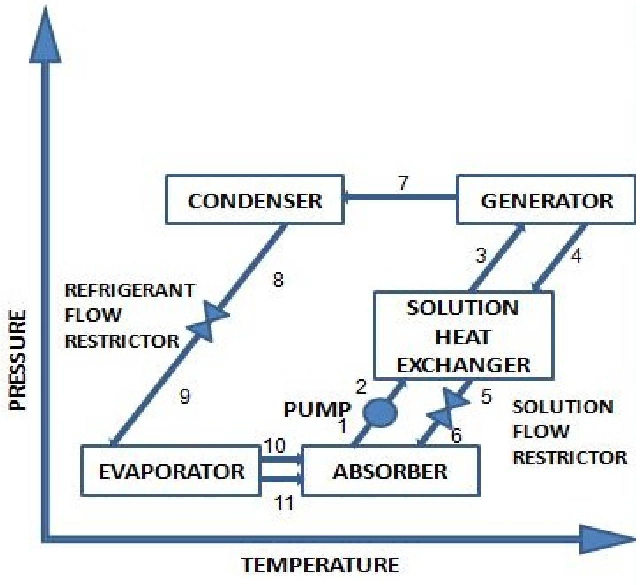

Fig.1: b) P-T diagram of LiBr-water absorption cooling cycle.

In the generator thermal energy is added and refrigerant boils off the solution. The refrigerant vapor (7) flows to the condenser, where heat is rejected as the refrigerant condenses. The condensed liquid (8) flows through a flow restrictor to the evaporator (9). In the evaporator, the heat from the load evaporates the refrigerant, which flows back to the absorber (10). A small portion of the refrigerant leaves the evaporator as liquid spillover (11) which is pumped back to the evaporator inlet again. At the generator exit (4), the steam consists of absorbent-refrigerant solution, which is cooled in the heat exchanger. From points (6) to (1), the solution absorbs refrigerant vapor from the evaporator and rejects heat through a heat exchanger. At point (1) the solution is reach in refrigerant and a pump (2) forces the liquid through a heat exchanger to the generator (3). The temperature of the solution in the heat exchanger is increased.

3.2 Design of a single effect LiBr-Water absorption cycle system

A single effect LiBr-water vapor absorption system is shown in Fig 3.1. Main components of the system are absorber, generator, condenser, evaporator and expansion valve.To perform designing of equipment size and performance evaluation of a single-effect LiBr-water absorption cooler basic assumptions are made. The basic assumptions are:

1. The steady state refrigerant is pure water.
2. There are no pressure changes except through the flow restrictors and the pump.
3. At points 1, 4, 8 and 11, there is only saturated liquid.
4. At point 10 there is only saturated vapor.
5. The pump is isentropic.
6. There are no jacket heat losses.
7. The capacity of the system is 2kW.

The design parameters are listed below:

Table 1: Design parameters for the single effect LiBr-water absorption cooler.

Parameter	Symbol	Value
Capacity	Q_e	2 kW
Evaporator temperature	T_{10}	10 ^{0}C
Generator solution exit temperature	T_4	90 ^{0}C
Weak solution mass fraction	X_1	55% LiBr
Strong solution mass fraction	X_4	60% LiBr
Solution heat exchanger exit temperature	T_3	65 ^{0}C
Generator vapor exit temperature	T_7	85 ^{0}C
Liquid carryover from evaporator	m_{11}	$0.025m_{10}$

3.3 Design of Heat Exchangers:

3.3.1 System heat exchangers sizing

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. In the heat transfer analysis, it is convenient to establish a mean temperature difference (ΔT_m) between the hot and cold fluids such that the total heat transfer rate Q between the fluids can be determined from the following expression:

$$Q = AU\Delta T_m \qquad (12)$$

Where, A(m^2) is the total heat transfer area and U (W/m^2-°C) is the average overall heat transfer coefficient, based on that area. For Eq. 12,

$$\Delta T_m = F\Delta T_{ln} = F\left(\frac{\Delta T_o - \Delta T_L}{\ln(\Delta T_o/\Delta T_L)}\right) \qquad (13)$$

F= Correction factor depending on the type of the heat exchanger used.

ISSN 2229-5518

The overall heat transfer coefficient (U) based on the outside surface of the tube is defined as [14]

$$U = \frac{1}{(D_o/D_i)(1/h_i)+(D_o/D_i)F_i+(1/2k)D_o\ln(D_o/D_i)+F_o+1/h_c} \quad (14)$$

For the design of the heat exchangers, the cooling water inlet and outlet temperatures are assumed. The cooling water inlet temperature depends exclusively on the available source of water, which may be a cooling tower or a well.

Table 2: Data for single effect LiBr-water cooling system

Point	h (kJ/kg)	m (kg/s)	P (kPa)	T (^{0}c)	X (%LiBr)
1	92.4	0.01287	1.227	34.9	55
2	92.4	0.01287	9.66	34.9	55
3	145.4	0.01287	9.66	65	55
4	212.2	0.0117975	9.66	90	60
5	154.38	0.0117975	9.66	59.93	60
6	154.38	0.0117975	1.227	44.5	60
7	2628	1.0725×10^{-3}	9.66	85	0
8	185.3	1.0725×10^{-3}	9.66	44.3	0
9	185.3	1.0725×10^{-3}	1.227	10	0
10	2519.24	0.000858	1.227	10	0
11	40.35	0.0002145	1.227	10	0

Table 3: Energy flows at various component of the system.

Description	Symbol	kW
Capacity	Q_e	2
Pump work	W	0.067
Absorber heat rejected	Q_a	2.8
Heat input to the generator	Q_g	3.45
Condenser heat rejected	Q_c	2.62

3.3.2 Absorber Design

For this design, the solution film can flow downward either on horizontal or on vertical tubes. The design of the horizontal tubes for the absorber, although theoretically well studied, presents a great problem with the shell tightness because of the large length of welds. In the case of this study the water vapor produced in the evaporator is absorbed in the flow of the LiBr-water solution and is condensing on the heat exchanger tubes. The design of the heat exchanger therefore requires values for heat and mass transfer coefficients.

Table 4: Absorber design parameters

Parameter	Value
Tube dimension; Inside diameter	Di =10.7 mm
Tube dimension; Outside diameter	Do = 12.7 mm
Cooling water inlet temperature	25°C
Cooling water outlet temperature	26°C
Mass flow rate of cooling water *(m)*	0.21 kg/s
Absorber load(Q_a)	2.8kW
Solution cooling	From 44.6°C to 34.9°C
Absorber pressure	1.227kPa
Inlet solution mass flow rate	0.0117975 kg/s

A practical model for absorption of vapors into a laminar film of water and LiBr falling down along a constant temperature horizontal plate was described by Andberg and Vliet (1983) [17]. The model developed considers non-isothermal absorption and the equations presented showed good agreement to experimental results. For this reason this method is chosen for the design of the absorber. The data are correlated with the introduction of the "absorption percentage (A_P)", defined as:

$$A_p = \frac{C_{in}-C_{out}}{C_{in}-C_{eq}} \times 100$$

(15)

Determination of the equilibrium concentration, C_{eq} requires the solution of the following set of expressions:

$A = -2.00755 + 0.16976\ X - 3.133362 \times .001\ X^2 + 1.97668 \times 0.00001\ X^3$

$B = 321.128 - 19.322\ X + 0.374382\ X^2 - 2.0637 \times .001\ X^3$

$C = 6.21147$

$D = -2886.373$

$E = -337269.46$

$T' = (-2\ E / [D + (D^2 - 4\ E\ (C - LOG(P / 6894.8))]^{.5}) - 459.72$

$T_W = (5 / 9)(AT' + B - 32)$

The above set of expressions requires an iterative type of solution to find C_{eq}, given T_W and P. In the case of this study T_W=31 °C and P=1227 Pa therefore C_{eq}= 0.52 and from Eq. (15) A_P = 62.5. A_P is correlated to the length of plate (L) by the expression:

$$L = am^b$$

(16)

Where, $a = -132\left(\frac{\ln(100-A_p)}{86}\right), b = 1.33$

An iterative solution gives, m=0.0292 kg/m-s corresponding to the area of 5.4 m length pipe.

The next step is to check the area of pipes needed to cool the solution to the required level.

Patnaik *et al.* (1993), suggest that Wilke's correlation [15], valid for constant heat flux wall with progressively decreasing difference from isothermal wall outside the entrance region, can be used for the falling film. It is assumed that the flow is fully developed in a wavy, laminar regime and that the bulk solution temperature profile is linear with respect to the transverse coordinate (Patnaik *et al.*, 1993). Wilke's correlation is:

$h_s = \frac{k_s}{\delta}\left(0.29(Re_s)^{0.53} Pr_s^{\ 0.344}\right)$
(17)
The film thickness δ is given by,
$\delta = \left(\frac{3\mu\Gamma}{\rho^2 g}\right)^{1/3}$
(18)
And the solution Reynolds number (Re_s) for the tube *is:*
$Re_s=4\Gamma/\mu$
(19)
In the case of this study the mean properties of the solution at 44.6°C and 57.5% LiBr are;
ρ= 1663 kg/m^3

μ= 4.20x10^{-3} N-s/m^2

k = 0.453 W/m-°C

c_p= 1991 J/kg-°C

Pr=18.46

Assuming 6m pipe and substituting the above values into Equations 17 - 19, a solution convective heat transfer coefficient h_s *of* 865 W/m^2-°C results.
The cooling water properties at the mean temperature of (25+26)/2=25.5°C are;
ρ= 997 kg/m^3

ν= 0.7876x10^{-6} m^2/s

k = 0.615 W/m-°C

Pr = 5.34

c_p= 4177.5 J/kg-°C

Therefore, substituting the above values into Eq. 20 and replacing $N_u = h_i\ D_i\ /\ K$, h_i=6175 W/m^2-°C.
By substituting the above values in Eq. 14 the resulting overall heat transfer coefficient (*U*) based on the outside surface of the tube is 650 W/m^2-°C.
In this case ΔT_{ln}= 9.3°C. Therefore from Eq. 16, the resulting length of pipe is 7.14m instead of 6m assumed. This means that the area of 6m length pipe is not enough to cool the solution to the required level. Checking for 7 pipes by repeating the above procedure, a length of 7 m results, with h_s= 840 W/m2-°C, h_i = 5450 W/m^2-°C and U= 625 W/m^2-°C, which indicates that 7 pipes are adequate to cool the solution.

Table 5: Obtained Heat exchanger size for absorber

Parameter	Value/Type
Tube diameter; Inside diameter	D_i =10.7 mm
Tube diameter; Outside diameter	D_o = 12.7 mm
Tube length	7 m
Tube material type	Copper

3.3.3 Evaporator Design

To facilitate construction, it was decided to construct the evaporator heat exchanger in a similar way as the absorber heat exchanger. A search in the literature has shown that this method is not studied and that the preferred method is to allow the liquid to enter inside a tube. The fluid inside the tube is heated by the run of fluid at the outer surface of the tube, so that progressive vaporization occurs. The heat transfer coefficient increases with distance from the entrance since heat is added continuously to the fluid. It is also not yet possible to predict all of the characteristics of this process quantitatively because of the great number of variables upon which the process depends and the complexity of the various two-phase flow patterns that occur as the quality of the vapor-liquid mixture increases during vaporization [18]. Therefore, in the case of this study the mean heat transfer coefficient is determined experimentally.

Table 6: Evaporator design parameters

Parameter	Value
Tube dimension; Inside diameter	*Di* =10.7 mm
Tube dimension; Outside diameter	*Do* = 12.7 mm
Heating water inlet temperature	25°C
Heating water outlet temperature	20°C
Mass flow rate of heating water *(m)*	0.0239 kg/s
Evaporator load(Q_e)	2kW
Evaporator pressure	1.227kPa
Vapor mass flow rate in evaporator	0.000858kg/s

3.3.4 Condenser Design

Table 7: Condenser design parameters

Parameter	Value
Tube dimension; Inside diameter	D_i=10.7 mm
Tube dimension; Outside diameter	D_o = 12.7 mm
Cooling water inlet temperature	25°C
Cooling water outlet temperature	28°C
Mass flow rate of cooling water*(m)*	0.21 kg/s
Condenser load (Q_c)	2.62kW
Condensed water temperature	From 85°C to 44.3°C
Condenser pressure	9.66kPa
Condensed water mass flow rate	0.00107 kg/s

The overall heat transfer coefficient is given by Eq. 14. For this equation, the value of the fouling factors (F_i,F_o) at the inside and outside surfaces of the tube can be taken as 0.00009m^2°C/W [14] and *k* for copper = 383.2 (W/m-°C). The heat transfer coefficients, h_i ,h_o, for the inside and outside flow need to be calculated.

The Petukhov-Popov equation [14] for turbulent flow inside a smooth tube gives:

$$N_u = \frac{Re.Pr}{X}\frac{f}{8} \quad (20)$$

Where, $X = 1.07 + 12.7(Pr^{2/3} - 1)(\frac{f}{8})^{1/2}$

Eq. 20 applies for Reynolds numbers $10^4<Re< 5x10^6$ and Prandtl numbers, $0.5<Pr<2000$. The Petukhov-Popov equation agrees with the experimental results for the specified range within ± 5%.The water properties at the mean temperature of (25+28)/2=26.5°C are;

ρ = 997kg/m^3

ν = 0.8365×10^{-6}m^2/s

k = 0.610 W/m-°C

Pr = 5.85

C_p = 4180 J/kg-°C

μ= 0.86×10^{-3} kg/m.s

Q_c equals to 2.62 kW, therefore

$$m = \frac{Q}{C_p\Delta T} = \frac{2620}{4180 \times 3} = 0.21\ kg/s$$

$$Re = \frac{4m}{\pi D\mu} = \frac{4 \times 0.21}{\pi \times 0.0107 \times 0.86 \times 10^{-3}} = 29159$$

Substituting the above values into Eq. 20 and replacing $Nu = h_iD_i / K$, gives h_i = 11080 W/m^2°C.

Nusselt's analysis of heat transfer for condensation on the outside surface of a horizontal tube, gives the average heat transfer coefficient as [14]

$$h_m = 0.725\left[\frac{g\rho_1(\rho_1-\rho_v)h_{fg}k_l^3}{\mu_1(T_v-T_w)D_o}\right]^{0.25} \quad (21)$$

The physical properties in Eq. (21) should be evaluated at the mean wall surface and vapor saturation temperature. The average temperature of the condensate film is (44.3+26.5)/2=35.4°C and its physical properties are;

ρ_1= 994.04 kg/m^3

ρ_v= 0.0396 kg/m^3

h_{fg}= 2419×10^{3}J/kg

k_l= 0.613 W/m-°C

μ_l= 801.4×10^{-6}kg/m-s

T_w = 26.5°C

T_v= 44.3°C

D_o = 0.0127 m

h_m= 9526 W/m^2-°C

By substituting the above values in Eq. 14 a resulting overall heat transfer coefficient of U=2429 W/m^2-°C is determined.

The logarithmic mean temperature difference is [14]

$$\Delta T_{ln} = \frac{(T_{sat} - T_i) - (T_{sat} - T_0)}{\ln[(T_{sat} - T_i)/(T_{sat} - T_0)]}$$

Finally, $\Delta T_{ln} = 19.46$

The tube length L is determined by writing an overall energy balance, [14]

$$Q_{tube} = (\pi D_0 L)U_0\Delta T_{ln} = mC_p(T_0 - T_i)$$

Which gives, $L=1.4\ m$

That means, 1.4m long tube of outside diameter D_o = 12.7 mm and inside diameter D_i =10.7 mm is required.

Table 8: Obtained Heat exchanger size for condenser

Parameter	Value/Type
Tube diameter	inside diameter , D_i =10.7 mm
	outside diameter , D_o = 12.7 mm
Tube length	1.4 m
Tube material type	Copper

3.3.5 Generator Design

The generator provides sensible heat and latent heat of vaporization. The sensible heat raises the inlet stream temperature up to the saturation temperature. This amount of heat, typical in practice, is 13% of the total heat required [15]. The heat of vaporization consists of the heat of vaporization of pure water and the latent heat of mixing of the liquid solution. Typically, the heat of mixing is about 11% of the heat of vaporization for water/ lithium bromide.

Table 9: Generator design parameters

Parameter	Value
Tube dimension	Inside diameter D_i =10.7 mm
	Outside diameter D_o = 12.7 mm
Generator pressure	9.66kPa
Generator solution	Entering: 55% LiBr at 65°C, and Leaving: 60% LiBr at 90°C
Water vapor mass flow rate	m =0.00107 kg/s
Generator load	Q_g =3.45kW

ISSN 2229-5518

The above analysis indicates that the heat to be provided by the generator can be based on the heat of vaporization of pure water, increased by about 23% in a typical design.

Although considerable research work has been done in the past on the pool boiling of liquids, the data on water/ lithium bromide solutions are not extensive [16]. Experimental results indicate that the boiling phenomenon is not significantly affected by tube diameter but is greatly affected by the solution concentration. As the solution concentration increases the heat transfer coefficient decreases. Also the heat transfer coefficient increases as the heat flux increases. Average heat transfer coefficients were found to vary between 1600 W/m²-°C and 7500 W/m²-°C [16]. The tube material affects the heat transfer coefficient as shown by an empirical relation developed by Rohsenow (in Kreith and Bohn, 1997) for nucleate boiling. Since no formula is available for calculating the exact heat transfer coefficient, the generator heat exchanger tube length is varied and fixed to a length for desired load.

4 CONSTRUCTIONS

4.1 Construction of the Vapor absorption system

All heat exchangers described in chapter 4, are constructed in a way that permits the use of varying number of tubes. After designing the basic heat exchangers each component are constructed. Copper tubes and galvanized iron plates are used for constructing these components. To carry out some performance test, thermocouples, vacuum pressure gauge, gate valves (for varying flow rate) are installed at various points of the unit for measurements and adjustments.

4.1.1 Absorber

The absorber is constructed from a number of horizontal tubes (total 30.48cm length) connected to each other to form a single pipe with 12.7mm outside diameter.

On the top part of tube, another tube 6.35mm inside diameter and 30.48cm in length is fitted. This tube has sophisticated holes in it in such a way that the LiBr-water solution can pass inside the tube and falls drop wise on the top of every horizontal tubes one by one. All the tubes are made of copper.

Fig. 3: Falling solution on absorber horizontal tubes

The film of the absorber solution can thus be cooled by water flowing inside the horizontal tubes. The vapor coming from the evaporator is absorbed in the absorber by the absorbent in solution. The whole system is placed in a vacuum box made of galvanized iron plates. The temperature of cooling water at heat exchanger inlet-outlet can be obtained from thermocouple arrangements. The absorber should be maintained at 9.66 kPa. So a vacuum pressure gauge is installed at the top of the absorber to maintain the pressure as desired.

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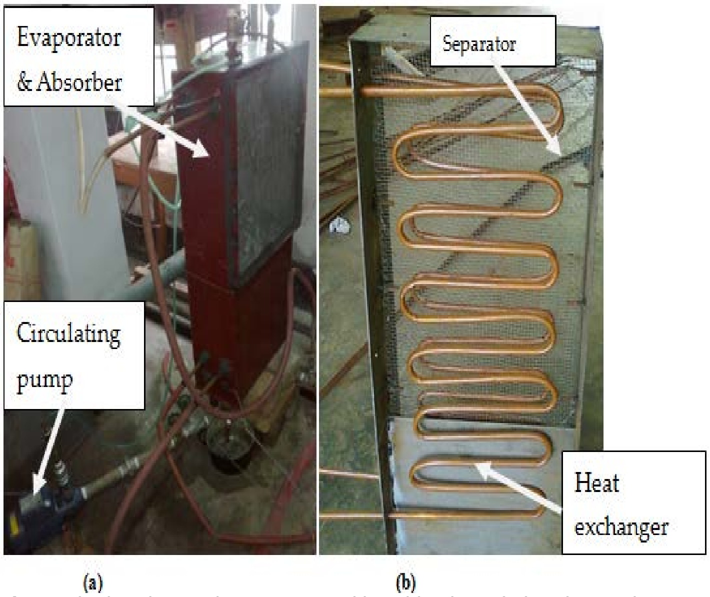

Fig. 4: a) Absorber and evaporator; b) Inside view of absorber and evaporator

4.1.2 Evaporator

The evaporator heat exchanger is constructed in the same way as the absorber heat exchanger. The shell is cube-shaped, made from 3mm thick galvanized iron plates.
Inside the shell, horizontal tubes similar to the absorber tubes are fitted. Supplied water runs through the horizontal tubes which supplies the sufficient heat needed to evaporate liquid refrigerant in the evaporator. In the case of falling films evaporating on the outer side of tubes it is difficult to predict the heat transfer coefficient. The advantage of high coefficient in falling-film exchangers is partially offset by the difficulties involved in distribution of the film and maintaining complete wet ability of the tube. For this reason the heat transfer coefficient of the specific construction is found experimentally.

4.1.3 Generator

In a vapor absorption system a generator is used for mainly two purposes. It evaporates the refrigerant and delivers to the condenser. It also supplies LiBr-Water solution to the absorber.
The main objective is only to compute COP of the system and minimum temperature that can be obtained in the evaporator and some performance characteristics. So to reduce the complexity of the system a vacuum box (made of galvanized iron plates) is used as an alternative of generator which has capacity of storing LiBr-Water solution under desired vacuum pressure and temperature. LiBr-Water solution should be fed manually into this to run the whole system. A thermocouple arrangement and vacuum pressure gauge is installed for taking any adjustments and measurements.

4.1.4 Condenser

For the purpose of running a vapor absorption system a condenser is a must. Condenser condenses vapor obtained from generator and transfers the liquid refrigerant (water) to evaporator through expansion valve. The main objective is only to compute COP of the system and minimum temperature that can be obtained in the evaporator and some performance characteristics.

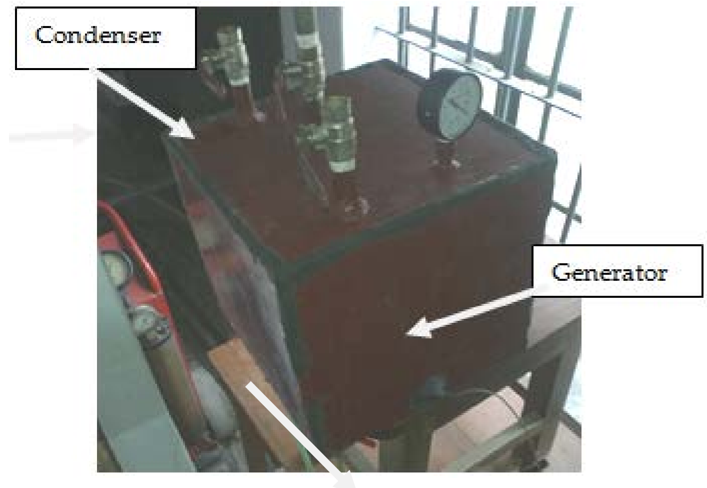

Fig. 5: Condenser and generator

So to reduce the complexity of the system a vacuum box (made of galvanized iron plates) is used as an alternative of condenser which has capacity of storing liquid water under desired vacuum pressure and temperature. Water should be fed manually into this to run the whole system. A thermocouple arrangement and vacuum pressure gauge is installed for taking any adjustments and measurements.

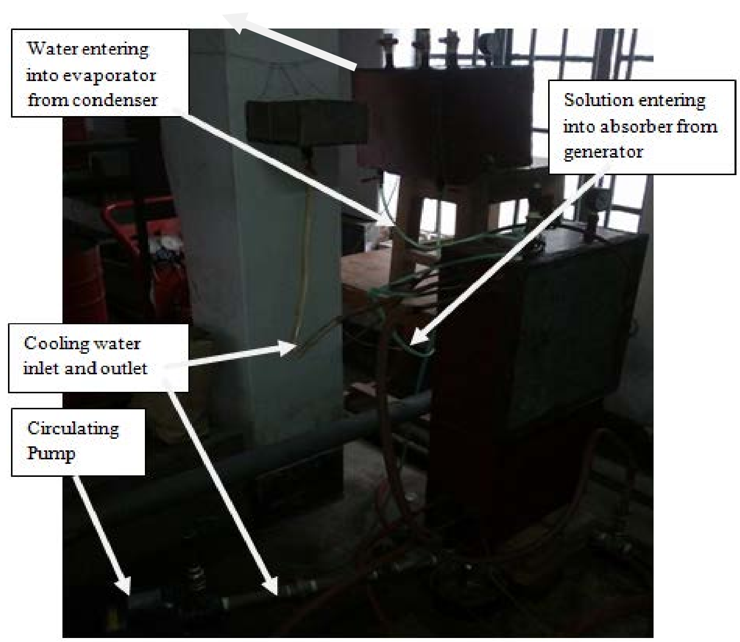

Fig. 6: Details of construction of the vapor absorption system

5 PERFORMANCE TEST

Table 10: Data collection for vapor absorption cooling system (for 50% LiBr Generator solution)

Observation no.	Generator			Condenser			Evaporator					Absorber				
	Pressure (-cmHg)	Temperature (ºC)	Mass flow (kg/s)	Pressure (-cmHg)	Temperature (ºC)	Mass flow (kg/s)	Inlet T (Tube) (ºC)	Outlet T (Tube)	Mass flow in Tube (kg/s)	Pressure (-cmHg)	Temperature (ºC)	Inlet T (Tube) (ºC)	Outlet T (Tube) (ºC)	Mass flow in (Tube) (kg/s)	Pressure (-cmHg)	Temperature (ºC)
1	65	59	0.009	65	38	0.008	45	31	0.02	69	27	31	32	0.015	69	41
2	60	56	0.009	60	38	0.008	51	39	0.032	65	29	31	32	0.019	65	44
3	55	58	0.011	55	37	0.012	62	42	0.021	60	30	31	33	0.017	60	44
4	50	55	0.008	50	37	0.009	66	47	0.018	55	31	31	34	0.02	55	41
5	68	59	0.012	68	44	0.015	30	23	0.019	72	22	31	32	0.018	72	38

6 RESULTS AND DISCUSSIONS

6.1 COP analysis

Here, COP= $\frac{Q_e}{Q_g}$

$Q_e = mC_p\Delta T = 1.1\ kW$

Where, ΔT is the heating water inlet-outlet temperature difference in the evaporator.

$Q_g = 3.45$ kW

So COP= 0.32 which is less than the COP obtained in design before. This is due to lower heat transfer through heat exchangers. The actual overall heat transfer coefficient of copper tubes is less than the ideal heat transfer coefficient. This is because of rust on the upper surface of copper tubes. Another reason is absence of insulation arrangements.

6.2 Absorber and generator heat transfers

For the analysis of the absorber and generator heat transfers, a condenser temperature of 38ºC is used. For the LiBr-H2O system, Fig.7 shows absorber heat reject against absorber temperature. It shows that a high absorber temperature not only reduces the COP but also increases the cooling requirements of the system. Therefore reducing absorber temperature has a twofold advantage. The absorber heat rejection is not too sensitive to variations in generator temperature.

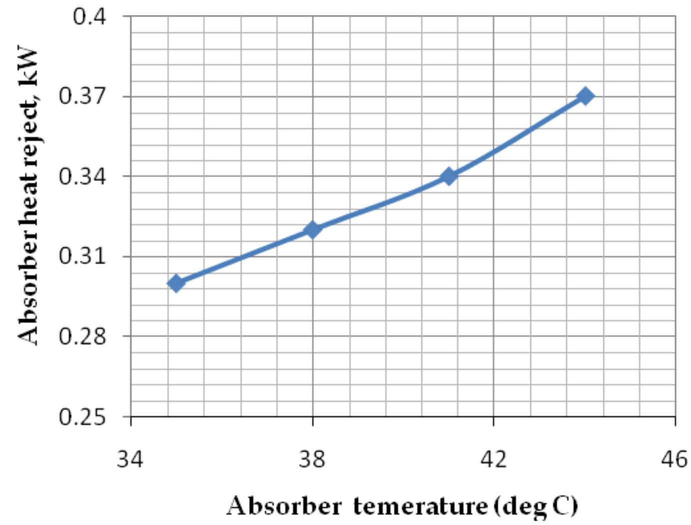

Fig .7: Absorber heat reject against absorber temperature

6.3 Absorption rate variations

The values of the absorption rate obtained for a solution of LiBr + water at 50 mass % after 1 to 5 min duration of loading are presented in Fig.8. The absorption rate remains constant for any time longer than 3 min.

ISSN 2229-5518

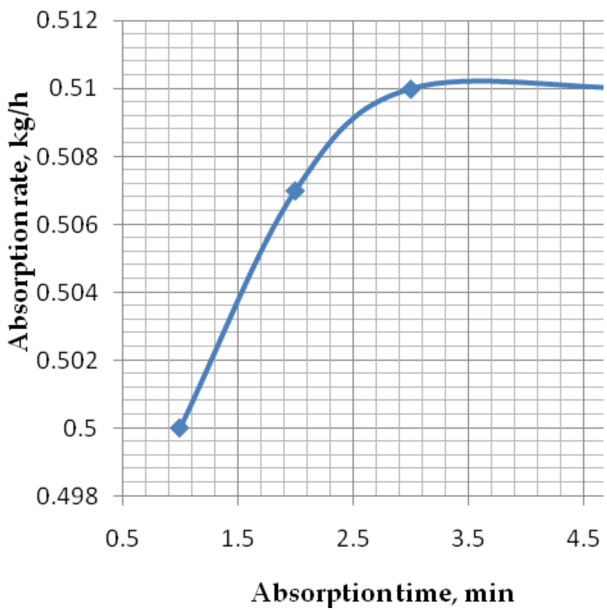

Fig. 8: Absorption of water vapor into 50 mass % LiBr + water solutions at different absorption times.

The water vapor absorbed increases exponentially with absorbent flow rate, so the maximum absorption rate is reached near a value next to the solubility limit of the salt. The solubility limits must accommodate the working conditions to avoid the crystallization of the salt in the absorber.

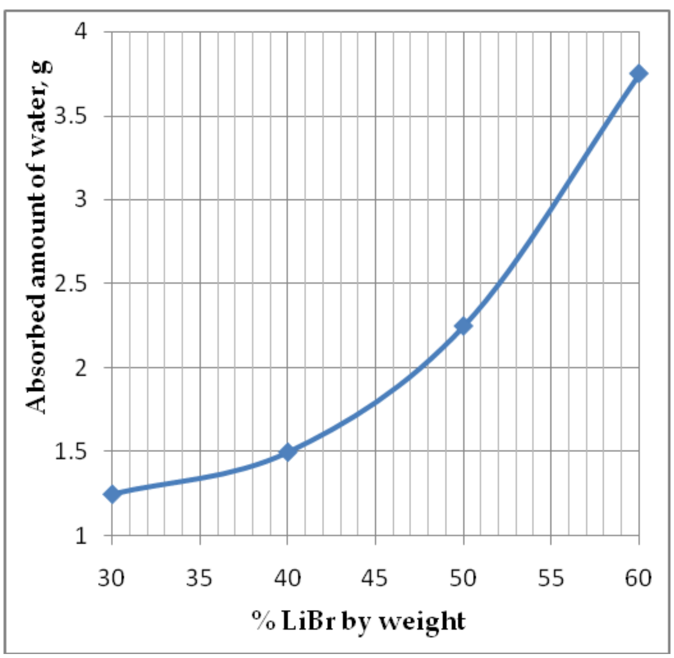

Fig.10: Influence of absorbent concentration in absorption rate.

6.4 Evaporator heat exchanger exit temperature

Outlet temperature of water through heat exchanger in evaporator varies inversely with the maintained vacuum pressure. It is possible to maintain the evaporator and absorber under a maximum of -72 cmHg pressure. The actual lower temperature that can be obtained in evaporator is 23°C.

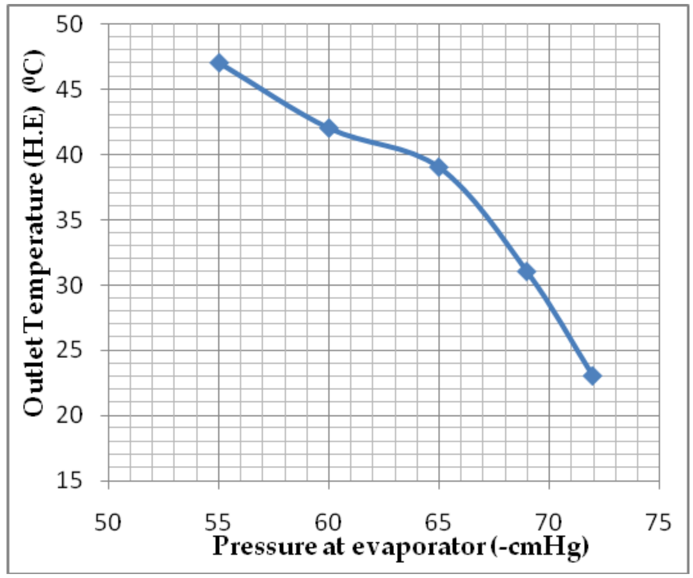

Fig. 9: Influence of pressure at evaporator in evaporator heat exchanger exit temperature

6.5 Experimental results

To investigate the accuracy of theoretical procedure described in section (4) for the design of various heat exchangers, a 2 kW model was designed and constructed. A summary of the experimental results is presented in table 11

Table 11: Inlet outlet temperature of various components

Heat Exchangers	Cooling/Heating water temperature (°C)	
	Entering	Leaving
Generator	-	59
Condenser	-	44
Evaporator	30	23
Absorber	31	32
And the experimentally obtained COP is 0.32		

A small capacity Vapor Absorption system is designed and constructed to compare actual performance with designed conditions. The experimentally obtained COP is 0.32 which is less than the design COP 0.58. This is due to lower heat transfer through heat exchangers. The actual overall heat transfer coefficient of copper tubes is less than the ideal heat transfer coefficient. This is because of rust on the upper surface of copper tubes. Another reason is absence of insulation arrangements.

Fig.7 shows absorber heat reject against absorber temperature. It shows that a high absorber temperature not only reduces the COP but also increases the cooling requirements of the system. Therefore reducing absorber temperature has an advantage. The absorber heat rejection is not too sensitive to variations in generator temperature.
The system was run for about 4 to 5 minutes. It was observed that absorption rates increases a little for first 1 to 3 minutes and after this absorption rate becomes constant. Absorption rate also varies proportionally with weight percentage of LiBr. Absorption rate is high for higher weight percentage of LiBr.

But with increasing weight percentage of LiBr, the possibility of crystallization of LiBr increases. Usually crystallization of LiBr occurs around or below 10°C. It is essential to have an accurate prediction of the crystallization temperature in this range in order to avoid crystallization during the design phase. The minimum temperature that can be obtained in this system is 23°C. So there is no possibility of crystallization of LiBr in this system. But it is strongly recommended to have an accurate prediction of the crystallization temperature to avoid crystallization if the load is high enough or temperature is below 10°C.

It is so important to maintain each component at required pressure. Every component should be leak proof. Otherwise, COP will decrease and system will be interrupted. To avoid leakage a vacuum pump can be run frequently to maintain pressure.
Since the vapor absorption system has lower COP than vapor compression system, vapor absorption system can be used economically to cool an insulated room or house. It can be run by renewable energy, waste energy or solar thermal energy.

7 CONCLUSIONS

A small capacity vapor absorption system is designed based on some correlations and formulae. Then heat exchangers used for each component are designed. Heat exchanger tube material and dimensions are determined. The unit designed is constructed and each heat exchanger is adjusted to the required output. In this way the designed COP is ensured. Some performance test is made to evaluate performance of the system. The present cost of the absorption unit together with its running cost is economically viable. Considering also the destruction of the ozone layer caused by the use of electric chillers, absorption units will offer a better environment, especially if some form of renewable or waste energy is used for their operation.

ACKNOWLEDGMENT

The authors wish to thank Prof. Dr. Mohammad Ariful Islam, Department of Mechanical Engineering, and Khulna University of Engineering & Technology (KUET) and also would like to gratefully KUET, Bangladesh for the financial support.

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Nomenclature

Q_e = Capacity
T_{10}= Evaporator temperature
T_4 = Generator solution exit temperature
X_1 = Weak solution mass fraction
X_4 = Strong solution mass fraction
T_3 = Solution heat exchanger exit temperature
T_7= Generator vapor exit temperature
m_{11} = Liquid carryover from evaporator
P = Saturation pressure
H = Enthalpy
W = Pump work
Q_a = Absorber heat rejected
Q_g = Heat input to the generator
Q_c = Condenser heat rejected
Q = Heat transfer rate
ΔT_m =Temperature difference
A= Total heat transfer surface area (m^2)
K= Thermal conductivity (W/m .^{0}C)
U= Overall heat transfer coefficient
h= Convective heat transfer coefficient (W/m^2.^{0}C)
ΔT_{lm}= Logarithmic mean temperature difference (^{0}C)
C_p = Specific heat of the fluid (J/kg. ^{0}C)
L= Length of heat exchanger (m)
Re = Reynolds number
Pr = Prandtl number
Nu = Nusselt number

Greek:

Γ= mass flow rate per wetted perimeter (kg/m-s)
δ = film thickness (m)
μ = absolute viscosity (N-s/m2) = ν ϱ
ν = kinematic viscosity (m2/s)
ρ = density (kg/m3)
ϱl = liquid density (kg/m3)
ϱv = vapour density (kg/m3)
μl = absolute viscosity of liquid (N-s/m2)

Subscripts

h= hot fluid
c=cold fluid
i= inside
o=outside