

# Fabrication and Experimental Analysis of Absorber Based LiBr – Water Absorption Refrigeration System

CH Jaya Prakash, K Dilip Kumar, T Srinivasa Rao



**Abstract:** The goal of this research work is to design, construct and analyse the performance of a single stage vapour absorption unit of 1kW rated cooling capacity. The correlations required for the design and analysis are mentioned in the section III. The solution heat exchanger of spiral tube and shell type was designed and installed in the unit. A forced horizontal tube heat exchanger type condenser was designed and used in the construction of the unit. A pool boiling generator has been installed in the unit. The determined hypothetical results including the overall heat transfer coefficient are compared with the test results inferred for a developed unit with a nominal output of 1kW. At last the performance parameters are examined in connection to the concentration of the absorber and mass flow rates. The actual COP of the system increased by 15% with the decrease in solution concentration at absorber inlet. Critical analysis is carried-out at each level to calculate the each accessory heat load and other parameters.

**Keywords:** Vapour absorption system; Lithium Bromide (LiBr); Water (H<sub>2</sub>O); Coefficient of performance (COP); Solution Heat Exchanger (SHE).

## I. INTRODUCTION

VAR systems are driven by thermal energy and no need of high grade energy as like Vapour compression systems. VAR systems provides greater cooling where there is a scarcity of high grade energy, reliable cooling with renewable energy sources and even with waste heat. Various refrigerant and absorbent compounds are available. The most widely recognized refrigerant and absorbent blends are H<sub>2</sub>O-NH<sub>3</sub> and LiBr-H<sub>2</sub>O. Both differ basically in the methodology used for compression. In VAR system the compressor is replaced by three components namely generator (desorber), absorber and pump. The power used for the pump is negligible as compared with the compressor used in VCR. Another major difference between the Vapour compression refrigeration and vapour absorption refrigeration is the refrigerant used. Generally vapour compression refrigeration systems uses compounds of chlorofluorocarbon refrigerants (CFCs), because of their inert nature at low temperatures and they do undergo significant reaction in the stratosphere.

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The depletion of stratospheric ozone was getting worse year by year. In this scenario the absorption systems are getting importance to reduce the usage of CFCs.

In the middle of 20<sup>th</sup> century Aqueous LiBr solution absorption systems are commercialized for chillers and other household purposes.

The NH<sub>3</sub>-H<sub>2</sub>O blend is not suitable to operate with solar system because there is a need of high temperature (120°C to 170°C) in the generator. The required temperatures inside the generator in LiBr-H<sub>2</sub>O pair are 75°C to 120°C. So LiBr-H<sub>2</sub>O blend is suitable to operate with Solar energy and other renewable energy sources.

The absorber analysis has been done by varying the inlet concentration ratio. The determined hypothetical results are compared with the test results inferred for a developed unit with a nominal output of 1kW. At last the performance parameters are examined in connection to the concentration of the absorber and mass flow rates.

## II. LITHIUM BROMIDE – WATER COOLING

In LiBr-H<sub>2</sub>O system water acts as a refrigerant and the Lithium Bromide acts as an absorbent. The pumping of aqueous LiBr solution from absorber to evaporator is easy and economic. The scheme of single stage lithium bromide – water (LiBr-H<sub>2</sub>O) system is illustrated in Fig. 1. Based on the mentioned number series in the scheme shown in Fig. 1, at point (1) i.e., at absorber the strong solution which is rich in refrigerant is forced by the pump (2) to the generator (3) through Heat exchanger. The low grade heat energy or heat energy extracted from the renewable energy is added to the strong solution which is rich in refrigerant in the generator and the solution boils off. The refrigerant vapours (6) flows to the condenser which is under forced convection, where the vapour converts its phase from vapour to liquid by heat rejection and later flows to the evaporator (8) through a flow restrictor called expansion valve (capillary). In the evaporator the refrigerant evaporates by extracting the rejected heat from refrigerated space and later flows back to the absorber (9). The temperature of the strong solution is increased in the solution heat exchanger (5) by extraction of heat energy from the weak solution which is returned from generator after the vaporisation of refrigerant (4). The 2-way pressure gauge is installed to avoid the high pressure fluctuations in high pressure side if any.

## III. DESIGN OF SINGLE EFFECT VAPOUR ABSORPTION SYSTEM

To design and analyse the performance of any system, we need to consider some fundamental assumptions and input esteems.



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The considered fundamental assumptions and input esteems for this system with reference to the illustrated Fig. 1 and 2.

- The steady state refrigerant is pure water.

- No change in pressure of the unit, except through the pump and the expansion valve.
- Saturated liquid at points 1, 4 and 7.
- Saturated vapour at point 9
- The expansion is adiabatic.
- The pumping is isentropic.
- There are no further heat losses.

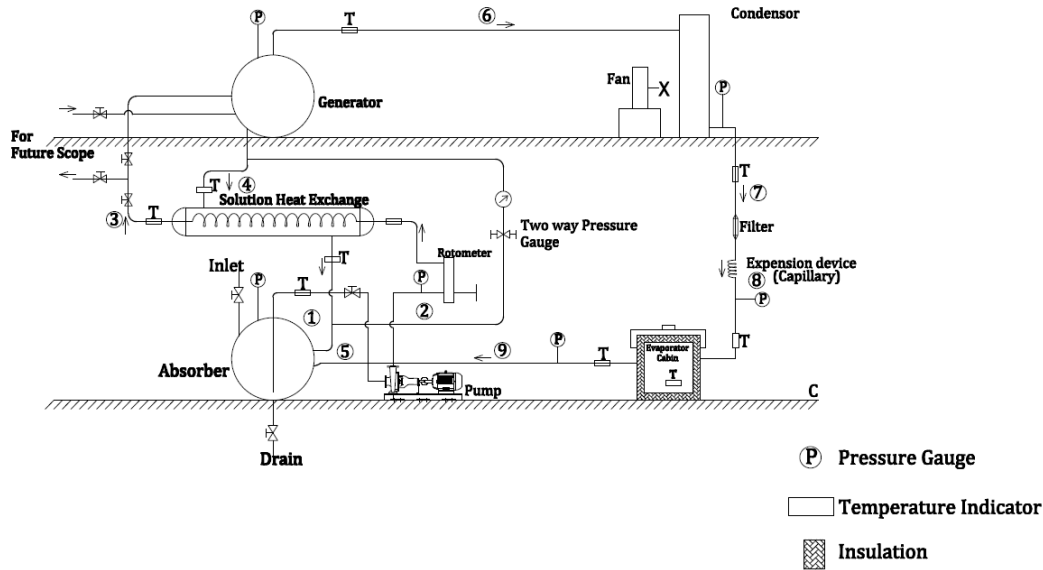


Fig. 1. Schematic representation of Designed Single Stage LiBr – H<sub>2</sub>O System

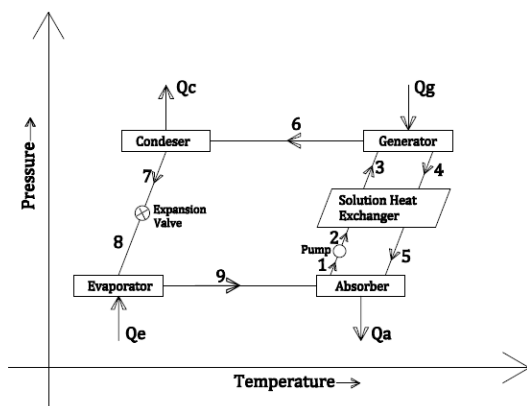


Fig. 2. Modified LiBr-H<sub>2</sub>O cooling cycle [4]

The goal is to design and construct a small pilot single stage Lithium Bromide – Water (LiBr-H<sub>2</sub>O) absorption chiller of 1kW of cooling power. Various properties of the lithium bromide – water solution and water are estimated by using the mathematical curve fitting equations. The system is designed to maintain generator operating temperature of 75 °C, which can be easily achievable and has a good COP of 0.75. Lower generator temperatures cannot be used because the lower generator temperature results the decrease in temperature values at condenser outlet which is not a preferable condition to achieve a good COP.

### A. Sizing of system Heat Exchanger:

The temperature difference ( $\Delta T$ ) is not constant throughout the heat exchanger it varies with the length of heat exchanger. The rate of heat transfer can be determined by utilizing the below mentioned expression.

$$\dot{Q} = UA\Delta T_m \quad (1)$$

Mean temperature difference ( $\Delta T_m$ ) between the hot and cold stream is obtained by using

$$\Delta T_m = F \Delta T_{ln} = F \left[ \frac{\Delta T_O - \Delta T_L}{\ln(\Delta T_O / \Delta T_L)} \right] \quad (2)$$

The overall heat transfer coefficient (U) based on the outside surface of the tube will be obtained by using [5]

$$U = \frac{1}{(D_o/D_i)(1/h_i) + (D_o/D_i)F_i + [1/(2k)]D_o \ln(D_o/D_i) + F_o + (1/h_o)} \quad (3)$$

Before designing the single stage VAR there is a need of mass flow rate at each point mentioned in the scheme, the methodology which is followed in mass flow rate calculations are deliberated below

Firstly, Volume of 3Litres refrigerant and absorbent (Lithium bromide – Water) blend density is calculated by using the methodology mentioned,

### B. Density of absorbent – refrigerant blend:

The density of mixture is calculated by

$$\rho_{mixture} = \frac{(\rho_1 V_1 + \rho_2 V_2 + \dots)}{(V_1 + V_2 + \dots)} \quad (4)$$

By substituting the Table I parameters in Equation 4 results, density of LiBr – H<sub>2</sub>O compound is

$$\rho_{LiBr - H_2O} = 1341 \text{ kg/m}^3$$

**Table- I: LiBr-H<sub>2</sub>O Blend Characteristics**

Parameter	Type/Value
Total Volume of Lithium bromide – Water blend	3Litres
Volume of absorbent (LiBr) - 55% of total Volume (V <sub>1</sub> )	1.65 Litres
Inlet Concentration (mass fraction) of solution (tested)	55.76%
Volume of Refrigerant (H <sub>2</sub> O) - 45% of total volume (V <sub>2</sub> )	1.35 Litres
Density of absorbent (LiBr) at 25 (ρ <sub>1</sub> )	1620 kg/m <sup>3</sup>
Density of Refrigerant (H <sub>2</sub> O) at 25 (ρ <sub>2</sub> )	999.63 kg/m <sup>3</sup>

Mass fraction of strong solution (ξ<sub>ss</sub>) = 0.6 at point 3 and Mass fraction of weak solution (ξ<sub>ws</sub>) = 0.5576 at point 1

For circulation ratio (λ) based on the mass fraction and strong solution and mass fraction of weak solution

$$\lambda = \frac{\xi_{ws}}{\xi_{ss} - \xi_{ws}} \quad (5)$$

For mass flow rate of refrigerant (m) based on the circulation ratio and mass flow rate of the strong solution

$$\text{Mass flow rate of strong solution}(m_{ss}) = \lambda m \quad (6)$$

For mass flow rate of the weak solution based on the mass flow rate of the refrigerant (m) and circulation ratio (λ)

$$\text{mass flow rate of the weak solution} \\ (m_{ws}) = (1 + \lambda)m \quad (7)$$

By substituting the mentioned values in the Equations 5-7

- Circulation ratio (λ) = 13.15
- Volume flow rate of strong solution for 1kW cooling power at point 1 = 0.23 LPM
- m<sub>ss</sub> = 0.005287021 kg/sec
- m = 0.000402026 kg/s for 1kW cooling load
- m<sub>ws</sub> = 0.005689047 kg/s

**C. Evaporator Heat Exchanger Design:**

The Refrigerant is in saturated vapour condition in the evaporator and the temperature (T<sub>o</sub>) is assumed to be 6°C for the heat exchanger design. The volume of refrigerating space is about 6 Litres. The designed cooling capacity of evaporator is of 1kW.

The refrigerant inside the tubes run with high velocity, low vacuum pressure and low temperature (6°C). The refrigerant being heated by extracting heat from the

refrigerating space and the progressive vaporisation of refrigerant occurs. The water which was filled in the evaporated cabin is in standstill condition then considered the free convection for the calculation of outside heat transfer coefficient (h<sub>o</sub>) and the refrigerant flowing inside the tubing then it was considered as a forced convection to calculate the inside heat transfer coefficient (h<sub>i</sub>).

Though it seems to be easy but it is not possible to predict all the parameters in this evaporation process quantitatively. So in this study the mean heat transfer coefficient is predicted experimentally. For outside heat transfer coefficient (h<sub>o</sub>), in a laminar flow for all values of GrPr and for the constant heat flux [6].

$$Nu = \left[ 0.825 + \frac{0.387(GrPr)^{0.167}}{\left\{ 1 + \left[ \frac{0.437}{Pr} \right]^{0.5625} \right\}^{0.292}} \right]^2 \quad (8)$$

Equating the nusselt number Nu=h<sub>o</sub>L/k, from this h<sub>o</sub> value is being calculated. For inside heat transfer coefficient in a laminar flow for constant heat flux, internal flow and assumed the flow is a fully developed one [6].

$$Nu = 4.36 \quad (9)$$

Equating the nusselt number Nu=h<sub>i</sub>L/k, from this h<sub>i</sub> value is being calculated. By substituting the values obtained through calculations in Equation 3 a resulting overall heat transfer coefficient (U).

Assuming the number of tubes = 4 and the characteristics of the evaporator heat exchanger is mentioned in the Table II.

Table- II: Characteristics of evaporator heat exchanger

Input Data		Unit	Evaporating heat transfer coefficient (hi)		Unit
Outside dia of tube (Do)	6.8	mm	Area	3.1669E-05	m <sup>2</sup>
Inside dia of tube (Di)	6.35	mm	Cp	4178	J/kg <sup>o</sup> K
k of Copper	383.2	W/m <sup>o</sup> K	Abs Viscosity	1.55E-03	Ns/m <sup>2</sup>
Cooling temp	6	<sup>o</sup> C	Density	1000	kg/m <sup>3</sup>
Atm temp	27	<sup>o</sup> C	K	0.5694	W/m <sup>o</sup> K
C <sub>p</sub> of water	4.178	kJ/kg <sup>o</sup>	Re	52.006550	
Density	1000	kg/m	Pr	11.3731986	
Enthalpy of water at 6 <sup>o</sup> C	25.2	kJ/kg	Nu	4.36	
Enthalpy of vapour at 6 <sup>o</sup> C	2512.6	kJ/kg	hi	<b>390.9581102</b>	W/m <sup>2o</sup> K
Mass flow rate of refrigerant	0.000402026	kg/sec			
<b>Evaporator Design Load</b>	<b>1</b>	kW			
Volume of water	0.006	m <sup>3</sup>			
Water mass flow rate	0.02	kg/sec	<b>Overall HT Coefficient (U)</b>		
Temp drop of water	11.96744854	<sup>o</sup> C			
Final temp of water	15.03255146	<sup>o</sup> C	(Do/Di)(1/hi)	0.002739082	
Del T1	21	<sup>o</sup> C	(Do/Di)Fi	0.096377953	
Del T2	9.03255146	<sup>o</sup> C	(1/2k)Doln(Do/Di)	2.93012E-06	
<b>LMTD</b>	<b>10.733</b>		Fo	0.09	
<b>Outside Heat Transfer Coefficient (ho)</b>			1/ho	0.139945322	
Dia	0.175	m		0.3290652	
Length of Cylinder	0.24945837	m	<b>U</b>	<b>3.03891642</b>	kW/m <sup>2o</sup> K
Temp difference of Water	11.96744854	<sup>o</sup> C	A	0.030659209	m <sup>2</sup>
Density	1000.35	kg/m <sup>3</sup>			
Abs Viscosity	1.14E-03	Ns/m <sup>2</sup>	L	1.43516704	m
Cp	4184.65	J/kg <sup>o</sup> K	N	4	Nos
k of water	0.5898	W/m <sup>o</sup> K	<b>Total Length</b>	<b>5.740650816</b>	m
Film temperatue	16.5	<sup>o</sup> C			
Pr	8.108557566				
Gr	5.51E+03				
GrPr	44680.97043				
Nu	3.0222816				
<b>ho</b>	<b>7.14564794</b>	W/m <sup>2o</sup> K			

D. Design of Condenser heat exchanger:

For the characteristics and design of the condenser, the data extracted from the literature review [1] is utilized. The condenser which is considered for the single stage LiBr – H<sub>2</sub>O is a forced air cooled condenser and some assumed values are shown in the Table III. To calculate overall heat transfer coefficient, there is a necessity of inside (F<sub>i</sub>) and outside (F<sub>o</sub>) fouling factor values, whose value is 0.09 m<sup>2</sup>-<sup>o</sup>C/kW [8].

The equation of Nusselt number (Nu) for external laminar flow (Re < 5x10<sup>5</sup>) over a flat plate is given by [6]

$$Nu = 0.332 Re_D^{0.5} Pr^{0.333} \quad 0.6 < Pr < 50 \quad (10)$$

air side average heat transfer coefficient h<sub>i</sub>. For transformation of refrigerant vapour to liquid, the inside average heat transfer coefficient (h<sub>i</sub>) is obtained by adopting an equation of condensation [5] which is mentioned below

$$h_m = 0.725 \left[ \frac{g \rho_l (\rho_l - \rho_v) h_{fg} k_l^3}{\mu_l (T_v - T_w) D_o} \right]^{0.25} \quad (11)$$

The physical properties are evaluated at mean temperature and vapour pressure which are required in Equation 11. The above mentioned equation is preferred in case of horizontal tubes, for practical cases, there is a need to consider 20% greater value than the average heat transfer coefficient while doing the characteristics calculations. The final characteristics are presented in the Table III.

Table- III: Characteristics of condenser heat exchanger

Input Data		Unit	Condensing heat transfer coefficient (hi)		Unit
Outside dia of tube	6.8	mm			
Inside dia of tube	6.35	mm	Avg temp	29.5	°C
k of Copper	383.2	W/m <sup>0</sup> C	Density of liquid	996.97	kg/m <sup>3</sup>
Condensing temp	32	°C	Density of Vapour	3.29E-02	kg/m <sup>3</sup>
Atm temp	27	°C	k of liquid	0.613	W/m <sup>0</sup> C
C <sub>p</sub> of air	1.005	kJ/kg <sup>0</sup> K	Latent Heat	2431.2	kJ/kg
ρ of air	1.225	kg/m <sup>3</sup>	Abs Viscosity of liquid	8.01E-04	kg/m-s
Enthalpy of water vapour at 72 °C	2629.46	kJ/kg	ρ <sub>L</sub> -ρ <sub>V</sub>	9.97E+02	kg/m <sup>3</sup>
Enthalpy of water at 32 °C	134.04	kJ/kg	T <sub>v</sub> -T <sub>w</sub>	5	°K
Refrigerant mass flow rate (m)	0.000402026	kg/s	<b>Hi</b>	<b>18407.39941</b>	<b>W/m<sup>20</sup>C</b>
<b>Condenser Design Load</b>	<b>1.003223721</b>	<b>kW</b>			
Volume of Blower (Assumed)	2	m <sup>3</sup> /s			
Mass flow rate of air	2.45	kg/s	<b>Overall HT Coefficient (U)</b>		
Temp rise of air	0.40744186	°K			
Final temp of air	27.40744186	°C	(Do/Di)(1/hi)	5.81759E-05	
Del T1	5	°K	(Do/Di)Fi	0.096377953	
Del T2	4.59255814	°K	(1/2k)Doln(Do/Di)	2.93012E-06	
LMTD	4.732		Fo	0.09	
<b>Air Side Heat transfer Coefficient (ho)</b>			1/ho	0.167045197	
Air Velocity(Assumed)	6	m/s			
Area	0.333333333	m <sup>2</sup>	U	2.828980313	kW/m <sup>20</sup> C
Face Dia	0.651469254	m			
Mean Temp of Air	27.20372093	°K	A	0.074941629	m <sup>2</sup>
Density	1.175946	kg/m <sup>3</sup>			
Abs Viscosity	1.8495E-05		<b>L</b>	<b>3.508030255</b>	m
C <sub>p</sub>	1005	J/kg <sup>0</sup> K			
k of air	0.0265256	W/m <sup>0</sup> C			
Re	248521.5933				
Pr	0.700759945				
Nu	147.0261941				
<b>Ho</b>	<b>5.986403792</b>	<b>W/m<sup>20</sup>C</b>			

**E. Design of Solution Heat Exchanger:**

For the design of solution heat exchanger the temperature at the outlet of pump is considered as 36°C [1], and the design is made to increase the solution temperature by 20°C. The remaining characteristics are mentioned in the Table IV.

The average overall heat transfer coefficient (U) for spiral tube and shell type heat exchanger is considered as 150W/m<sup>2</sup>-°C [9]. Corresponding to the R & S values of 1.168 & 0.487, F = 0.98 [6] By substituting the above mentioned and the Table IV values in Equation 1 and 2 results, A= 0.07756m<sup>2</sup> with respect to the heat transfer Q = 217.49W Length of the tube = 3.63m considered 4 m for construction. Outside diameter d<sub>o</sub> = 6.8mm and Inside diameter d<sub>i</sub> = 6.35mm. Spiral diameter = 0.06929m and the

Shell inside diameter by adding the clearance of 0.003m = 72.29 mm. Considered shell Outside diameter = 75mm

Table- IV: Characteristics of solution heat exchanger

Parameter	Type / Value
Hot solution inlet temperature	75°C (60%)
Hot solution outlet temperature	55.15°C
Hot solution Mass flow rate	0.005689047 kg/s
C <sub>p</sub> of Hot solution	1926 J/kg °C
Cold solution inlet temperature	36°C(55.76%)
Cold solution outlet temperature	56°C(55.76%)
Cold solution Mass flow rate	0.005287021kg/sec
C <sub>p</sub> Cold solution	2057 J/kg °C

**F. Desin of expansion device (Capillary):**

The outlet temperature of the expansion device is 6°C and the state of the refrigerant is in liquid. The pressure required at outlet of the capillary to maintain 6°C is 0.9437 kPa.



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From Table III, the inlet data of capillary is being obtained. To facilitate this temperature and pressure sufficient length of capillary is required with respect to the available diameters of 1/8” or 1/16”. If the considered diameter of the tube is 1/8” the remaining characteristics are obtained by using the below mentioned Equation 12. The loss of pressure due to viscous effect is calculated by utilizing the Darcy – Weisbach equation.

$$\frac{\Delta P}{L} = f_D \left( \frac{\rho}{2} \right) \left( \frac{v^2}{D} \right) \quad (12)$$

Darcy’s Friction Factor  $f_D = \frac{64}{Re}$ , by substituting required values which are mentioned in the Table III and some outlet data mentioned result the length of capillary (L) = 24.61m, considered 25m for construction.

### G. Design of Generator Heat Exchanger:

The generator heat exchanger characteristics are detailed in the Table V.

The generator gives sensible heat and latent heat of vaporization to the working solution. The sensible heat is utilized to rise the temperature of stream to saturation condition and the latent heat is required for phase change. After an extensive literature review it has been found very less research work has been done in this area.

The average overall heat transfer coefficient (U) varies from 1600 – 7500 W/m<sup>2</sup>-°C [7].

Table- V: Characteristics of generator heat exchanger

Parameter	Type/Value
Desorber load	1.39 KW
Pressure inside the desorber	4.82kPa

Table- VII: LiBr – H<sub>2</sub>O absorption refrigeration system calculations

Point	T(°C)	P(kPa)	m(kg/sec)	h(kJ/kg)	%Li-Br	Remarks
1	36	0.9437	0.005287021	88.16	55.76	
2	36	4.82	0.005287021	88.16	55.76	
3	55	4.82	0.005287021	127.07	55.76	Subcooled Liquid
4	75	4.82	0.005689047	183.23	60	
5	55.15	0.9437	0.005689047	144.92	60	
6	72	4.82	0.000402026	2629.46	0	Saturated Vapour
7	32	4.82	0.000402026	134	0	Saturated Liquid
8	6	0.9437	0.000402026	25.2	0	Saturated Liquid
9	6	0.9437	0.000402026	2512.6	0	Saturated Vapour

Each accessory energy transfer can be determined by the energy balance mentioned in the Equations 13-17

- Evaporator energy balance

$$\dot{Q}_e = m_9 h_9 - m_8 h_8 \quad (13)$$

And the result is  $\dot{Q}_e = 1.000031634 \text{ kW}$

- Condenser energy balance

$$\dot{Q}_c = m_6 (h_6 - h_7) \quad (14)$$

And the result is  $\dot{Q}_c = 1.003239802 \text{ kW}$

SHE to Desorber inlet mass flow rate ( $m_{SS}$ )	0.005287021kg/sec
Mass flow rate of refrigerant	0.000402026 kg/s
Desorber to SHE mass flow rate after vaporization ( $m_{ws}$ )	0.005689047 kg/s

### H. Design of Absorber Heat exchanger:

The characteristics of absorber heat exchanger are detailed in Table VI. The cooling is necessary to attain the outlet stream temperature to 36 °C, for this the natural convection is sufficient to attain 36 °C from 55.15 °C. The average overall heat transfer coefficient (U) of an absorber heat exchanger is about 650 W/m<sup>2</sup>°C.

Table- VI: Characteristics of absorber heat exchanger

Parameter	Type/Value
Absorber load	1.368 kW
Absorber Pressure	0.9437kPa
Evaporator outlet to absorber	0.000402026
SHE to absorber mass flow rate	0.005689047
Absorber to SHE mass flow rate	0.005287021

The calculated characteristics of LiBr – H<sub>2</sub>O absorption refrigeration system operates with a generator temperature of 75 °C are tabulated in the Table VII.

- Absorber energy balance

$$\dot{Q}_a = m_9 h_9 + m_5 h_5 - m_1 h_1 \quad (15)$$

And the result is  $\dot{Q}_a = 1.368161836 \text{ kW}$

- Pump power

$$= \dot{m}_1 v_1 (p_2 - p_1) \quad (16)$$

And the result is  $w = 0.012571498 \text{ kW}$

- Generator energy balance

$$\dot{Q}_g = m_4 h_4 + m_6 h_6 - m_3 h_3 \quad (17)$$

And the result is  $\dot{Q}_g = 1.38766169 \text{ kW}$

- The COP is calculated as

$$\text{COP}_a = \frac{\dot{Q}_e}{\dot{Q}_g} \quad (18)$$

And the result is  $\text{COP}_a = 0.720659539$

- The theoretical COP is calculated as

$$\text{COP}_{th} = \left( \frac{T_c}{T_c - T_e} \right) \left( \frac{T_g - T_a}{T_g} \right) \quad (19)$$

And the result is  $\text{COP}_{th} = 1.203566$

- The relative COP is calculated as

$$\text{COP}_r = \frac{\text{COP}_a}{\text{COP}_{th}} \quad (20)$$

And the result is  $\text{COP}_r = 0.6$

#### IV. CONSTRUCTION OF THE UNIT

All the heat exchanger characteristics are discussed and tabulated in Table VII. These are constructed in such a way that, if there is any need to increase the surface area of the tube to obtain the desired characteristics mentioned in the tables from I-VII. All the heat exchangers are arranged in such a way as shown in the Fig. 1. To maintain the unit in required parametric region, pressure, temperature and flow meters are installed. By using these we can measure the parameters at different points in the system and the final calculations are done based on the measured parameters, through this, the COP and the energies mentioned in the Table VII can be achieved.

##### A. Condenser Heat Exchanger:

The installed condenser in the unit is a tube and fin type one. A fan is used to blow the air on the condenser heat-exchanger causes the removal of latent heat from the refrigerant flowing inside the condenser tube. The outside diameter of the tube is of 6.8 mm and the inside diameter of the tube is 6.3 mm. These tubes are made up of copper which is having a heat transfer coefficient of  $383.2 \text{ W/m}^2\text{ }^\circ\text{C}$ . The length of the condenser tube considered in construction is 20% more than the designed value in the Table III. The effectiveness of the heat exchanger is 10% more than the designed value because of increase in average overall heat transfer coefficient. To facilitate the unbalanced pressure situations during the phase change in condenser, a Two-way pressure gauge between the generator and absorber is installed.

##### B. Generator (Desorber) Heat Exchanger:

There is an unavailability of data to correlate the average overall heat transfer coefficient, so the coil heater is considered of power capacity of 1500W to heat the solution

inside the generator shell. Coil heater power is regulated by using a dimmerstat and to notify the power usage of the coil, arranged a digital voltmeter and ammeter in series between the dimmer output to heater input. The renewable energy usage provision is provided to the generator and those lines are isolated with valves.

##### C. Absorber heat exchanger:

The absorber heat exchanger is constructed in the similar manner like the generator. The only difference is that the cooling is provided to the absorber by the phenomenon of natural convection. To decrease the outlet temperature less than the designed temperature, the forced convection phenomenon is adopted and a fan is arranged (Blower type). Pressure and temperature ports are created for measurement.

##### D. Evaporator heat exchanger:

For construction of evaporator heat exchanger a stainless steel can of 6Litres capacity is used. The final temperature of the refrigerating space is  $15 \text{ }^\circ\text{C}$  which is similar to the designed value results in 5 mins after attaining the required properties at all locations.

Copper tube of 6.8mm outside diameter, 6.35 mm inside diameter and 6m length is considered to run the refrigerant through it, which winds on the surface of the evaporator can. Then it is brazed with equal pitching. The entire evaporator setup is placed in a wooden cabinet and the remaining space is being insulated.

##### E. Solution heat exchanger:

The spiral heat exchanger type is installed as a solution heat exchanger in the constructed unit. The tubes are made of copper of 6.8mm outer diameter and 6.35mm inside diameter. With respect to the designed data considered a GI pipe of 75 mm outer diameter and thickness of 1.5mm. The length of the tube is 4m. Counter flow is considered for best results and achieved a rise in temperature of  $20 \text{ }^\circ\text{C}$  of strong solution with the considered length. The U value during the -experimentation is of  $165 \text{ W/m}^2\text{ }^\circ\text{C}$ , which is 10% greater than the assumed value.

##### F. Expansion Device (Capillary):

The capillary tube considered in unit construction of  $1/8''$  diameter and 25mm length copper capillary. The predicted pressure at the outlet of capillary in the design is obtained and the length predicted, matches with the designed value.

##### G. Pump:

A self-priming centrifugal pump is considered to facilitate the solution pumping between the absorber and generator (between two vacuum pressures). To regulate the required mass flow rate of solution an acrylic rotameter is installed at the outlet of the pump having a range of 0.2 to 2LPM. The specific gravity of the liquid which is calculated in the design is being utilized in the rotameter selection, which facilitates the required mass flow rate of solution across the unit. The installed pump contains a head range up to 24m.

The Constructed LiBr – H<sub>2</sub>O system illustrated in Fig. 3-6.

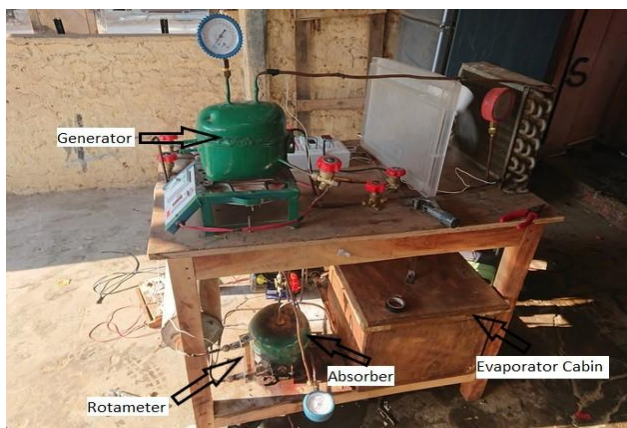


Fig. 3. Generator, Absorber and Evaporator

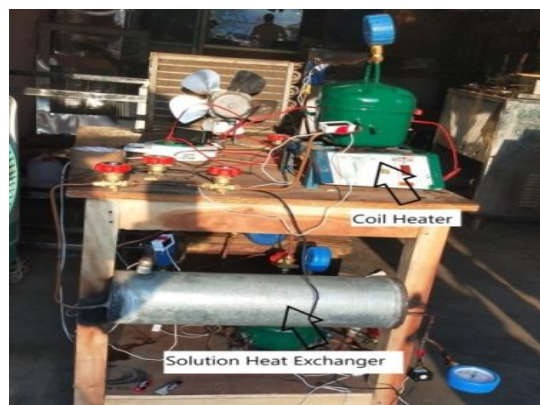


Fig. 6. SHE & Coil heater



Fig. 4. Pump and Condenser Fan



Fig. 5. Condenser

### V. ANALYSIS OF ABSORBER CONCENTRATION

Based on the general considerations the absorber, inlet concentration is of 55.76% (Lab Tested). By reducing the temperature of the absorber at constant pressure by transferring the heat to the cooling medium results a great impact on performance of the unit. The decrement in concentration at some constant parameters will reduce pump work, generator load and concentration ratio. The increment in COP and absorber load is due to absorber concentration.

Constant parameters are

- Generator temperature = 75°C.
- Condenser inlet temperature = 72 °C
- Condenser outlet temperature = 32 °C
- Generator and Condenser pressure = 4.82kPa
- Absorber and Evaporator pressure = 0.9347 kPa
- Evaporator temperature = 6°C
- Generator concentration = 60%

### VI. RESULTS AND DISCUSSION

The change in parameters observed during the expermination based on the mentioned methodology and fundamental assumptions in section Vand calculations are done by using the equations mentioned in the section III. And the obtained results are tabulated in Table VIII & IX.

Table-VIII: Change in Concentration ratio, mass flow rates & COP's

Absorber temperature (°C)	Absorber Concentration (%)	Concentrati-on Ratio	Mass flow rate of Strong Solution (kg/s)	Mass flow rate of weak solution (kg/sec)	COP (Actual)	COP (Thoritical)	COP (Relative)
19	45	3	0.001206078	0.001608104	0.852080978	1.728197	0.493046
22	47.5	3.8	0.001527699	0.001929725	0.835509875	1.635615	0.510823
26	50	5	0.00201013	0.00241215	0.814360358	1.512172	0.538537
31	52.5	7	0.002814182	0.003216208	0.785675481	1.357869	0.578609
35	55	11	0.004422286	0.004824312	0.740936671	1.234426	0.600227
36	55.76	13.1509434	0.005287021	0.005689047	0.720659539	1.203566	0.59877



Table- IX: Change in accessory heat load

Absorber Concentration	Absorber load (kW)	Pump work (kW)	Desorber load (kW)	Condenser load (kW)	Evaporator load (kW)
45	1.20492	0.00320	1.17363	1.00323	1.000031
47.5	1.22912	0.00395	1.19691	1.00323	1.000031
50	1.25613	0.00507	1.22799	1.00323	1.000031
52.5	1.28639	0.00692	1.27283	1.00323	1.000031
55	1.34052	0.01060	1.34968	1.00323	1.000031
55.76	1.36816	0.01257	1.38766	1.00323	1.000031

The change in the concentration ratio with respect to change in absorber concentration is plotted in the Fig. 7. A considerable decrease of 10 in the concentration ratio is observed due to the decrease in absorber concentration and generator outlet concentration is constant.

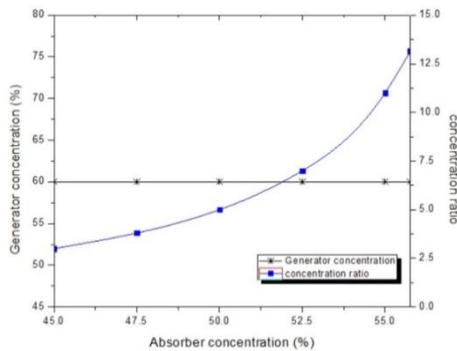


Fig. 7. Effect of Absorber concentration on concentration ratio

The change in the mass flow rates with respect to the change in concentration ratio is plotted in the Fig. 8. A considerable decrease in strong solution mass flow rate is found that is from 0.005287021 kg/s to 0.001206078kg/sec. In the similar manner observed the decrease in weak solution mass flow rate from 0.005689047 kg/s to 0.001608104 kg/s, throughout the experimentation procedure the mass flow rate of refrigerant remains constant.

The change in condenser heat load and absorber heat load with respect to change in absorber concentration is plotted in the Fig. 9.

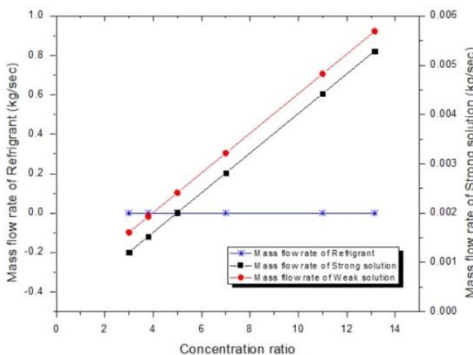


Fig. 8. Effect of Concentration ration on Mass flow rates

Decrease in absorber heat load from 1.368161836 kW to 0.852080978 kW is observed and the condenser load remains constant throughout the experimentation.

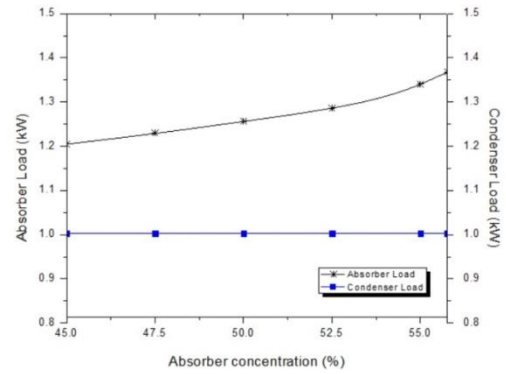


Fig. 9. Effect of Absorber concentration on Absorber load and Condenser load

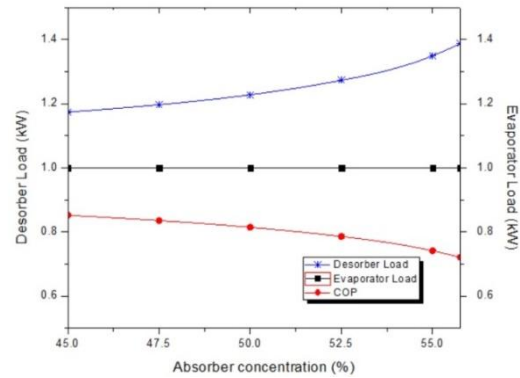


Fig. 10. Effect of Absorber Concentration on Desorber, Evaporator Loads & COP

Finally a considerable increase of 15% in COP by changing the concentration of absorber from 55.76% to 45%.by decreasing the concentration ratio the outlet temperature of the absorber is also changed (36 to 19 °C) The change in all the parameters with the change in Absorber concentration id plotted in the Fig. 11.

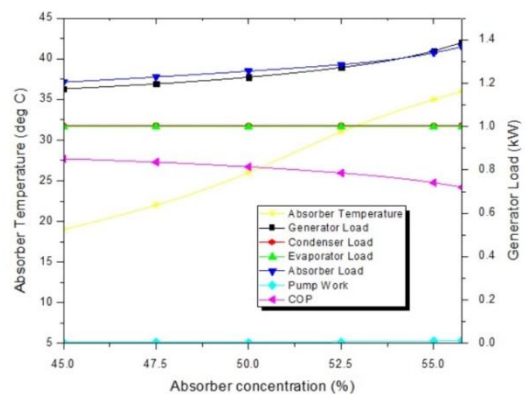


Fig. 11. Variation in Absorber temperature, each accessory load and COP with Absorber concentration

VII. CONCLUSIONS

The COP of the unit increases by 15% through the experimental analysis of 1kW unit by decreasing the concentration ratio from 55.76% to 45% with the constant evaporator temperature of 6 °C and the constant generator temperature of 75 °C. [1]

studied theoretically the effects on COP by reducing the absorber concentration and concluded that there is an increase of 12% in actual COP. Due to this reduced concentration ratio the generator load decreases from 1.38766169 kW to 1.173634502 kW and pump work decreases from 0.012571498 kW to 0.003200521 kW. By maintaining 60% solution concentration at generator and 0.000402026 kg/sec mass flow rate of refrigerant constantly, observed a greater decrease in concentration ratio, mass flow rates of strong solution and weak solution. By the reduction in concentration ratio, there is an increase of 43.58% of theoretical COP and 21% decrease in relative COP.

Finally, this analysis provides the best operating concentrations for absorber of a Single stage LiBr-H<sub>2</sub>O system.

By replacing the air cooling of absorber with different types of cooling techniques to predict which method results best performance as a future work, and by installing a thermosiphon phenomenon instead of mechanical pump to reduce the pump work. As provided the provision for the future scope the renewable or even waste heat recovery system can be installed to provide the heat energy to the generator. Parametric analysis can be performed on the fabricated unit with different energy sources, different concentration ratios and different flow rates.



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