Design and Fabrication of 2kW LiBr-H_2O absorption Air Conditioning System

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Abstract*- The objective of this paper is to design and fabricate a LiBr-H_2O Vapour Absorption System (VARS) for air-conditioning purpose. Design of various components like evaporator, condenser, heat exchanger used in system will be shown and their capacity will be given. The fabricated system will be of 2KW cooling capacity and its working fluid will be LiBr-H_2O solution where LiBr acts as absorbent and water acts as refrigerant. Present day problems with absorption air conditioning system are described and accordingly its optimization is suggested. The fabricated VAR system will be tested and results obtained will be compared with similar literature data and cost analysis for whole system will be done.*

Keywords- LiBr-H_2O; Vapour Absorption Refrigeration System; air conditioning; 2kW system.

Nomenclature

h	Specific enthalpy (KJ/kg)
m	Mass flow rate (kg/sec)
P	Pressure (kPa)
Q	Heat transfer rate (kW)
s	Specific Entropy (kJ/kg^{O}K)
T	Temperature (^{O}K)
X	LiBr mass fraction (%)
COP	Coefficient of Performance

I. INTRODUCTION

In hot climate the heating and cooling demand of domestic dwelling can be reduced substantially with various measures such as good insulation, double glazing, use of thermal mass and ventilation. Due to the high summer temperatures, the cooling demand cannot be reduced to the level of thermal comfort with passive and low energy cooling techniques and, therefore, an active cooling system is required. It is a preferable that such a system is not powered by electricity, the production of which depends entirely on fuel. Solar energy, which is available in such climates, could be used to power an active cooling system based on the absorption cycle. A number of refrigerant-absorbent pairs are used, for which the most common ones are water-lithium bromide and ammonia-water. These two pairs offer good thermodynamic performance and they are environmentally benign. Among various absorbent and refrigerant pair, LiBr-Water is most promising in chiller application due to high safety, high volatility ratio, high affinity, high stability and high latent heatLi-Br water absorption units are the most suitable for solar application, since low cost solar collectors used to power the generator of the machine. Such absorption units though, are not radically available in small in small residential sizes. After a search in the world market, only one manufacture was found commercially producing Li-Br water absorption refrigerator. Therefore, the possibility of producing absorption air conditioning system in small sizes for residential buildings and the economics of using such a refrigerator, assisted by solar energy, need to be investigated.

Various literature data were investigated for absorption system. Thanga et al.[1](2015) showed performance improvement of diesel engines using waste heat. Bachhav et al.[2] (2014) presented a paper in which micro channels were used to reduce size of condenser coil without reducing its effectiveness, means condenser was optimized and a cost effective and energy efficient system was built. Mendoza et al. [3] (2014) stated that a new combined absorption system using a scroll expander and three different working fluids using ammonia as refrigerant is proposed for the cogeneration of mechanical power and cooling. It can be driven by low solar thermal collectors or other suitable sustainable energy resources. Romero et al. [4] (2014) showed various experimental results of different absorber designs are reviewed in paper so that the generator and absorber size could be reduced. Thakre et al. [5] (2014) presented cooling of truck cabin using engine exhaust. Patil et al. [7] (2014) demonstrated that heat transfer rates were improved using micro channel condensers and various results of other papers were also reviewed and their data was analyzed. Rahman et al. [8] (2014) showed hypothetical designing of LiBr absorption system for air conditioning using waste heat from steam turbine exhaust of industries.

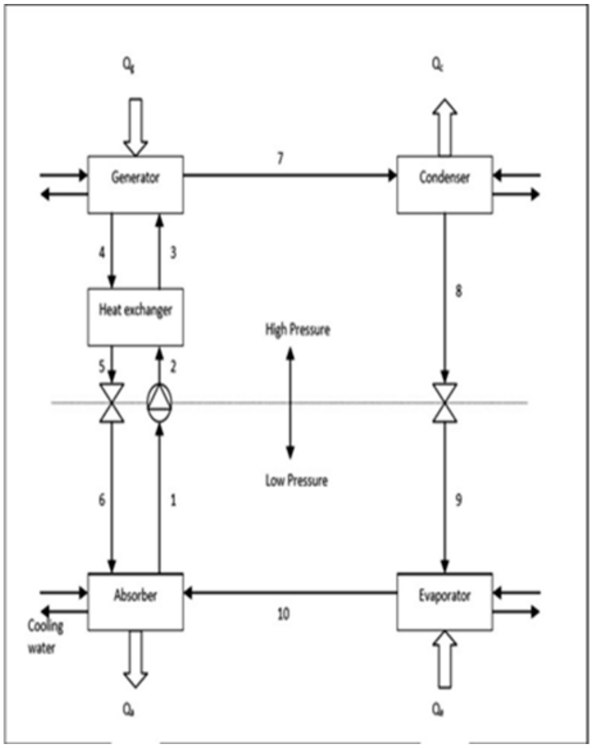

Fig. 1Schematic of a Single effect LiBr-Water Absorption System

At same pressure, the vapors from evaporator are absorbed by LiBr-aqueous solution supplied form generator having higher concentration of LiBr. The absorption of water vapor reduces the concentration of LiBr in aqueous solution; this solution is then passed to generator through pump at higher pressure. A solution heat exchanger is used between absorber and generator to increase the efficiency of system.

In this paper, a 0.56 TR (2kW) capacity absorption system is fabricated; to which heat is supplied from generator of 85^0C temperature at atmospheric pressure. This low grade energy can be obtained from solar panel, cooling of automobile engine, and other source of waste heat. Various components of the cycle are configured using empirical correlations and their specifications had been determined. To release the heat, cooling water is circulated in absorber and condenser at ambient temperature.

II. PROBLEM STATEMENT

1. The present VARS is bulky, heavy and hence unsuitable for mobile applications. Hence it should be made more compact.
2. The Coefficient of Performance (COP) of existing VARS is very low compared to VCRS, so it should be improved to give better performance.

III. HEAT AND MASS TRANSFER CALCULATIONS

3.1 Assumptions taken:

1. Condenser temperature= 35^oC
2. Evaporator temperature= 7^oC
3. Absorber temperature= 30^oC
4. Generator temperature= 85^oC

Pressure values taken from p-h chart of water as refrigerant for condensing temperature 35^oC and evaporating temperature 7^oC.

P_E= 1kPa

P_C= 5.696kPa

Assuming 60% concentration of strong LiBr solution and 55% concentration of weak LiBr solution.

3.2. Mass Flow Rate Calculations

Mass flow rate in evaporator (m_9) = Load / (Change in enthalpy) Mass flow rate for weak and strong solution

$m_4 \times X_{ia} = m_3 \times X_{ig}$

Also, $m_3 = m_4 + m_7$ and

$m_1 = m_2 = m_3 = 0.01010$ kg/sec ;

$m_4 = m_5 = m_6 = 0.00928$ kg/sec ; and

$m_7 = m_8 = m_9 = m_{10} = 0.000844$ kg/sec

3.3. Heat Transfer Rate

Heat transfer rate in evaporator is taken as 0.56 TR (Load), based on this selected parameter heat transfer rate at other components are calculated as follows.

3.3.1. At Condenser

$Q_c = m_7 (h_7 - h_8)$

= **2.113kW**

h_7= Enthalpy of super-heated steam at saturation temperature of solution in generator and Condenser pressure

h8 = Enthalpy of water (saturated liquid) at condenser pressure and saturated temperature.

3.3.2. At Generator

$Q_g = m_4 h_4 + m_7 h_7 - m_3 h_3$

=**2.725kW**

h_4 = Enthalpy of solution at exit of generator temperature and X_{ia} Concentration

h_3 = Enthalpy of solution at inlet of generator temperature and X_{ig} concentration.

3.3.3. At Absorber

$Q_a = m_6h_6 + m_{10}h_{10} - m_1h_1$
= **2.567kW**
h_6 = Enthalpy of solution at inlet of absorber temperature and X_{ia} concentration
h_{10} = Enthalpy of saturated water vapor at evaporator pressure and its saturation temperature.
h_1 = Enthalpy of solution at exit of absorber temperature and X_{ea} concentration.

3.3.4. At Solution Heat Exchanger

$Q_{sx}= m_1(h_3-h_2)$
= **0.416Kw**

3.4. Coefficient of Performance (COP)

COP (ideal) = Heat taken out in Evaporator / Heat supplied in generator
= $Q_e/Q_{g=}$ **0.734**

In the above ideal COP, work by the pump, pressure drops and other losses are not included. Lower value of COP for absorption system compared to VCRS is becauseof using low grade energy rather than highly concentrated electrical energy.

IV. DESIGN OF COMPONENTS

4.1Assumptions taken:

C_p (water) = 4.2kJ/kgK
Effectiveness (Ɛ) = 0.6

4.2 Design Calculations for each component of Absorption System

4.2.1 Design calculations for Generator

The generator is a helical coil heat exchanger and its shell is made of M.S. and the helical tube is of copper. LiBr solution is filled inside the shell and the exhaust will flow through the copper tube thus heating LiBr-water solution and separating the vapour which travels condenser. We make use of NTU method for designing the generator.

$NTU= AU_m/C_{min}$
Where A= total heat transfer area
U_m= overall heat transfer coefficient
C_{min}= heat capacity
$A_g=(NTU \times C_{min})/ U_m$
$= 0.111m^2$

As $A_g= \pi d_oL$
So, L= 2.96m
Pitch P= $1.5d_o$= 0.018m

The diameter(D) of the helix that can be made and one which is feasible to fabricate is:
D= **0.15m**
No. of turns of helical tube:
$L= N\times ((2\pi R)^2+ P^2)^{0.5}$
So, N=**7**

The shell diameter can be selected from manufacturer catalogue that is best suitable for tube helix diameter:
D_{si}= 0.2m
D_{so}= 0.22m
Thickness= 0.01m

4.2.2 Design calculations for solution Heat Exchanger

Solution heat exchanger is used in system to transfer heat between strong (hot) solution from generator and weak solution from absorber tank.

For designing solution heat exchanger we make use of LMTD method.

LMTD obtained is **21.32^{o}C**
Prandtl number for strong solution(Pr_{ss}) = ($\mu C_P/K_f$) = 14.72.
Reynolds number for strong solution (Re_{ss}) = ($4m/\pi\mu_f d_i$) = 298.37.
Nusselt number, $Nu_{ss}= 0.023\times (Re_{ss})^{0.8} \times (Pr_{ss})^{0.4}$
Nu_{ss} =6.43

Outside heat transfer coefficient (h_o)= ($Nu_{ss} \times (K_{SS})$)/ d_o
= **228.8 W/m^{2}K**

Similarly calculating for weak solution:
Inside heat transfer coefficient (h_i) = **221.379 W/m^{2}K**
Overall heat transfer coefficient (U):
$1/U= 1/h_o + 1/h_i$
U= 112.51 W/m^{2}K
$Q_{hx}= U\times A_{ex}\times \Delta t_{LMTD}$
So, $A_{ex}= 0.128m^2$
Thus L= 3.1m
And P= 0.018m

The diameter of helix that can be made and the one which feasible to fabricate is **D= 0.150m**
No. of turns of helical tube:
$L= N\times ((2\pi R)^2+ P^2)^{0.5}$
So, N=**7**

The shell diameter can be selected from manufacturer catalogue that is best suitable for tube helix diameter:

D_{si}= 0.2m

D_{so}= 0.22m

Thickness= 0.01m

4.2.3 Design calculations for Evaporator

The evaporators are used for cooling and dehumidifying the air directly with help of refrigerant flowing inside tubes made of copper.

(For Water/ tube inside):

Prandtl number (Pr) = ($\mu C_P/K_f$) = 11.18

Reynolds number (Re) = ($4m/\pi\mu_f d_i$) = 78.165

Nusselt number, Nu= 0.023× $(Re_{ss})^{0.8}$ × $(Pr_{ss})^{0.4}$

Nu = 1.9744

Tube side heat transfer coefficient (h_i) = NuK_f/d_i

= 124.186W/m^2K

Now similarly for air/tube outside:

Assuming U=40m/sec, m_a = 0.525 kg/sec, Q_a = 2.63m^3/sec

Face area F_a= Q_a/u = 0.065m^2

Equivalent diameter D_e= $((4/\pi)\times F_a)^{0.5}$= 0.0287m

Prandtl number (Pr) = ($\mu C_P/K_f$) = 0.708

Reynolds number (Re) = ($4m/\pi\mu_f d_i$) = 1029152.97

Nusselt number, Nu= 0.023× $(Re)^{0.8}$ × $(Pr)^{0.4}$

Nu = 893.88

Air side heat transfer coefficient (h_o) = 81.55 W/m^2K

Overall heat transfer coefficient (U):

$1/U= 1/h_o + 1/h_i$

U= 49.22 W/m^2K

Q= $UA\Delta T_{LMTD}$

Therefore, Area of evaporator (A_e) = **2.7161m^2**

4.2.4 Design calculations for Condenser

Condenser consists of coils placed in a number of rows with fins mounted on it to increase heat transfer area. Material used for making tubes is copper.

Prandtl number (Pr) = ($\mu C_P/K_f$) = 4.5521

Reynolds number (Re) = ($4m/\pi\mu_f d_i$) = 169.493

Nusselt number, Nu= 0.023× $(Re_{ss})^{0.8}$ × $(Pr_{ss})^{0.4}$

Nu = 2.56

Tube side heat transfer coefficient (h_i) = NuK_f/d_i

= 177.152 W/m^2K

Now similarly for air/tube outside:

Assuming u = 35m/sec , m_a= 1.05kg/sec

Face area F_a= Q_a/u = 0.07m^2

Equivalent diameter D_e= $((4/\pi)\times F_a)^{0.5}$= 0.30m

Prandtl number (Pr) = ($\mu C_P/K_f$) = 0.714

Reynolds number (Re) = ($4m/\pi\mu_f d_i$) = 666374.47

Nusselt number, Nu= 0.023× $(Re)^{0.8}$ × $(Pr)^{0.4}$

Nu = 685.31

Air side heat transfer coefficient (h_o) = 59.66 W/m^2K

Overall heat transfer coefficient (U):

$1/U= 1/h_o + 1/h_i$

U= 44.632 W/m^2K

Q= $UA\Delta T_{LMTD}$

Therefore, Area of condenser (A_C) = **6.80 m^2**

4.2.5 Design calculations for Absorber

At absorber mixing of absorbent and refrigerant vapours takes place. For mixing to occur it is equipped with heat rejecting design as absorption takes place at low temperature only.

Heat rejected from absorber Q_a= 2.567kW

Specific heat of solution C_{pw}= 3.09 × 10^3J/kg^{o}C

Temperature of inlet air = 27^{o}C

Temperature of outlet air = 29^{o}C

Heat rejected by absorber Q_a= heat removed by air Q_{air}

Q_{air}= $m_a \times C_{pa} \times \Delta t$

Therefore **m_a = 1.27kg/sec**

Face area of fan (f_a) = 0.07m^2

mass of air can also be written as m_a= p × q

so, **q = 1.0794m^3/s**

Velocity of air, v = q/f_a

v = 15.42m/s

Rejected heat, $Q_{ab} = m_{ws} \times C_{pw} \times \Delta t$

so, m_{ws} = 0.103kg/s

LMTD = 15.86^{o}C

Heat transfer area of absorber is calculated as:

$Q_{ab} = A \times U \times \Delta t_{LMTD}$

A = 4.046m^2

V. COST ANALYSIS

The unit which will be manufacture is a prototype unit and the total cost of its construction is about Rs.25000 a large part of this amount is spent for flow meters, auxiliary equipment like vacuum pump, glass tubes, and experimentation with different types of valves and materials. Other equipment's for weld testing and soundness is necessary the cost of such infrastructure is not included in the unit costing. A breakdown of the system is shown in the table.

Table 1 Cost estimation

Component	Cost
Generator and Condenser unit	3500
Evaporator	2500
Absorber	2000
Piping	1500
Pumps	2000
Li-Br water solution	3000
Valves, regulators and sockets	1500

VI. CONCLUSIONS

1. The Absorption Air Conditioning System is a feasible alternative to the traditional vapour compression system for automotive case. The Absorption Air Conditioning System use environmentally refrigerants and very little power for operation when compared to traditional vapour compression system.
2. The reduction in power can be achieved because the system can be operated using waste heat rejected, for example from the engine coolant system and because no compression is required.
3. Finalized the capacity of absorption system with the help of various standard papers and literature data.
4. The absorber refrigerant pair of LiBr- water would be used in VARS for air conditioning system.
5. Temperature drop of about 3-4^{0}C below the atmospheric temperature is expected output from system.
6. Coefficient of performance (COP) obtained will be about 0.67 for such absorption air conditioning system.

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