DESIGN AND ANALYSIS OF SOLAR VAPOUR ABSORPTION COOLING SYSTEM

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Abstract-- A simple ammonia-water absorption system is driven by solar power is analysed. Also presents the performance of low capacity single stage absorption cooling systems, suitable for small building and residential applications. The primary heat source is heat energy supplied from solar collectors. When evaporator temperature is maintained at constant temperature and the condenser temperature is fixed at particular temperatures. Design of elements such as generator, evaporator, and condenser are presented. Work has been carried out to check the COP which is obtained as 0.58 and Circulation ratio using solar based system to observe the variables like flow rate and composition.

Index Terms — **Solar energy, vapour absorption refrigeration system (VARS), water heating, generator, evaporator**

I. INTRODUCTION

Global warming and climate change are because of industries which use fossil fuels burning of coal, gas and oil, they release harmful gases, which trap heat in the atmosphere and cause global warming [1]. The refrigeration sector including heat pumps, air conditioning and cryogenics accounts for global greenhouse gas emissions about 7.8 %. Among these emissions of HFCs, CFCs and HCFCs about 63% are indirect emissions related to the production and transportation of refrigeration systems (CO_2 emissions mainly) [2]. The energy prices are increasing exponentially worldwide. One of the most energy consuming sector is the industrial refrigeration. For cutting down the energy costs and preserving our environment use of vapour absorption refrigeration system is necessary.

Greenhouse gases and their damaging effects on the atmosphere are put in attention. The domestic refrigerator-freezers uses R-134a as an alternative refrigerant contribute indirectly to global warming. Also, the amount of electrical energy consumed by the system leads to an indirect global warming, because production of 1 kW electricity by the power plant emits 0.6 kg of CO_2 to the environment. This contribution is nearly 100 times greater than the direct contribution of the refrigerant alone. Worldwide about 62 million new units are being manufactured every year and hundreds of millions are currently in use. Besides, climatic changes because of energy consumption have become an international issue that was treated in every international summit. To reduce energy consumption and CO_2 gas emissions only concept will be a solar chilled water production through absorption cycles may be considered and is one of the most desirable applications. To drive cooling cycles for air conditioning use of solar energy is an attractive concept because of the coincidence of peak cooling loads with the available solar power [3].

II. DETAILS OF SOLAR COLLECTOR

Cover plate of 3.2-6.4 mm thick (glass tempered with a low iron content). The collector has transmittance about 85%. Copper is used as an absorber plate because of its high conductivity and is corrosion resistant. The dimensions of copper plates as 0.05 mm thick and 1.25 cm diameter tubes. Tubes efficiency is 97 % and are spaced 15 cm apart. The black body made of copper plate painted with black color which has absorptance about 0.85-0.9 and emittance is 0.08-0.12. Steel, Aluminum or fiber glass, one of these materials are used as enclosure/insulation. Fiber glass is widely used.

III. MATHEMATICAL MODELING

The enthalpy concentration chart helps to determine the system operating pressures to carry further calculations,

A. Condenser Pressure (P_c)

Where in the water-cooled condenser, the water is supplied at atmospheric temperature 30°C to change vapour to liquid refrigerant. Corresponding pressure can be noted from the ammonia refrigeration table (R-717). Then the condenser pressure is fixed which is, P_c=11.66 bar

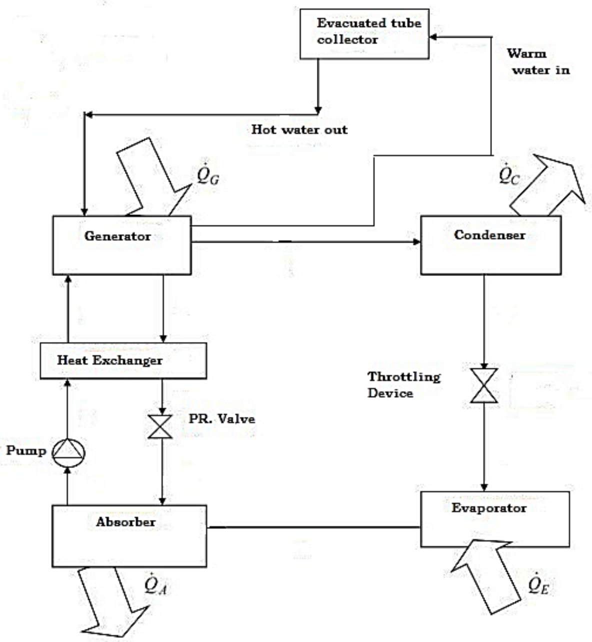

Fig: 1 Design layout of solar powered vapour absorption system

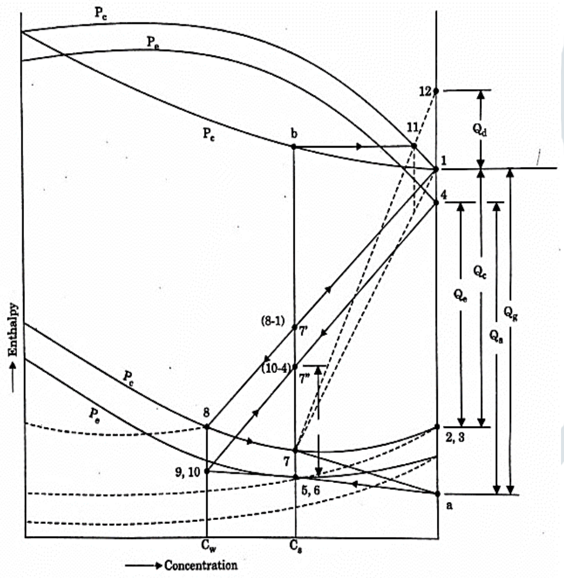

Fig: 2 Enthalpy concentration charts

B. Evaporator pressure (P_e)

The evaporator Pressure of the evaporator can be determined based on the minimum temperature required to be maintained in the evaporator chamber. The Pressure of the evaporator is kept close to the atmospheric pressure (1 bar), to ensure design economy. The corresponding evaporator saturation temperature (ammonia vapours) becomes -330C [4]. Now the pressure points of condense and evaporator can be plotted on the pressure enthalpy chart as points 1, 2, 3 and 4. Pure NH_3 saturated vapour at condenser pressure P_c is represented by Point 1 and concentration C=1. Pure NH_3 saturated liquid represented by point 2 at P_c and C=1.

This point is marked in liquid region. The condition of pure NH_3 (wet) is represented by point 3 but at pressure P_e and C=1. Point 2 coincides with point 3 as 2-3 is a throttling process in which enthalpy remains constant. The condition of pure NH_3 is represented by Point 4 at pressure P_e these are saturated vapours which absorbs heat in evaporator and converts from wet vapour to saturated vapour. This point is marked in vapour region.

The enthalpies from the chart at points 1, 2, 3 and 4 can be noted as h_1= 1635 kJ/kg, h_2= h_3= 500 kJ/kg and h_4= 1530 kJ/kg [5]. Now, let us consider 1TR will be the refrigeration capacity of the unit.

1. The refrigeration effect produced by the ammonia refrigerant in the evaporator is

$Q_e = h_4 - h_3$ kJ/kg of ammonia. (3.1)

Say, M_r be the mass flow rate of ammonia in the evaporator

$M_r \times (h_4 - h_3) = 1$ TR (3.2)

Therefore, $M_r = 0.20$ kg/min

2. Now, the temperature of the water entering into the generator will be more than 80 ºC (say, 84 ºC). considering this temperature in the generator T_g= 80ºC (assuming losses) [4] Thus, the point 8 can be marked on the pressure enthalpy chart, where 80 ºC temperature line and 11.66 bar pressure line intersects. The hot weak liquid having concentration C_w inside the generator is represented by point 8. From the chart the corresponding concentration of the weak solution can be directly noted down as C_w= 0.425. The point 5 can also fixed only after fixing the point 8. The strong aqua coming out of the absorber after absorbing the vapours coming out of the evaporator is represented by point 5. The concentration of the strong solution, say C_5, can be determined by knowing the degasifying factor.

The amount of NH_3 vapours removed from the strong solution in the generator is called as degasifying factor. To prevent water from being evaporated, which creates trouble, and is necessary to be removed before entering into condenser for that higher value of this factor is. In this system, a mass of 0.20 kg/min is required to flow across the evaporator for steady state. The degasifying factor becomes 0.20. Thus, the concentration of strong aqua solution becomes C_5= C_w + 0.18 i.e., C_5= 0.425 + 0.20 = 0.625 (1) taking the concentration of strong solution to be C_s=C_5=0.625. Hence, we know the concentration and pressure at point 5, thus point 5 can be located on the chart at C_5= 0.625 and pressure P_c = 11.66 bar.

Point 6: The concentration C_5= 0.625 this is the condition of the aqua solution, but the pressure is increased from P_e to P_c as it passes through the pump.
Enthalpy does not change when the aqua pressure increase passing through the pump, is to be represented on C-h chart where point 6 coincides with point 5
Point 7: As the low temperature strong aqua solution passes through heat exchanger it gains the heat and its enthalpy increases, but its concentration C_s remains same as well as pressure remains same as P_c. Now the point 7 can be marked on the C-h diagram as pressure at point 7 and concentration C_7 are known.
Extend the line till it cuts the Y axis i.e. enthalpy, and is marked as point 'a' after joining the points 8 and 7, then join points 'a' and 5 and extend till it cuts the vertical line passing through 8. From this the position point 9 and 10 can be decided.
Point 9: This represents the condition of weak liquid coming out of the heat exchanger after giving heat to the strong solution. So, enthalpy is reduced. Subtracting the heat lost by the weak solution in heat exchanger, point 9 can be marked and it tells that concentration does not change.
Point 10: The enthalpy at point 9 and 10 is same but the point 10 is at lower pressure (P_e) because the point 10 is after the pressure reducing valve.

IV. DESIGN OF COMPONENTS

a. Mass flow rate of ammonia refrigerant,
M_r = 0.20kg/min

b. Heat removed in the evaporator = refrigeration effect

$$= M_r \times (h_4 - h_3) = 1\ \text{TR} = 210\ \text{kJ/min} \qquad (4.1)$$

If cold water flow rate is M_w then, $M_w C_p\ \Delta T$ = 210 kJ/min
If, $\Delta T = 17^0C$ Then, M_w = 3.0 kg/min

c. Heat removed in the condenser by the circulating cooling water is given by the equation:

$$Q_c = (h_1 - h_2)\ \text{per kg of ammonia} \qquad (4.2)$$
$$= M_r \times (h_1 - h_2) = 231.4\ \text{kJ/min}$$

d. Heat removed from absorber.

When the NH_3 vapor at point 4 and aqua at point 10 are mixed, the resulting condition of the mixture in the absorber is represented by 7" and after losing the heat in the absorber (as it is cooled), the aqua comes out at condition 5.

$$Q_a = (h_{7''} - h_5)\ \text{per kg of aqua.} \qquad (4.3)$$

Extend the triangle 10-7"-5 towards right till 10-7" cuts at 4 and 10-5 cuts at point 'a' on x axis. Therefore, heat removed per kg of NH_3 is given by, $Qa = (h_4 - h_a)$ per kg of ammonia

$$= M_r \times (h_4 - h_a) = 291.55\ \text{kJ/min}$$

Now the resultant aqua is at condition 7", which loses heat up to condition 5.
Temp at 7" = i.e. $T_{7''}$ = 85ºC (from C-h chart)
Say, water gets heated to a temp of 85ºC from 30^0C while removing heat from the absorber.
If, M_w = mass of cooling water required

$$\text{Then, } M_w \times C_p \times (T_i - T_O) = 291.55, \qquad (4.4)$$
$$M_w \times 4.18 \times (85 - 30) = 291.55$$
$$M_w = 1.244\text{kg/min}$$

e. Heat supplied in the generator,

Say Q_g is the heat supplied in the generator and Q_d is the heat removed from water vapour then the net heat removed per kg of aqua is given by,

$$q_g - q_d = (h_{7'} - h_7)\ \text{per kg of aqua} \qquad (4.5)$$

as, the aqua goes in at condition 7 and comes out at condition 8 and 1, which can be considered as a combined condition 7'. By extending the triangle 8-7-7' towards right till 8-7'cuts at 1 and 8-7 cuts at a on y axis, then the heat removed per kg of NH_3 is

$$Q_g - Q_d = (h_1 - h_a)\ \text{per kg of ammonia}$$

Now for finding out Q_d separately, extend the vertical line 7-7' till it cuts the auxiliary line P_c and mark point 'b' as shown. Then draw a horizontal line through b which cuts P_c line in vapour region at point 11. Then join the points 7 and 11 and extend the line till it cuts y axis at 12.
Then Q_d is given by,

$Q_d = (h_{12} – h_1)$ per kg of ammonia (4.6)
$= 29.56$ kJ/min
Now, using equation $Qg – Q_d = (h_1 - h_a)$, (4.7)
$Q_g - 29 = 0.20 \times (1635 - 100)$
Therefore, $Q_g = 336$kJ/min

Thus, the amount of heat required in the generator for running the unit is given by, $Q_g = 342.52$ kJ/min

Now this amount of heat is provided by the hot water coming out of solar collector. The temp of hot water coming out of solar water heater is about 84 °C. We can reasonably assume that the heating in generator is produced at about 80 °C, considering heat loss during the process.

f. Calculations of solar water heater.

Useful energy (energy absorbed by the collector plate) is given by,
$Q_u = K \times S \times A$ (4.8)
Where, K = efficiency of collector plate (assume k=0.85)
S = average solar heat falling on earth's surface = 6 kw- hr./m^2/day = 250 W/m^2
A = Area of collector plates

Now, Heat required in the generator, $Q_g = 5708.7$ W

Hence approximate area of the collector plates required for providing this much amount of energy,
$= 5708.73 / (250 \times K) = 24\ m^2$ (approx.)

Total Area of collector tubes $A = 24m^2$

Therefore, we can use 4 collector plates of having dimensions of 3x2 m^2.
Thus, $Q_u = 0.85 \times 250 \times 24 = 5100$ J/s

The energy absorbed by the collector helps in heating of the water flowing in the tubes of the collector plates.
$U = m \times C_p \times (T_o\text{-}T_i)$ (4.9)

Let, the rate of water flowing through the tubes,
m = 1.2 kg/min (typical example)
Specific heat of water, $C_p = 4200$ J/kg/k
T_o = Outlet temp. of water in the collector tube
T_i = Inlet temp. of water in collector tube=30^0C
Therefore, $Q_u = 0.02 \times 4200 \times (T_o - 30)$ (4.10)
i.e. $5100 = 0.02 \times 4200 \times (T_o - 30)$
therefore, $T_o = 84^0$C

The temperature (T_o) should be the inlet temperature to generator, but assuming heat loss while water flowing through the tubes. Hence net heating in the generator can be assumed to be taking place at 80^0C.

This is the net heat input to the system, which runs the system of 1 TR capacity. The work done by the pump for raising the pressure is negligible and hence neglected.

Total number of tubes required;
N = Area of each tube/ Area of each tube = A/A'
Area of each tube = A'= 0.765 m^2
So; N = 30, Therefore total number of tubes required are 30.

h. Outlet temperature of water from collector tube $= (Qu/ (m\times c_p)) + T_i$ (4.11)
T_i = inlet Temp of water in collector tubes= 30^0C
M = Mass flow rate of water through tubes = 0.0243 kg/s
$= (5520/(0.0243\times4.2)) + 30 = 84^0$C

Therefore, outlet temperature from collector tubes are 84^0C; which is an inlet temperature to the generator.

V. DESIGN OF EVAPORATOR

Evaporator is an equipment in which refrigerant vaporizes to generate the desired refrigeration. It is also known as chiller. Considering the evaporator made of tubes and air cooled.

Heat absorbed in the evaporator, $Q_e = 210$ kJ/kg = 3500W
Let air inlet and outlet temperatures of evaporator, $T_i = 30^oC$ and, To = 2^oC
Temperature of refrigerant entering the Evaporator, $T_{ei} = -32^o$C
Therefore, $\theta_1 = T_i – T_{ei} = 30 - (-32) = 62^o$C
$\theta_2 = T_o – T_{ei} = 2 - (-32) = 34^o$C
$LMTD(\theta_m) = (\theta_1\text{-}\theta_2)/\ln (\theta_1/ \theta_2)$ (5.1)
$= (62\text{-}34) / \ln (62/34) = 46.6^o$C

Taking, Overall heat transfer coefficient to be, U = 1000 /m^2
Taking correction factor as, F = 1
$Q_e = FUA\theta_m$,$3500 = 1\times1000 \times A_e \times 46.6$ (5.2)
Area of the evaporator, $A_e = 0.075\ m^2$
Considering the number of evaporator tubes, n = 1
Length of each tube, L = to be calculated
Diameter of evaporator tube, D = 0.01m

The effective area of evaporator; $A_e = n \times \pi \times D \times L$ (5.3)

$0.075 = 1 \times \pi \times 0.01 \times L$

therefore, L = 2.39m

COP of the system

The COP of the refrigerating unit can be calculated by using the equation:

$$COP = \frac{\text{Reftrigeration Effect}}{\text{Heat given to generator}} = \frac{Qe}{Qg} \quad (5.4)$$

Neglecting pump work, Therefore, COP = 0.613

Now the COP of the system as a whole (system including solar water heater) can be calculated as,

$$COP = \frac{\text{Net Refrigeration Effect Produced}}{\text{Heat input to the generator}}$$

$$\text{Heat input at the collector} = \text{Solar constant} \times \text{Area}$$
$$= 250\ W/m^2 \times 24\ m^2$$
$$= 6000\ W = 360\ kJ/min$$

Therefore, COP of the system is given by, COP = 0.58

VI. RESULTS AND DISCUSSION

Designed calculations have been done for necessary components and the values with respect to various parameters are listed in the below tables.

Table:6.1 Solar collector design

Sr. No	Parameters calculated	Values
1	Mass flow rate of refrigerant in evaporator (M_r)	0.203 kg/min
2	Concentration of strong solution (C_s)	0.628
3	Heat removed in evaporator (Q_e)	210 kJ/min
4	Heat removed in condenser (Qc)	231.40 kJ/min
5	Heat removed in absorber (Q_a)	291.55 kJ/min
6	Heat given in the generator (Q_g) i) Heat removed from water vapour (Q_d)	5708.73 W 29.563 kJ/min
7	Area of collector plates(A)	24 m^2
8	Useful energy (Q_u)	5520 J/s
9	No. of tubes	35
10	Outlet Temp of water from collector tube	84.08 °C

Table: 6.2 Evaporator design

Sr. No	Parameters calculated	Values
1	Heat absorbed in evaporator (Q_e)	3500 W
2	Effective area of the evaporator (A_e)	0.075 m
3	Length of evaporator tube (L_e)	2.390 m^2

Table: 6.3 Condenser design

1	Heat removed in condenser (Q_c)	3856.79 kJ/min
2	Effective Area of the condenser (A_c)	0.120 m^2
3	Length of condenser tube (L_c)	3.84 m

Table: 6.4 Generator design [7].

Sr. No	Parameters calculated	Value
1	Heat added in generator	342kJ/min
2	Difference between inlet and outlet temp dT	68.11°C
3	Effective Area of the generator (A_g)	0.109 m^2
4	Length of generator tube (L_g)	0.698 m

VII. CONCLUSION.

The purpose of this work is the design and analysis of solar energy driven vapour absorption cooling system. The amount of solar energy available is large which can be used without any cost. Using VARS can obtain cooling system which will save use of electricity. It can be concluded that:

- The performance of VARS is found to be 0.58 which is of 1 TR capacity.
- To get water at 84 °C from the solar collector, the total collector area required will be 24 m^2 and the no of evacuated tubes is found to be 32 with dimensions of 2.1m in length and 58mm diameter.
- Evaporator with 1 TR capacity is designed having an effective area of 0.075 m^2 and a length of 2.39m and 10mm diameter when a single evaporator tube is considered to be in use.

- Condenser for a system of 1 TR capacity is designed having an effective area of 0.1278m^2, length 3.846m and 10mm diameter. Similarly, for generator an effective area of 0.109 m^2, length 0.698m and 10mm diameter when a single generator tube is considered to be in use.

Hence feasibility of solar powered vapour absorption system has been proved. The performance vales obtained are slightly higher than the actual because of ideal heat exchange process. This unit can be used for refrigeration only during summer season because of solar potential is quite high. Investment made on systems will be high and the energy source will not remain idle during peak producing period.

REFERENCES

[1]. Abdeen Mustafa Omer" The Energy Crisis, the Role of Renewable and Global Warming" Greener Journal of EnvironmentManagement and Public Safety. ISSN: 2354-2276. Vol. 1 (1), pp. 038-070, November 2012.
[2]. The impact of the refrigeration sector on climate change, 35th informatory note on Refrigeration Technologies/November 2017.
[3]. Project proposal "Solar absorption refrigerator" Absorption Refrigeration Team Group for Environment and Energy Engineering IIT Kanpur.
[4]. V. K. Bajpai "Design of Solar Powered Vapour Absorption System". Proceedings of the World Congress on Engineering 2012 Vol. III WCE 2012, July 4 - 6, 2012, London, U.K."
[5]. V. D. Patel, "Theoretical and Experimental Evaluation of Vapour Absorption Refrigeration System" ISSN: 2248-9622 National Conference on Emerging Trends in Engineering & Technology (VNCET-30 Mar'12).
[6]. Gomri R. 2009. "Second law comparison of single effect and double effect vapour absorption refrigeration systems". Energy Conversion and Management. 5(9):1279-1287.
[7]. Arun Bangotra, "Design Analysis Of 3 TR Aqua Ammonia vapour Absorption Refrigeration System" ISSN: 2278-0181, Vol. 1 Issue 8, October - 2012
[8]. V. D. Patel, "Theoretical and Experimental Evaluation of Vapour Absorption Refrigeration System" ISSN: 2248-9622 National Conference on Emerging Trends in Engineering & Technology (VNCET-30 Mar'12).
[9]. J. Krope, "Comparison of the performances of absorption refrigeration cycles" ISSN: 1790-5044, Issue 3, Volume 4, July 2009.
[10]. Dr. S. Sankar, "Design and Performance Analysis of a Solar Thermal Assisted Vapour Absorption Air Conditioning System". ISSN (Online): 2319 – 8753, Vol. 4, Issue 4, April 2015.
[11]. C. Yap and T. K. Chang, "Effect of Vapour Absorption Enhancement in Solar-Heat Driven Air-Conditioning System", DOI: 10.7763/JOCET. 2015. V3.166, Vol. 3, No. 1, January 2015
[12]. G. Karthik, "Development of Air Conditioning System Based on Vapour Absorption Refrigeration Cycle for Automobiles Using Exhaust Gasses with R134a-Dmf" ISSN 0976 – 6359, Journal Impact Factor (2014): 7.5377.
[13]. Pongsid Srikhirin, "A Review of Absorption Refrigeration Technologies" Renewable and Sustainable Energy Reviews 5 (2001) 343–372, 12 February 2001.
[14]. R. Senthil Kumar, "Study of Vapour Absorption System Using Waste Heat in Sugar Industry". ISSN: 2250-3021 Volume 2, Issue 8 (August 2012).
[15]. K Karthik, "Design, Fabrication and Analysis of Solar Vapour Absorption Refrigeration System". ISSN 2250-2459, ISO 9001:2008 Certified Journal, Volume 4, Issue 9, September2014.