Detailed Design of Components for Libr-H_2O Vapour Absorption Refrigeration System

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Abstract— Vapor Absorption Refrigeration (VARs) is going to be one of the major devices which will be able to substitute its place in the market as it has got large potential to get easily linked with solar energy. The effective energy utilization can be done by integrating solar water heaters with solar cooling system particularly in summer. Parametric analysis provides the operating ranges, but the next is to design the various the components of the system using appropriate conditions. In this paper the detailed design of each and every component of the system is provided for the given the input parameters.

Key words: Vapour Absorption Refrigeration, Libr-H_2O, Design, Generator, Condenser, Evaporator, Absorber Heat Exchanger, ETC (Evacuated tube Collector) water heater

I. INTRODUCTION

Well today's era is energy driven for any nation of the globe, whether it may be generation or effective utilization of energy ultimately it tends to have more availability than required for consumption. So in such scenario it is been tried to utilize energy that would be wasted if not utilized in appropriate manner. By doing so, conservation of the high grade energy is done which is precious for end consumption, as it utilizes fossil fuels for its production.

Hence by accounting for waste energy recovery, definitely energy is saved which is somewhat nearly equal to twice energy produced because no extra resource is consumed for the production and also energy that is necessary is obtained by energy which had been wasted otherwise. Many research's are being done for optimum utilization of available energy and to check that energy which is rejected might be useful to some other purpose or not and result obtained are like CHP which provide heat and power both with same given input .But culprit for its quick implementation is none other than higher initial cost of installing equipments that are required to attach so as to utilize waste recovery energy.

It is been found that in Summer many of Solar water Heaters are either out of utilization or their utilized at their partial capacity. Simply it states that heat duty for water heater heating which is main purpose of water heaters by utilizing, hot water decreases demand due to various factors. Hence many a times FPC and ETC are covered by so that Solar Energy is not intercepted by them. Hence energy is wasted.

The potential for solar refrigeration is more appreciated as is noted that most countries that are poor in fossil fuels have abundant supply of solar energy and also the need for air conditioning and refrigeration is greatest when the sunshine is highest. The various methods for solar refrigeration are like (1) Vapour Compression System (2) Vapour Absorption System (3) Vapour Adsorption System (4) Thermoelectric System.

II. VAPOUR ABSORPTION SYSTEM

The vapour absorption system is viable of all the refrigeration systems for harnessing the solar energy The solar evacuated tube collector efficiency was found between 65% to 76% including all the optical losses related to transmission, absorption etc. Hence can be one of the important sources of energy that may be used to drive the solar vapour absorption refrigeration system with Libr Water cycle, particularly in summer when cooling load increases parallel along with available solar energy.

After performing parametric analysis [1], the optimum parameters for working in Bhavnagar Gujarat based on availability of the cooling water the following parameters were decided where different points on the Pressure temperature chart are as follows:

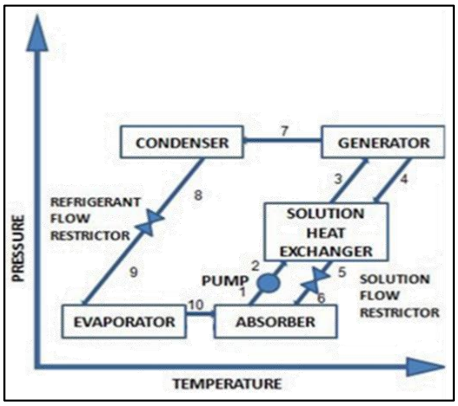

Fig. 1: VAR components on Pressure-Temperature Chart

State	Enthalpy (kJ/kg)	Mass fraction (% LiBr)	Mass Flow rate (kg/s)	Temperature(^{0}C)	Pressure (kPa)
1	53.05106	48	0.00336	30	1.4
2	53.05106	48	0.00336	30	5.63
3	70.83364	48	0.00336	38.5	5.63
4	160.5012	56	0.000294	70	5.63
5	137.6994	56	0.000294	59	5.63
6	100.4319	56	0.000294	41	1.4
7	2618	0	0.00042	65	5.63
8	146.7	0	0.00042	35	5.63
9	146.7	0	0.00042	35	1.4
10	2523.4	0	0.00042	12	1.4

Table 1: Enthalpy at different states of Vapour Absorption Refrigeration System

III. ENERGY COLLECTION BY SOLAR WATER HEATER

The present work is based on availability of the unused solar energy collected by solar water heaters of evacuated tube type in the summer season. Borosilicate glass is used for construction of the evacuated type collector which is of following dimensions:

- Mean Diameter: 47mm
- Length: 1500 mm
- Refractive Index: 1.474
- Transmittivity: 0.92

The solar evacuated type collector efficiency can be given by,

$$\eta = 0.82 - 7.884\left(\frac{T_i - T_a}{I_T}\right) \quad (1)$$

Where T_i, T_a and I_T are the temperature of water entering the collector (◦C), ambient temperature and total incident radiation on a flats surface per unit area (kJ/h m2), respectively according to F. Assilzadeh et. Al.

The collecting area of each tube is πDL that is 0.110685 m^2. There are 20 such tubes in 200 LPD Solar water heater. Hence total collecting area is 2.2 m^2.Considering average solar radiation (DNI) for month of March April and May we get 6.81kWh/m^2 day (as per NREL).

For evacuated tube collector neglecting the second term we get efficiency assuming a bit on the lower side η=0.72

So 6.81 *0.72= 4.9 units of energy/m^2 is available during the whole day. Here we have 2.2 m^2 as the collector area. So the total energy collected is 10.78kWh. That indicates in ideal situation with generator load 1.33 kW it can provide energy for 8 hours a day. The outlet temperature of evacuated type solar water heater depends on various parameters like wind speed, clearness of sky, ambient temperature etc. Considering the ideal situation for noon time from 11.00 am to 4.00 pm inlet temperature of water ranges from 75 to 77°C.

IV. DESIGN OF THE COMPONENTS

The design of the various components of the system are done for the above mentioned values of fig. 2 on the procedure chart mentioned as follows:

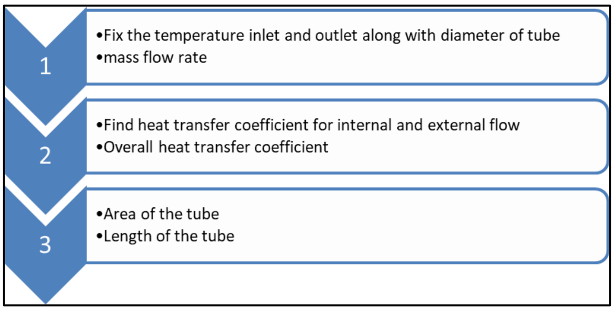

Fig. 2: Procedure chart for designing the components of the system

A. Condenser

$$Q_c = m\,(h_7 - h_8) = 1.03 \text{ kW}$$

Condenser used is water cooled horizontal shell and tube type. The copper tube with outer diameter D_o= 9.5 mm and D_i = 8.1 mm is used.

To obtain mass flow rate (m).

$$Q_c = m * C_p * (T_o - T_i) \quad (2)$$

Where, $Q_c = 1.03$ kW

$C_p = 4180$ j/kg°C $T_o = 27.5$°C $T_i = 26$°C

m = mass of water = 0.164 kg/s.

To obtain the overall heat transfer coefficient inside and external coefficient are found the property water at the mean temperature of $\frac{26+27.5}{2} = 26.75$°C and its physical properties are;

$$\rho = 996.54\frac{kg}{m^3} \quad v = 0.858 * \frac{10^{-6}m^2}{s} \quad k = \frac{0.610W}{m}°C \; Pr = 5.72$$

$$C_p = 4181 \text{ j/kg°C} \quad \mu = 0.858 * 10^{-3}\text{kg/ms}$$

$$R = 4m/D_i\pi\mu \quad (3)$$

R = 29990

$$Nu = h_i * D_i/k \quad (4)$$

For the heat transfer coefficient Nusselt number is to be found which, for turbulent flow inside the tube is expressed by well- known Dittus-Boelter correlation:

$$Nu = 0.023\, R^{0.8}Pr^{0.4} \quad (5)$$

The equation is applicable, if following condition fulfills:

Reynolds Number: 2300 < *R* < 1.2*10^5

Prandtl Number: 0.7 < *Pr* < 100

Therefore we get, $h_i = 13277$ W/m^2°C

The average of the wall and saturation temperature of the vapour of the condensate film is $\frac{35 + 26.75}{2} = 30.80$ °C and its physical properties are;

$$\rho_l = 995.36\text{kg/m}^3 \quad \rho_v = 0.0317\text{kg/m}^3 \quad h_{fg} = 2428 * 10^3\text{j/kg}$$

$$k_l = 0.616\text{W/m°C} \quad \mu_l = 783.96 * 10^{-6}\text{kg/ms}$$

$$T_w = 26.75°C \quad T_v = 35°C$$

To obtain the external heat transfer coefficient nusselt's equation is used for horizontal tube which is like

$$h_o = 0.725 * \left[\frac{g*(\rho_l)^2 h_{fg} * k_l^3}{\{\mu_l * D_o * (T_v - T_w)\}}\right]^{0.25} \quad (6)$$

Therefore we get, $h_o = 12680$ W/m^2°C

To find out log mean temperature difference (ΔTm)

$$\Delta T_m = \frac{T_o - T_i}{\ln\left[\frac{T_w - T_i}{T_w - T_o}\right]} \quad (7)$$

Hence, ΔT_m = 8.2 °C

The overall heat transfer coefficient based on the outside surface of tube is defined as:

$$U_0 = 1/\left[\left(\frac{D_o}{D_i * h_i}\right) + \left(\frac{D_o}{D_i}\right) * F_i + \left(\frac{D_o}{2*k}\right) * \ln\left(\frac{D_o}{D_i}\right) + F_o + \left(\frac{1}{h_o}\right)\right] \quad (8)$$

U_0 = 1656 W/m^2-° C is the overall heat transfer coefficient for the tube in the condensor.

To find out size of heat exchanger (condenser).

$$Q = A_0 * U_0 * \Delta T_m \quad (9)$$

By substituting above value we get, A_0 = 0.0758$m^2$the tube length L is determined $A_o = \pi D_o L$ we get L = 2.5 m

B. Evaporator

Q_e=m (h_{10}-h_9) kJ/s

Q_e= 1kW

Substituting the values we get m= 0.00042 kg/s. Here let the water inlet temperature be 26 °C and outlet temperature be 21°C then mass flow rate may be obtained

$$Q_e = m * C_p * (T_i - T_o) \quad (10)$$

Therefore we get m = 0.0478 kg/s Now further following the same procedure to find heat transfer coefficient, the property of water at the average temperature of (26+21)

/2 =23.5 ° C and its physical properties are; $\rho = 997 kg/m^3$ $v = 0.921 * 10^{-6} m^2/s$ $k = 0.604 W/m°C$ $Pr = 6.04$ $C_p = 4180 j/kg°C$ $\mu = 0.92 * 10^{-3} kg/ms$.

Following the same procedure as mentioned in condenser and solving we get $h_i = 18642\ w/m^2°C$

The average of the wall surface and water saturation temperature of the evaporator is (23.5+12)/2=17.75 ° C and its physical properties are;

$$\rho_l = 998.59 kg/m^3 \quad \mu_l = 106.08 * 10^{-6} kg/ms$$
$$k = 0.594\ W/m°C$$

$$h_0 = 0.606\ (kl) \left(\frac{\mu_l^2}{g * \rho_l^2}\right)^{-\frac{1}{3}} * \left(\frac{\Gamma}{\mu_l}\right)^{-0.22} \quad (11)$$

Therefore we get, h_0 =8447 W/m²-° C. Further $\Delta T_m = 11.3$ °C U=1618 W/m² °C

Now to get the area $Q = U * A_0 * \Delta T_m$. Therefore, we get $A_0 = 0.093\ m^2$

The tube length L is determined $A_o = \pi D_o L$ we get L = 3.12 m

C. Absorber

$$Q_a = mh_{10} + \lambda mh_6 - (1+\lambda)mh_1,\ kW$$

Where circulation ratio, $\lambda = \xi_{ws} / (\xi_{ss} - \xi_{ws})$, $\lambda = 7$

Q_a= 1.17 kW Let the water inlet temperature be 26°C and outlet temperature be 28°C then mass flow rate will be m = 0.139kg/s.

Further following the same procedure as done in condenser the Nu and hi are found. The water property at the average temperature of (26+28) /2 =27 ° C and its physical properties are;

$$\rho = 996.51\ kg/m^3$$
$$v = 0.8539 * 10^{-6} m^2/s$$
$$k = 0.610\ W/m°C$$
$$Pr = 5.82\ C_p = 4177 j/kg°C\ \mu = 0.85 * 10^{-3} kg/ms$$

$h_i = 8329\ W/m^2\ °C$

For external flow the mean properties of the solution at 41° C and 52 % LiBr are;

$$\rho = 1564\ kg/m^3\ \ k = 0.470\ W/m°C \quad Pr = 39.89\ \ C_p = 2138\ J/kg°C$$
$$\mu = 8.77 * 10^{-3} Ns/m^2$$

The solution convective heat transfer coefficient is,

The Wilke's correlation is:

$$h_s = k_s/\delta(0.29(Re_s)^{0.53}(Pr)^{0.344} \quad (4.11)$$

The film thickness is given by

$$\delta = (\frac{3\mu\Gamma}{\rho^2 g})^{1/3} \quad (4.12)$$

The solution Reynolds number $Re = 4\Gamma/\mu$

Therefore we get $h_s = 540\ W/m^2\ °C$

We get, $U = 506\ W/m^2\ °C$ and $\Delta Tm = 8.5$ °C to find size of heat exchanger (absorber)

$$Q = A_0 \Delta T_m U_0$$

Therefore, we get $A_0 = 0.273\ m^2$ the tube length L is determined $A_o = \pi D_o L$ we get L = 9.15 m

D. Solution Heat Exchanger

$$Q_{HX} = (1+\lambda)m(h_3 - h_2) = \lambda\ m(h_4 - h_5)$$
$$Q_{HX} = 59.7\ J/s$$

Let the Solution heat exchanger be single pass annulus heat exchanger.

To obtain heat transfer coefficient for the inside flow

The 56 % LiBr solution property at average temperature of (70+59)/2 = 64.5 ° C and its physical properties are obtained which are like: $k = 0.469 W/m°C$ $Pr = 14.6$

$$C_p = 1868 j/kg°C\ \ \mu = 3.67 * 10^{-3} kg/ms$$

$$R_e = \frac{4m}{\pi D\mu} \quad Nu = h_i * \frac{D_i}{k}$$

Where, $Nu = 3.66$ Hence flow is laminar

Therefore we get, $h_i = 114\ W/m^2°C$.

The 48% LiBr solution is (38.5+30)/2=34.25° C and its physical properties are;

$$k = 0.476 W/m°C\ Pr = 14\ C_p = 2039\ j/kg°C\ \ \mu = 3.21 * 10^{-3} kg/ms$$

The hydraulic diameter D_H for the annulus is the difference between the inside diameter of external tube (D_2) and the outside diameter of the internal tube (D_1).

$$D_h = D_2 - D_1 = 0.013 - 0.0095 = 0.0035\ m.$$

So, h_0 = 486.5 W/m²-° C on the basis of $N_u = h_0 D_h/k$

To find out log mean temperature difference (ΔTm) and overall heat transfer coefficient (U)

$$\Delta T_0 = 70 - 38.5 = 31.5°C,$$
$$\Delta T_L = 59 - 30 = 29°C$$

Therefore we get, $\Delta Tm = 30.23\ °C$.

U_0 = 93.63 W/m²-° C

To find out size of heat exchanger needed is $Q = A_0 * U_0 * \Delta T_m$

By substituting above value we get, $A_0 = 0.021\ m^2$

The heat exchanger tube with diameter 9.5mm and length of 0.70m is used so as to get the area of 0.021m².

E. Generator

$$Q_g = mh_7 + \lambda mh_4 - (1+\lambda)m_3,\ kW \text{ as per } (3.21)$$
$$Q_g = 1.33\ kW.$$

Further mass flow rate may be obtained as m = 0.079 kg/s.

Hence for generator experimental work done by Varma ET. al is considered which states the overall heat transfer coefficient as between 1600-7500 W/m²K, if the hot water is available at 76°C then let outlet becomes 72°C then dT = 4 °C . Hence mass flow rate would be m=0.079 kg/s. Further Q= UA LMTD

$$LMTD = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\left(\ln\frac{(T_{hi} - T_{co})}{(T_{ho} - T_{ci})}\right)}$$

T_{hi}= 76 °C T_{ho}= 72 and T_{co}= 70 and T_{ci}= 70

The LMTD 4.93 °C. The average heat transfer coefficient be the average of stated values then U=4550 W/m²K. So the area required would be 0.059 m² and length will be 1.9 m.

V. Future Scope

The experimental validation should be done so as to validate the parametrical analysis performed [1] and design mentioned here with so as to compare the practical results. Also CFD modelling and analysis must be performed so that further simulation may be done in the software which may finally lead to technology in use for common man.

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