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European Journal of Advances in Engineering and Technology, 2016, 3(11): 37-41

Research Article ISSN: 2394 - 658X

Design and Fabrication of Solar Powered Hydrogen Ammonia Absorption Refrigeration System

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ABSTRACT

The word Solar in the title of the project has key importance because our main aim to develop this project was to introduce such a refrigeration system which could utilize the solar energy instead of electricity. Besides solar energy, the system could also be run on waste heat from engines, turbines etc. As sun is the ultimate source of energy and can be found everywhere in the universe and especially Pakistan is one of those countries where there are maximum number of sunny days per annum. The rural areas where there is inadequate electricity can be easily benefitted with such a refrigeration system. We have tried our best to make the system as efficient as possible by using Titanium Nitride Oxide coating to enhance the absorbing capacity of solar collector. Styrofoam has been used to increase insulation. This system can easily be used by the tourists during camping stay far away from home. The working process is very simple with three fluid systems namely water as an absorber, ammonia as a refrigerant and hydrogen as a pressure regulating gas having its currents within the evaporator and absorber. Water after absorbing ammonia is moved to the generator with the help of a bubble pump where heat is given to the mixture from the solar collector. After heating, ammonia is separated from water because of its low boiling point and moves to the condenser where heat is extracted from it and is sent to the evaporator. In the evaporator the overall pressure of ammonia and hydrogen is divided causing faster evaporation of ammonia in the evaporator by achieving low boiling point and causing cooling.

The work illustrated deals with the design, fabrication and testing of the aforementioned concept. The system was operated on a sunny day, the collector fluid temperature and cabin temperatures were monitored. The obtained results have been tabulated and the C.O.P and tonnage of the refrigerator were also computed.

Key words: Generator, styrofoam, C.O.P, bubble pump, refrigeration

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INTRODUCTION

In 1858 a French scientist named Ferdinand Carré invented an absorption cooling system using water and sulphuric acid. In 1922 Baltzar Platen and Carl Munters, improved the system in principle with a 3-fluid configuration. This 'Platen-Munters' design was capable of operating without a pump [6]. During 1926-1933 Einstein and Szilárd joined hands to improve the technology in the area of domestic refrigeration. The two were inspired by newspaper reports of the death of a Berlin family due to seal failure which caused a leakage of toxic fumes into their home. Einstein and Szilárd proposed a device without moving parts that would eliminate the potential for seal failure, and worked on its practical applications for various refrigeration cycles. The two in due course were granted 45 patents in their names for three different models. At atmospheric conditions, ammonia is a gas with a boiling point of - 33°C. The system is pressurized to the point where the liquefaction of ammonia occurs. The cycle is closed, with hydrogen, water and ammonia and recycled perpetually. The cooling cycle starts with high pressure liquefied ammonia entering the evaporator at ambient temperature. The partial pressure of the hydrogen gas affects the boiling point of the ammonia. As ammonia boils in the evaporator, it requires energy to overcome the enthalpy of vaporization. This energy is drawn from the refrigerator cabin providing the desired cooling effect [1]. The influence of operating conditions such as: ammonia fraction in inlet solution and tube diameter on the functioning of the bubble pump was presented and discussed by Ali *et al* [2]. It was found that, the liquid velocity and pumping ratio increase with increasing heat flux, and then it decreases. Optimal heat flux depends namely on tube diameter variations. Ali *et al* [5] found that the optimum heat flux can be correlated as a function of the mass flow rate and tube diameter, while the minimum heat flux required for pumping can be correlated as a function of tube diameter. The effects of hot water inlet temperatures on the coefficient of performance (COP) have been studied by Mittal *et al* [3]. Gabsi *et al* [4] designed and simulated an absorption diffusion refrigerator using solar energy for domestic use. The climatic conditions and the cost due to technical constraints for various components such as the solar generator, the condenser, the absorber and the evaporator limits the system's application for small scale purposes. Mass and energy conservation equations showed that the new absorption cycle could produce viable amounts of cooling for domestic applications.

DESIGN OF SOLAR COLLECTOR

Calculation of Collector Area

Here the collector used is parabolic in shape.

Taking maximum temperature at the generator, $T_G = 90^0C$

Solar constant (Isc) = 1353W/m^2, Extra-terrestrial radiation (Io) = 1398 W/m^2

Geographical location of the place where the solar collector was placed: (Faisalabad, Pakistan)

Latitude = 31.4187° N and Longitude = 73.0791° E

Also the month of operation is assumed to be march ($\delta = 0$; $t = 0$)

Direct radiation reaching the surface at earth is a function of time of the day, latitude location and declination angle

Let Z-Zenith angle

The zenith angle is calculated thus, $\text{Cos } z = \sin\gamma\sin\delta + \cos\gamma\cos\delta\cos t$

Where, γ- latitude of location, δ -declination angle, t – hour angle

i.e. $\text{Cos } z = \sin\gamma\sin\delta + \cos\gamma\cos\delta \cos t$, Cos z = sin(9.733421) sin (0) + cos (9.733421) cos (0) cos(0)

Cos z =0.169064818 x 0 + 0.985605 x 1 x1, Cos z =0.985605, Z = cos-1(0.0985605)

Intensity of solar radiation,

$I_z = I_{sc}$ e(-c(sec z)s),

I_z =1353 e (-0.357(1/0.985605)0.678) =1353e (-0.2455)

I_z =1058.4698 W/m^2

The value of radiation on a horizontal surface (I_h) is

$I_h = I_z \cos z$, I_h =1058.4698 x 0.985605 =1043.233 W/ m^2

Available radiation intensity =1043.233 W/m^2

Assume, 50% efficiency due to: - 1. Variance 2. Collector efficiency and 3. Manual tracking system

This implies solar radiation intensity=500W/m^2 (approx.)

Now,

Reflected intensity (R_i) = 0.9x500 = 450 W/m^2 (Reflectivity of chromium coated aluminium = 0.9)

Then, Heat required at collector box (Q_i),Q_i =5 x 4.18 x (90-30)/3600, Qi = 0.348 kW = 348W

Area of parabolic dish (A_d) = 348/450

$A_d = 0.77m^2$, Take depth (h). h=0.25m

Surface area (A_s) , $A_s = \pi/6[r/h^2] [(r^2+4h^2)^3/2-r^3]$,$0.77 = \pi/6[r/0.252] [r^3] \{[1+(0.5/r)^2]^3/2-1\}$

By trial and error; r=0.45m, D=0.90m, Focal length (F)=$r^2/4h = 0.45^2/4x0.25$ =0.2025m

Ideal operating temperature required at generator=85^0C-90^0C

Condenser pressure=10bar and Evaporator pressure=1bar

Concentration of aqua-ammonia=17moles/l, Quantity of aqua-ammonia filled=300ml

Pump selection, Type=MONOBLOCK

Head=15m, Flow rate (Q), Q=250liter/hour, Q=0.25m3/3600sec = 0.697x10-4m^3/sec

CALCULATION

A. Heat Transfer at Generator

L=0.25m, Di=0.02m, Do=0.04m, Dm=Do-Di=0.02m ,Ts =30^0c ,Tw =90^0c

$T_f = T_w + T_s/2 = 90^0C + 30^0C/2 = 60^0C$, At $T_f = 60^0C$, μ =0.548m/s ,v = 0.478x10-6m^2/s

P_r = 3.02 ,K = 0.6513 W/mK ,$R_e = \mu Dm/v$ = 0.578x0.02/0.478x10-6m^2/s

R_e =22,928.87

Now, since the flow is forced convection in a concentric annulus duct. (Note: Dittus-boelter is less accurate for D_i/D_o > 0.2) ,Using Monrad and Pelton's equation(1942) ,$N_u = 0.02(Re^{0.8}) (Pr)^3 [D_o/D_i]^{0.53}$

N_u =0.02 x {22,928.87]$^{0.8}$ x [3.02]$^{0.33}$ x [0.04/0.02]$^{0.53}$,N_u =128.02 ,h_l = N_u x k/D_m ,h_l =128.02x0.6513/0.02
h_l =4169.03W/m^{2}K , Q_G = $h_1\bar{}$A(T_∞-T_s) = 4169.03 x (3.14 x 0.02 x 0.25) x (90 – 85) = 327.27 W

B. Temperature of Refrigerant at Generator Exit

Since the bubble pump is a capillary tube of very thin material. Assume the wall/surface temperature of bubble pump= temperature of NH_4OH inside it. Now, at steady state, $T_{solution}$=85^{0}C
We also know that miscibility of NH_3 reduces with increasing temperature. Hence NH_3 vapours (refrigerant) are generated at 85^{0}C.

Heat Rejected at Condenser, Qc

Assumption: We will assume that NH_3 vapour condenses at 25^{0}C and the liquid NH_3 generated exits at the same temperature (ie;25^{0}C). It is not cooled further.
Note: The above assumption was made as H_2 gas was filled to keep the system pressure at 10bar while charging the system and we know that at 10 bar, the saturation temperature of NH_3 is 25^{0}C.
Hci (85^{0}C& 10bar) = 1650 KJ/Kg, H_{co} (25^{0}C & 10 bar) = 1484 KJ/kg,
To calculate heat transfer coefficient, for finned section. For rectangular blocks, Temp at fin =100^{0}C, Air temp = 40^{0}C, Film temp = 70^{0}C, Lv=7 cm
L =LHLV/LH+LV, L =2.92 cm, At 70^{0}C air, V =20.02x10-6m^2/s, k =0.02966 W/mK, Pr=0.694, β=1/343
R_a=G_r x P_r , R_a =β g (T_s-T∞)/v^2 x L^3 x P_r ,R_a =9.81/343 x (100-40)/{(20.02x10^{-6})^2} x (2.92x10^{-2})^3 x 0.694
R_a =7.39x10^4 ,N_u =0.55 Ral ¼ ,Nu =0.55 x (7.39x10^4)$^{1/4}$
N_u =9.07 ,h1 = N_u x k/L + 9.07 x (29.66x10^{-3})/2.92x10^{-2} ,h_1=9.212 W/m^{2}K
h_1 = N_u x k/L + 9.07 x (29.66x10^{-3})/2.92x10^{-2} ,h_1=9.212 W/m^{2}K .For unfinned section ,Condenser length = 32cm
Diameter =1.4cm, Unfinned section length = 320 - (40x1) = 280mm
Taking it as a whole cylinder. At T_f=70^{0}C ,V = 20.02x10^{-6} m^2/s ,k = 0.02966 W/mK ,P_r =0.694 ,β =1/343 K^{-1}
R_a = G_r x P_r ,R_a =βg (Ts-T∞)/v^2 x D^3 x Pr ,Ra = 9.81x(100-40)(0.014)^{3}x0.694/343x(20.02x10^{-6})^2
R_a =8153.453 ,N_u =C(Ra)^n ,C =0.85; n =0.188 ,N_u = 0.85(8153.453)$^{0.188}$,N_u = 4.62 ,h_2 = N_u x k/D
h_2 = 4.62 x 29.66 x10^{-3}/0.014, h_2= 9.7878 W/m^{2}K, Q_{total}(condensation) = [h_1 x A_1 x 40 x (T_s-T_∞)] + [h_2 x A_2 x (T_s-T_∞)], Q $_{total}$(condensation) = [9.212 x 0.07 x 0.05 x 40] + [9.7878 x π x 0.014 x 0.28] (100-40), Qc = 84.61W

Mass Flow Rate

At condenser; ṁR (h_{ci}-h_{co}) = Q_c ,ṁR (1650-1484) x 10^3 = 84.61 ,ṁ R = 5.096x10^{-4} Kg/sec ,ṁR = 0.0306Kg/min

Heat Absorbed at Evaporator, Qe

Assuming laminar flow around all surfaces, we can use the following simplified equations for air: Vertical Plate
hv = 1.42 x (ΔT/L)$^{1/4}$,Now, Outside surface temp = 250^{0}C ,Inside surface temp = 100^{0}C ,Ambient temp = 400^{0}C
Cabin temp. = 80^{0}C ,h_{va} = 1.42 (40 – 25/ 0.55)1/4 = 3.24 W/m^{2}K ,h_{vb} = 1.42 (10 – 8/ 0.55)1/4 = 1.96 W/m^{2}K
Horizontal Plate, hh = 1.32 x (ΔT/L)1/4 ,h_{ha}= 1.32 (40 – 22 / 0.45)$^{1/4}$ = 3.32 W/m^{2}K ,h_{hb}= 1.32 (9 – 8 / 0.45)$^{1/4}$ = 1.61 W/m^{2}K
Q_E Calculation Front & Back
Q_{E1} = 2 x T_1 – T_2 / ((1/h_{va}) + (L_1/k_1 A) + (L_2/k_2 A) + (1/h_{vb})) ,Q_{E1} = 2 x (40 – 8) / (1/3.24) + (0.02/0.035 x 0.4 x 0.55)+(0.002/205 x 0.4 x 0.55) + (1/1.96) ,Q_{E1} = 18.74 W Sides.
Similarly,
Q_{E2}=2x(40–8)/(1/3.24)+ (0.02/0.035x0.45x0.55)+(0.002/205x0.45x0.55)+ (1/1.96) ,Q_{E2} = 20.46 W
Top & Bottom
Q_{E3}=2x(40–8)/(1/3.32)+ (0.02/0.035x0.4x0.45)+ (0.002/205x0.4x0.45) + (1/1.61) ,Q_{E3}= 15.62 W
Q_E = Q_{E1} + Q_{E2}+Q_{E3} = 18.74 + 20.46 + 15.62 = 54.82 W

Tonnage of Refrigerator

Let TCL = Total Cooling Load, Then, TCL = Q_E + Q_{inf} ,Heat due to infiltration, Q_{inf}= q x (1/v_1) x (h_1-h_2)
Now,
q = (L x W x H x Ac) / 60 (m^3/min), A_C = A_{C1} + A_{C2} = 0.5 + 2 = 2.5 ($A_{C1}$from table for air changes by Ashrae (1972) for room with no windows or doors & A_{C2} = 2 times /hr doors are opened to check temp.)
q = 0.4 x 0.45 x 0.55 x 2.5 / 60 = 4.125 x 10^{-3} m^3/min.
Outside conditions:

At 300^0C & RH = 60%

h_1 = 71 KJ/Kg ,v_1 = 0.88 m^3/kg , Inside cabin conditions:Assume 60% RH for inside cabin as well.

At 80C & 60% RH ,h_2 = 16.5 KJ/Kg

Now, $Q_{inf} = 4.125 \times 10^{-3} \times (1/ 0.88) \times (71 – 16.5) = 0.25546$ KJ/min = 4.25 W,

TCL = $Q_E + Q_{inf}$ = 54.82 + 4.25 = 59.07 W

Now 1 TR = 3.517 KW, So, Tonnage of refrigerator = TCL/ (3.517 x 1000) = 0.0168 TR

Coefficient of Performance (COP)

COP = QE / QG

COP= 54.82 / 327.27 = 0.1675

RESULTS AND DISCUSSION

After the total assembly and calculations were complete the setup was tested. The testing was performed from 9:00 am to 4:00 pm and the reading was noted. Every half an hour the parabolic dish was adjusted manually to track the movement of the sun. From the testing done it was noted that the lowest temperature achieved was 8^0C. It was noted that the cabin temperature increased for a certain period and then dropped. The C.O.P of the system was obtained from the calculations as 0.1675 for a mini fridge of 40 liters. The tonnage of the system for the test conditions was 0.0168TR and the mass flow rate of the refrigerant obtained was 0.0306 kg/min. It may be combined or kept separate and may be further divided into subsections. It should deal with the major findings and their explanation/interpretations.

CONCLUSION

The setup was successfully made and the testing was done and vapour absorption system was successfully run using hot water as source of heat obtained from a solar collector of area 0. 64m^2.The mechanical modifications made to the mini fridge to accommodate the vapour absorption system and the design of the solar collector were successful and a lowest cabin temperature of 8^0C was achieved for a mini fridge of 40L capacity. The testing conditions were a sunny day and the duration of testing was 9 hours. The C.O.P obtained was 0.1675. The tonnage of the system was computed as 0.0164TR. The mass flow rate calculated was 0.0306 kg/min. The prognosis for the future of solar refrigeration and air conditioning seems to be very good and no doubt it will find its place in future industrial applications. But there are a few drawbacks with the system which need to be addressed. The major limiting factor at present is the availability of solar energy whenever it is required, for example at nights and extended cloudy days we cannot attain a high enough temperature and hence refrigeration is poor. Modifying the design of solar collector for wider acceptance angle and making generator tubes with material of higher thermal conductivity yield can be improved.

Table -1 Variance of Collector Fluid Temperature with Local Time

Local time in (Hours)	Ambient Temperature in (C°)	Collector fluid temperature in (C°)
9:00	27	40
10:00	29	45
11:00	32	55
12:00	33	65
13:00	34	85
14:00	35	90
15:00	36	90
16:00	33	88

Table -2 Variance of Cabin Temperature with Local Time

Local time in hours	Cabin temperature in C°
9:00	34
10:00	26
11:00	24
12:00	20
13:00	14
14:00	8
15:00	12
16:00	15

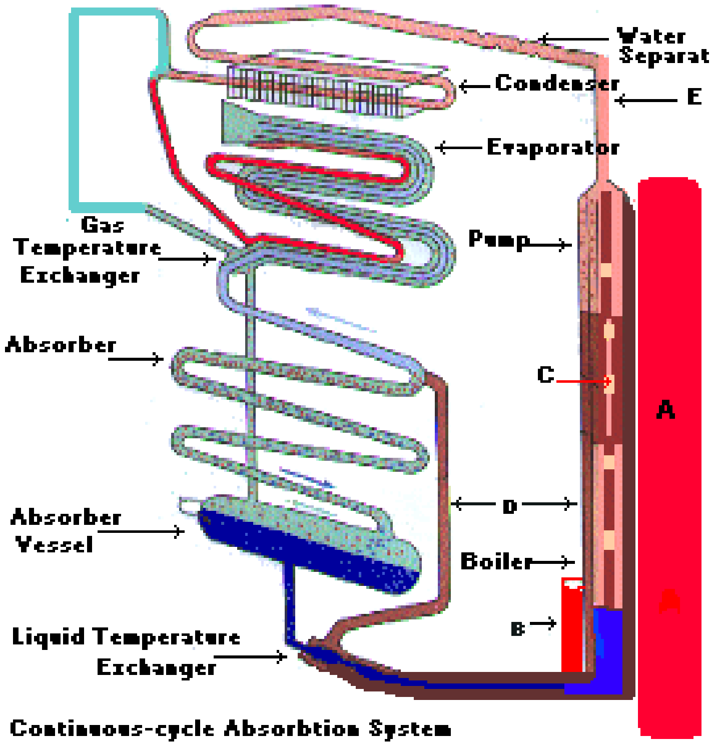

Fig.1 Illustration of A Vapor Absorption System

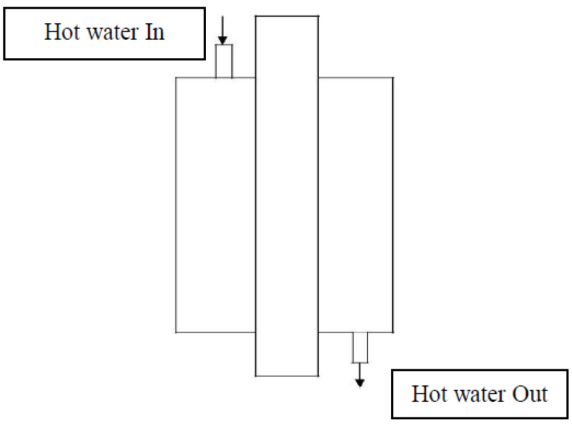

Fig.2 Schematic Diagram of Heat Exchanger Around Bubble Pump

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