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By Ghassan Al-hassan, Naif K. Al Shammari & Mohammed AL-Meshaal
*Al Imam Mohammad Ibn Saud Islamic University (IMSIU), Saudi Arabia*

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DEVELOPMENTOFANABSORPTIONREFRIGERATIONSYSTEMPOWEREDBYSOLARENERGYTOEXTRACTFRESHWATERFROMHUMIDAIRINSAUDIARABIA

# Development of an Absorption Refrigeration System Powered by Solar Energy to Extract Fresh Water from Humid Air in Saudi Arabia

Ghassan Al-hassan [α], Naif K. Al Shammari [σ] & Mohammed AL-Meshaal [ρ]

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## I. Introduction

The next challenge to human life in the world is how to solve the future problems that appeared in the world. The most three critical problems that face human life are energy crisis, water crisis and pollution [1]. Fresh water supply and sustainable energy sources are of the most important topics on the international environment and development plans. They are also, critical factors that govern the lives of humanity and promote civilization. The history of mankind proves that water and civilization are two inseparable entities. This is proved by the fact that all great civilizations were developed and flourished near large sources of water. Rivers, seas, oases and oceans have attracted mankind to their coasts because water is the source of life [2].

*Author α: Department of Mechanical Eng., Al Imam Mohammad Ibn Saud Islamic University (IMSIU), Riyadh, Saudi Arabia.*
*e-mails: gaalhassan@imamu.edu.sa, maalmeshaal@imamu.edu.sa*
*Author σ ρ: Department of Mechanical Eng., University of Hail, Hail, Saudi Arabia. e-mail: naif.alshammari@UoH.edu.sa*

Saudi Arabia is facing a water scarcity due to the prevailing weather conditions especially for remote areas caused to over population, industrialization and agricultural expansion. The problem of providing remote areas with fresh water can be solved by using three techniques [3],

a) Transportation of water from other location.
b) Desalination of saline water (ground, or underground.
c) Extraction of water from atmospheric air.

Water transportation from other locations is usually expensive and of high initial cost to those remote areas. The desalination of saline water (ground and underground) is also expensive, high initial cost and related to water existence in zone.

Atmospheric air is a huge and renewable reservoir of water. This endless source of water is available everywhere on the earth surface. The amount of water in atmospheric air is evaluated as 14000 $Km^3$ , where as the amount of fresh water in rivers and lakes on the earth surface is only about 1200 $Km^3$ [4].

The extraction of water from atmospheric air can be accomplished by different methods, the most common of these methods are cooling moist air to a temperature lower than the air dew point [5], and absorbing water vapor from moist air using a solid or a liquid desiccant [6, 7].Choice of methods is an engineering decision dependent on local climatic conditions and economic factors such as capital, operating, and energy costs.

The first major project on an all solar absorption refrigeration system was undertaken by Trombe and Foex (1964)[8]. Ammonia-water solution is allowed to flow from a cold reservoir through a pipe placed at the focal line of a cylinder-parabolic reflector. Heated ammonia-water vaporized in the boiler is subsequently condensed in a cooling coil. The evaporator is a coil surrounding the container used as an ice box. In the prototype trials, the daily production of ice was about 6 to 4 kilograms of ice per square meter of collecting area for four-hour heating. The design by Trombe and Foex is very promising and should be studied further although modifications may be necessary on the solar collector, boiler, and condenser.

Farber (1970) [9] has built the most successful solar refrigeration system to date. It was a compact solar ice maker using a flat-plate collector as the energy source. It was reported that an average of about 42,200 kJ of solar energy was collected by the collector per day and ice produced was about 18.1 kilograms. This gave an overall coefficient of performance of about 0.1 and 12.5 kilograms of ice per $m^2$ of collector surface per day.

Swartman and Swaminathan (1971) [10] built a simple, intermittent refrigeration system incorporating the generator-absorber with a 1.4 $m^2$ flat-plate collector. Ammonia water solutions of concentration varying from 58 to 70 percent were tested. Tests were relatively successful; evaporator temperatures were as low as -12°C, but due to poor absorption, the evaporation rate of ammonia in the evaporator was low. Staicovici [11] made an intermittent single-stage H2O-NH3 solar absorption system of 46 MJ/cycle (1986). Solar collectors heat the generator. Installation details and experimental results were presented. The system coefficient of performance (COP) varied between 0.152 and 0.09 in the period of May–September. Solar radiation availability and the theoretical (COP), also applicable to the Trombe-Foex system, were assessed. Reference was made to evacuated solar collectors with selective surfaces. Actual (COP) system values of 0.25–0.30 can be achieved at generation and condensation temperatures of 80°C and 24.3°C respectively. In 1990 Sierra, Best and Holland [12] made a laboratory mordent of absorption refrigeration. Using ammonia - water solution at 52% concentration by weight and the total weigh is 38 kg. this system was operated intermittently using this heat source .A heat source at temperature no higher than 80 °C was used to simulate the heat input to an absorption refrigeration from solar pond. In this system the temperatures of generator was as high as 73°C and evaporator temperatures as low as -2 °C. Tap water was used to remove the heat generated from the condensation of the ammonia vapor and the absorption of the refrigerant in the water. The temperature of the tap water was near the ambient laboratory temperature of 28°C. The COP for this unit working under such condition was in the range 0.24 to 0.28. In 2000 Hammad and Habali [13] made a steel sheet cabinet of 0.6 m x 0.3 m face area and 0.5 m depth. The cabinet was intended to store vaccine in the remote desert area, away from the electrical national grid. A solar energy powered absorption refrigeration cycle using Aqua Ammonia solution was designed to keep this cabinet in the range of required temperatures. The ambient temperatures reached about 45°C in August. A computer simulation procedure was developed to study the performance and characteristics of the cooling cycle. The simulation included MATLAB computer programs for calculation the absorption cycle. In this system using a cylindrical solar concentrator extended the daily operating time to about 7 h and increased the output temperature up to 200 °C and the range of the COP was between 0.5 to 0.65 .While the temperature which gives optimum condition (of COP =0.65) was 120 °C. For this study a solar absorption refrigeration unit was constructed. The working fluids employed was aqueous ammonia (25wt% NH3 – H2O). The system was operated during the months of June and July (2009) for a period of 8 hours per day (8 am to 4 pm).

## II. Ammonia Absorption System

The absorption cycle is a process by which refrigeration effect is produced through the use of two fluids and some quantity of heat input, rather than electrical input as in the more familiar vapor compression cycle [14]. Both vapor compression and absorption refrigeration cycle accomplish the removal of heat through the evaporation of refrigerant at a low pressure and the rejection of heat through the condensation of the refrigerant at a higher pressure. The method of creating the pressure difference and circulating the refrigerant is the primary difference between the two cycles. The vapor compression cycle employs a mechanical compressor to create the pressure differences necessary to circulate the refrigerant. In the absorption system, a secondary fluid or absorbent issued to circulate the refrigerant. Because the temperature requirements for the cycle fall into the low – to – moderate temperature range, and there is significant potential for electrical energy savings, absorption would seem to be a good prospect for cooling application [15].

Thus, the purpose of this paper is to utilizing an absorption refrigeration system driven by solar energy in order to generate a cold surface with which extract water from moisture air passing though the generated cold surface.

## III. Cycle Description

Figure 1 illustrates the main components of the absorption refrigeration cycle and the humid air tunnel. High-pressure liquid refrigerant from the condenser (2) passes into the evaporator (4) through an expansion valve (3) that reduces the pressure of the refrigerant to the low pressure existing in the evaporator, which is located in the humid air tunnel and where the process of water separation from the moisture air stream begins due crossing the cooling coil and reaching a temperature beyond its dew-point temperature. The liquid refrigerant evaporates in the evaporator by absorbing heat from the humid air being cooled and the resulting low-pressure vapor of ammonia passes to the absorber, where it is absorbed by the week solution coming from the generator (1) through an expansion valve (5), and forms the strong solution. The strong solution is pumped to the generator pressure, and the refrigerant in it is boiled off in the generator due of solar energy. The remaining solution flows back to the absorber and, thus, completes the cycle. By weak solution is meant that the ability of the solution to absorb the refrigerant vapor is according to the ASHRAE definition [5]. In order to improve system performance, a solution heat exchanger is included in the cycle. I order to remove water vapor from the refrigerant mixture leaving the generator before reaching the condenser a rectifier is added into the cycle. For the current study, it is assumed that the refrigerant vapor contains 100% ammonia for compatibility of the results for NH3-H20 cycle.

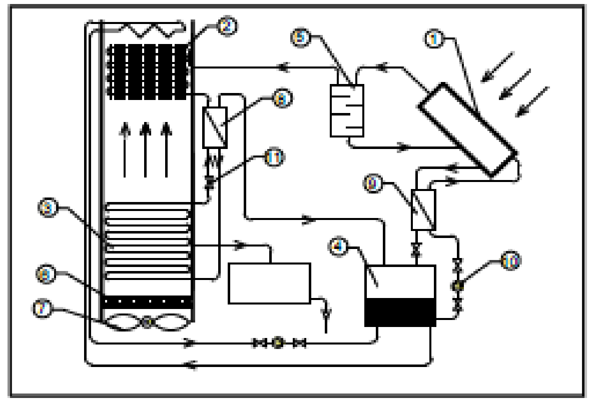

*Fig. 1 :* Schematic diagram of the Solar–Absorption refrigeration system for water extraction from humid air

## IV. Design of the Major Components of the Absorption Refrigeration System

a) *Design of the Generator and collector*

i. *The cooling capacity of the system (the heat absorption from the humid air passing within the tunnel by the R717 during the evaporation process)*

$$Q_o^{\bullet} = Q_a^{\bullet} = m_a^{\bullet}.C_{pa}.\Delta T_a$$

$$Q_o^{\bullet} = Q_a^{\bullet} = 50X1.2X1.002X(40-10) = 1803.6Kj/h = 501W$$

ii. *Determining the flow rate of ammonia in the evaporator. The refrigeration capacity of the system is found out from following formula;*

$$Q_o^{\bullet} = m_R^{\bullet}.\Delta h$$

And by using the P-h diagram of R717 and at $P_C = 17.85bar$ and $P_o = 175bar$, then $\Delta h = 1000kj/kg$, thus $m_R^{\bullet} = 0.000501kg/s$, Therefore the whole amount of the refrigerant running in the cycle (assuming that the original mass of the solutions are taken 10% more) is: $M_R^{\bullet} = 1.15Xm_R^{\bullet} = 0.000575kg/s$.

iii. *The flow rate of weak solution leaving the generator-collector (considering that 90% of the ammonia leaves the collector toward the condenser and density difference) is*

$M_{WE}^{\bullet} = 0.0029 + 0.1XM_R^{\bullet} = 0.0039$ $kg/s$, and the flow rate of strong solution running between the generator and absorber is:

$$M_{ST}^{\bullet} = M_R^{\bullet} + W_W^{\bullet} = 0.000575 + 0.0039 = 0.00448kg/s$$

iv. *Determination of the volume rate of R717 vapor flows through the generator*

$V_R^{\bullet} = M_R^{\bullet}Xv_R$ where $v_R = 0.0726m^3/s$ the specific volume of the ammonia vapor in the generator at operation pressure ($P_G = P_C = 17.82bar$), therefore $V_R^{\bullet} = 0.000042m^3/s$.

v. *Calculate the volume rate of $H_20$ vapor flows through the generator (considering 2.5% of water will evaporate) and the specific volume of water vapor (in the generator at $P_G = P_C = 17.82bar$) is* $v_{WV} = 0.111667m^3/kg$, *thus*

$$V_{WV}^{\bullet} = \frac{2.5}{100}M_R^{\bullet}Xv_{WV} = \frac{2.5}{100}0.000575X0.111667 = 0.000002m^3/s$$

vi. Determine the total volume rate of the strong solution entering the generator considering that the specific volume of the strong solution cycling through the generator at the operation pressure ($P_G = P_C = 17.82bar$) is $v_{ST} = 0.0726m^3/kg$

$$V_{ST}^{\bullet} = M_{ST}^{\bullet}Xv_{ST} = 0.00033m^3/s$$

Total volume of the generator = volume of R717 vapor + volume of $H_2O$ vapor+ volume of strong solution, $V_G = 0.000374 m^3/s = 374 cm^3/s = 22440 cm^3/\min$ considering that the volume of the generator is more 15% bigger than the calculated value, thus the required volume of the generator and collector is $V_G = 25806 cm^3$, therefore the sizes of the generator and collector are $100 X 50 X 5 cm^3$.

*b) Design of the air cooled condenser*

An air cooled finned condenser is used for domestic refrigerators and considering that it employs as a counter flow type heat exchanger for heat transfer with following design factors:

- Heat transfer and pressure drop characteristics
- Working pressure
- Type of fluid
- Manufacturing ease and service

Temperature of R717 vapor at inlet $T_{Ri} = 80^oC$,

Temperature of R717 liquid at outlet $T_{Ro} = 45^oC$,

Assuming that the humid air passing through the evaporator and after lossing part of its humidity will be used to accomplish two functions, first absorbs the generated mixing heat (mixing the saturated ammonia vapor coming from the evaporator and the weak solution returned from the collector) formed in the absorber and second also absorbs rejected heat from the condenser, therefore, assuming that the temperature of cooled air stream entering the condenser is 25 $^oC$.

Inlet temperature of ambient air $T_{ai} = 25^oC$,

Outlet temperature of ambient air $T_{ao} = 27^oC$,

Thus, $\Delta T_1 = 80 - 25 = 55^oC$, and $\Delta T_2 = 45 - 27 = 18^oC$, then the logarithmic mean temperature difference (as counter flow heat exchanger) is $\Delta T_{Lm} = \dfrac{\Delta T_1 - \Delta T_2}{Ln(\dfrac{\Delta T_1}{\Delta T_2})}$,

so that $\Delta T_{Lm} = 33.13^oC$. Using following equation to find out the overall heat transfer coefficient between NH3 and ambient:

$U = \dfrac{1}{\dfrac{1}{h_o} + \dfrac{d_o}{k}[(\dfrac{d_0 - d_i}{d_0 + d_i})] + \dfrac{1}{h_i} X \dfrac{d_o}{d_i}}$, so it is required to determine the values of heat transfer coefficients by convection, namely, $h_o$ and $h_i$. First considering that $d_0 = 0.012m$ and $d_i = 0.01m$ for the stainless steel condenser coil and the properties of ambient air at $40^oC$ are: $\rho = 1 kg/m^3$, $k = 0.648 W/m^3.K$, $v = 0.498 X 10^{-6} m^2/s$ and $P_r = 3.58$ [16]. For natural convection, it is assumed that $h_o = 17 W/m^2.K$.

Heat transfer coefficient by convection for vapor flowing inside a metallic tube with decrease in temperature and film condensation inside horizontal tube is given by following equation [17]:

$$h_i = 0.555 \left[\frac{{k_{avg}}^3 \times \rho_L \times (\rho_L - \rho_v) \times g \times h_{fg}}{\mu \times (t_s - t) \times d_i}\right]^{\frac{1}{4}}$$

Where

$k = thermal\ conductivity\ of\ R717\ Refreigerant = 0.232645\ w/m.k$

$\rho_L = Density\ of\ liquid\ ammonia\ at\ 17.82\ bar = 571.3\ kg/m^3$

$\rho_v = Density\ of\ vapor\ ammonia\ at\ 17.82\ bar = 13.8\ kg/m^3$

$h_{fg} = Enthalpy\ of\ super\ heated\ vapor\ at\ 17.82\ bar = 1075\ kj/kg$

$\mu = Dynamic\ viscosity\ at\ 17.82\ bar = 0.0001094\ kg/m.s$

$g = Acceleration\ due\ to\ gravity = 9.81\ m^2/s$

By substituting the given values

$$h_i = 0.555 \left[\frac{(0.232645)^3 \times 571.3 \times (571.3 - 13.8) \times 9.81 \times 1075}{0.0001094 \times (45 - 25) \times 0.01}\right]^{\frac{1}{4}}$$

$$= 1356.75\ w/m^2.k$$

Thus, the U value is

$$U = \frac{1}{\frac{1}{18} + \frac{0.012}{50.2} \times \left[\frac{0.012 - 0.01}{0.012 + 0.01}\right] + \frac{1}{925.5} \times \frac{0.012}{0.01}} = 17.71\ w/m^2.k$$

The heat removed from the refrigerant in the condenser is

$$Q_c = \dot{m}_R(h_2 - h_3) = 0.00198(1570 - 396.8) = 2.32\ kw = 2320\ W$$

Therefore

$$Q_c = U \times A_c \times LMTD$$

$A_c = \frac{Q_c}{U \times LMTD} = \frac{2320}{17.71 \times 33.13} = 3.95\ m^2$ , by considering that the tube area of the condenser forms 25% from the total area on the air cooled condenser.

$$A_c = \pi d_o L$$

$$L = \frac{A_c}{\pi d_o} = \frac{3.95X0.33}{\pi \times 0.012} = 34.6\ m$$

c) *Design of the Evaporator*

An evaporator should transfer enough heat with smaller size as possible. It should be light, compact, safe and durable. The pressure loss in the evaporator should be as low as possible as well. Some design factors should be considered:

- Evaporator temperature.
- Refrigerant properties.
- Refrigerant effect.
- Tube selection.

Heat absorbed by NH3 vapor= 1803 W. The refrigerant R717 evaporates due to absorbing heat from the humid airstream entering the tunnel and passes through the cooling coil.

i. *Determination of $\Delta T_{Lm}$ (Assuming counter flow heat exchanger)*

Temperature of air entering the evaporator, $T_{ei} = 40^oC$

Temperature of air leaving the cooling coil after few seconds $T_{eo} = 15^oC$ .

Temperature of NH3 liquid entering evaporator $T_{Ri} = -18^oC$ ,

Temperature of NH3 liquid leaving evaporator $T_{Ro} = -18^oC$ ,

Thus $\Delta T_{Lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\frac{\Delta T_1}{\Delta T_2}}$ , $\Delta T_1 = 40 - (-18) = 58^oC$ ,

and $\Delta T_1 = 15 - (-18) = 33^oC$ .

Therefore, $\Delta T_{Lm} = 14.23^oC$

ii. *Determination of $U$ (The overall heat transfer coefficient between the NH3 in cooling coil and surrounding humid hot air).*

$$U = \frac{1}{\frac{1}{h_{oa}} + \frac{x_t}{K_t} + \frac{1}{h_{ia}}}$$

Assuming: $h_{oa} = 35\frac{W}{m^2.K}$ , $x_t = 1mm$ Stainless tub thickness, $K_t = 50.2\frac{W}{m.K}$ and Also consider that $h_{ia} = h_{i_{condenser}} = 1356.75\frac{W}{m^2.K}$

Therefore:

$$U = \frac{1}{\frac{1}{35} + \frac{0.001}{50.2} + \frac{1}{1356.75}} = 34.09W/m^2K$$

Finding out $\dot{Q}_w$ (heat transfer rate between NH3 and surrounded humid air stream in the tunnel)

$$30661 = 33.6XA_e \times 14.23$$

$$A_e = 64.13\ m^2$$

$$L = \frac{2A_e}{2\pi d_0} = \frac{2 \times 64.13}{2\pi \times 0.012} = 1701\ m$$

d) *Design of the absorber*

Determination of the volume of the absorber.

$V_{Ab}$ = [Volume of NH3 vapor] + [volume of weak NH3+H2O solution]

Temperature of NH3 vapor leaving the evaporator is -30° C

Density of NH3 vapor $\rho_v = 0.701\ kg/m^3$

Density of weak solution $\rho_{Lw} = 596\ kg/m^3$

$$V_{Ab} = \left[\left(\frac{1}{\rho_v} \times \dot{m}_r\right) + \left(\frac{1}{\rho_{Lw}} + \dot{m}_{ws}\right)\right]$$

$$V_{Ab} = \left[\left(\frac{1}{0.701} \times 0.11574\right) + \left(\frac{1}{596} + 0.1794\right)\right] = 0.346\ \frac{m^3}{min} = 346{,}000\ \frac{cm^3}{min}$$

Thus,

Volume of the absorber $= 75 \times 75 \times 62\ cm$

e) *Design of the solution pump*

Quantity of rich solution to be supplied to the generator (Discharge)

$= \text{Mass flow rate} \times specific\ volume$

$$= \frac{0.29514}{60} \times \frac{1}{\rho_{rs}} = \frac{0.29514}{60} \times \frac{1}{689.01} = 7.14 \times 10^{-6}\frac{m^3}{s}$$

The density of the rich solution $(\rho_{rs}) = 689.01\ \frac{Kg}{m^3}$

The required work output$= Discharge \times Pressure\ rise$

## V. Determine of the Amount of Heat Energy Transferred from the Solar Radiation into the Solution (Ammonia-Water) in the Solar Collector

In general the total radiation [specula + diffused] on any surface on earth is about 955 W/ m², taking the inclination of flat plate collector at 40° placed North – South, then the energy received by the solar collector can be calculated as following.

*a) The emission received outside the earth's atmosphere is determined from this relation*

$$= 3.816X10^{26} / 4\pi X(15X10^{10}) = 1349.6W/m^2$$

*b) Total energy received by the earth*

Assuming the earth a spherical body, the energy received by it will be proportional to the perpendicular projected area, i.e., that of a circle, $= \pi X\ r_e^2$ ($r_e$ is the radius of the earth).

Energy received by the earth $= 1346.6X\pi X(6.4X10^6)^2$ $= 1.736X10^{17}W$

- Energy received by the solar collector

The direct energy reaching the earth $= (1-.42)X1349.6$ $= 782.77W/m^2$

- The diffusion radiation

$$= 0.22X782.77 = 172.21W/m^2$$

- Total radiation reaching the solar collector

$$= 782.77 + 172.21 = 955W/m^2$$

- Solar collector area

$$A = 1X0.5 = 0.5m^2$$

The projected area $= AX\cos\theta$ $= 0.5X\cos 40^o$ $= 0.383m^2$

Therefore, Energy received by the solar collector $= 0.383X955 = 365.765W$

*Note:* Since solar energy is not constant throughout the day, an excess capacity of the solar collector is accepted and taken.

## VI. The Efficiency of the System

Basically, the efficiency of the absorption refrigeration unit powered by solar energy to produce a cooling effect in order to absorb heat from domestic water tank is determined based on the coefficient of performance as refrigeration reverse cycle.

COP=Cooling Effect /Heat aborted from solar radiation

$$\boxed{COP = \frac{501}{365.765} = 1.369}$$

## VII. Amount of Fresh Water Extracted from the Humid Air

Based on the atmospheric conditions in Jeddah-Saudi Arabia in summer which they are $T_{a1}= 40^oC$ ; $\phi_{a1} = 50\%$ ; $T_{de1} = 27^oC$ ; $\omega_{a1} = 23.25g_{H2O}/Kg_{dry.air}$ and since the properties of humid air leaving the air tunnel after passing the cooling coil where part of its humidity has been separated due reaching a temperature beyond of its dew point temperature are;

$T_{a2}= 10^oC$ ; $\phi_{a2} = 100\%$ ; $\omega_{a2} = 7.85g_{H2O}/Kg_{dry.air}$ , therefore the fresh water separated from the humid air stream is $\boxed{\dot{M}_W = \dot{M}_a X\rho_{aX}(\omega_{a1} - \omega_{a2}) = 50X1.2X(0.02325 - 0.00785) = 0.942Kg/h}$

## VIII. Conclusion

This project presents, a study of absorption refrigeration system for water extraction from humid air in Saudi Arabia is carried out by designing the components of the cycle through establishing a comprehensive mathematical model describing the entire processes accomplished within the major components of the unit based on heat and mass conservation balancing considering steady flow processes. The sizes of major components involved within the system have been determined. The amount of the extracted fresh water from humid air is determined for certain operation conditions in the Kingdome of Saudi Arabia, it was found for real atmospheric conditions to be $0.942Kg/h$ and the efficiency of the system is 1.369. The designed system can be widely used for water production from moisture air for remote regions.

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