Adekeye et al: Proc. ICCEM (2012) 229 - 242

Design and development of vapor absorption refrigeration system for rural dwellers

Adekeye, T. Oyedepo, S.O and Oyebanji, J.A

Mechanical Engineering Department, Covenant University, Ota Nigeria

Correspondence: tunde.adekeye@covenantuniversity.edu.ng

Abstract

In this study, the experimental analysis of the performance of vapor absorption system was developed and developed. The influence of generator, evaporator and condenser's temperatures on the system performance was studied using gas burner as source of energy, ammonia-water combination as working fluid and energy equations as governing equations for the work. There was variation in the results when compared with the earlier worker who used kerosene burner as source of energy and ammonia-water combination. Result of this study shows that if higher cooling capacity and also lower evaporator`s temperature are desired from the system, generator`s temperature should be increased considerably. Also the combination of an analyzer, rectifier and heat exchanger would produce better COP of the System.

***Key words**: Refrigeration, Vapour absorption, cooling load, COP*

1. Introduction

The role of energy to human being on the earth cannot be overemphasized; its application for basic human needs is inevitable.

But, a large population of people resides in rural areas where access to conventional electricity as source of energy for human use mainly for operation such as refrigeration which includes the processing and preservation of foods, mixed vegetables and other uses in the area is grossly unavoidable and an alternative source of energy is required (Omoniyi,2003).

In hot climates, the heating and the cooling demand of domestic dwellings can be reduced substantially with various measures such as good insulation, double glazing, use of thermal mass and ventilation.

However, 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 preferable that such a system is not powered by electricity.

During the last few decades, an increasing interest ,based on research and development, has been concentrated on utilization of non-conventional energy sources, namely solar energy ,wind energy ,tidal waves ,biogas ,geothermal energy ,gas burner, kerosene burner, hydropower ,exhaust from furnace ,biogas etc.

Among these sources, gas burner, this is available and could be used to power an active cooling system based on the absorption cycle.

Ammonia-water and lithium bromide-water absorption combinations are the most suitable for gas burner applications. Research has been performed for ammonia-water absorption system theoretically and experimentally.

The first purpose of the study is to design and construct an absorption refrigeration system powered by gas burner and ammonia-water as working fluid. The second purpose of the study is to investigate experimentally the effects of the operating temperatures on the coefficient of performance of the system (Adekeye, 2006).

Most essentially, a refrigerating system must incorporate a condenser, an evaporator, a generator and an absorber that serve to increase and decrease the pressure of the working fluid called the refrigerant which circulates between these components through connecting tubes. The energy required might be supplied through a gas burner in form of heat to replace the conventional electricity that is readily available (Williams,1955).

Refrigeration means the cooling or removal of heat from a system. The equipment employed to maintain the system at a low temperature is termed refrigerating system and the system which is kept at lower temperature is called refrigerant. Refrigeration is generally produced in one of the following three ways:

(i) By melting a solid

(ii) By sublimation of solid and

(iii) By evaporation of liquid (Williams, 1955)

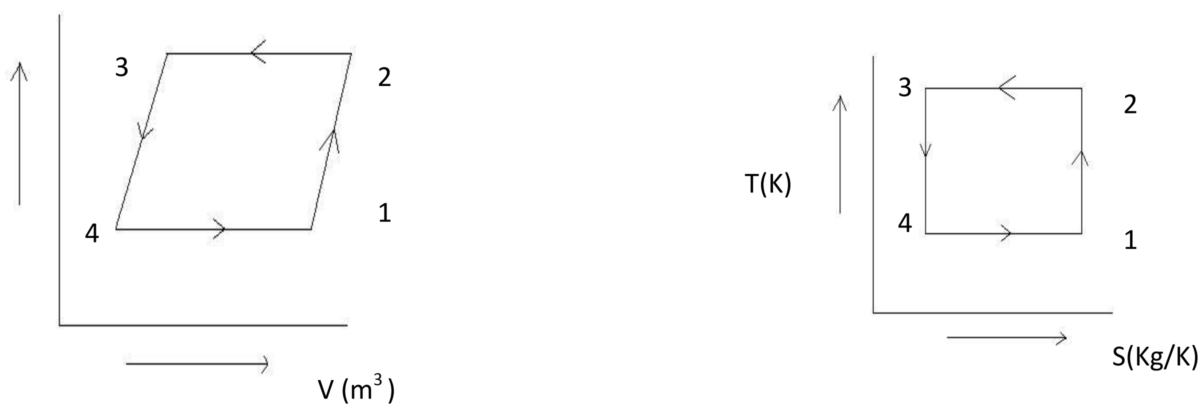
The refrigeration system can be classified into two main groups

(i) Vapor compression refrigeration system

(ii) Vapor absorption refrigeration system

1.1 Vapor compression system

The vapor compression refrigeration system cycle operates ideally on four processed as indicated in the diagram shown below.



Pressure – volume diagram

Temperature Entropy diagram

Fig 1-vapor-compression refrigeration cycle. 1-2-Adiabatic compression at constant entropy; 2-3-Isohermal heat rejection

3-4-Adiabatic expansions at constant entropy; 4-1 –Isothermal heat absorption (Bacom and Stephen,1991).

The compression process is affected by the vapor compression incorporated in the system, which compressed the vapor compression from the evaporator and raised its temperature and pressure to a certain value to allow the condensation of the vapor in the condenser. The condenser fluid then expands into evaporator through the expansion device .In the evaporator, the low pressure refrigerant vaporizes, taking heat of vaporization from the conditioned space surrounding it thus producing the cooling effect. The vaporized refrigerant flows to the compressor via the suction line and the cycle continues (Bacom and Stephen, 1991).

1.2 Vapor absorption refrigeration system

The absorption refrigeration system cycle is similar to the vapor compression cycle in that it employs a volatile refrigerant usually either ammonia-water, which alternatively vaporizes under low pressure in the evaporator by absorbing latent heat from the material being cooled and condenses under high temperature in the condenser by releasing the latent heat of the condensing medium.

The principal difference in the absorption and compression cycles is the motivating force that circulates the refrigerant through the system and provides the necessary pressure differential between the vaporizing and condensing processes.

In the absorption cycle, the vapor compressor employed in the vapor compression cycle is replaced by an absorber and generator, which perform all the functions performed by the compressor in the vapor compression cycle. In addition, the energy input required by the vapor compression cycle supplied by the mechanical work of the compressor, the energy input in the absorption cycle in the form of heat supplied to the generator is usually low pressure steam or hot water, although in smaller systems the heat usually supplied to the combustion of an appropriate fuel such as natural gas, propane or kerosene, directly in the generator.

The system consists of four basic components, an evaporator, an absorber which are located on the low pressure side of the system generator and a condenser which are located on the high pressure side of the system. Two working fluids are employed as refrigerant and absorbent.

The flow cycle for the refrigerant is from the condenser to the evaporator to the absorber to the generator and back to the condenser, while the absorbent passes from the absorber to the generator and back to the absorber .High pressure liquid refrigerant from the condenser passes into the evaporator through an expansion valve that reduces the pressure of the refrigerant to the low pressure existing in the evaporator. The liquid refrigerant vaporizes in the evaporator by absorbing latent heat from the material being cooled, and the resulting low pressure vapor then passes from the evaporator through an unrestricted passage to the absorber when it is absorbed and goes back into solution in the absorber.

The refrigerant flows from the evaporator to the absorber because the vapor pressure of the absorbent –refrigerant solution in the absorber that determines the pressure on the low pressure side of the system and consequently the vaporizing temperature of the refrigerant in the evaporator.

As the refrigerant vapor from the evaporator is dissolved into the absorbent solution, the volume of the refrigerant is decreased (compression occurs) and the heat of absorption is released. In order to maintain the temperature and vapor pressure of the absorbent solution at the required level, the heat released in the absorber which is equal to the sum of the latent heat of condensation of the refrigerant vapor and the heat of dilution of the absorbent, must be discarded to the surroundings(Bacom and Stephen,1991.)

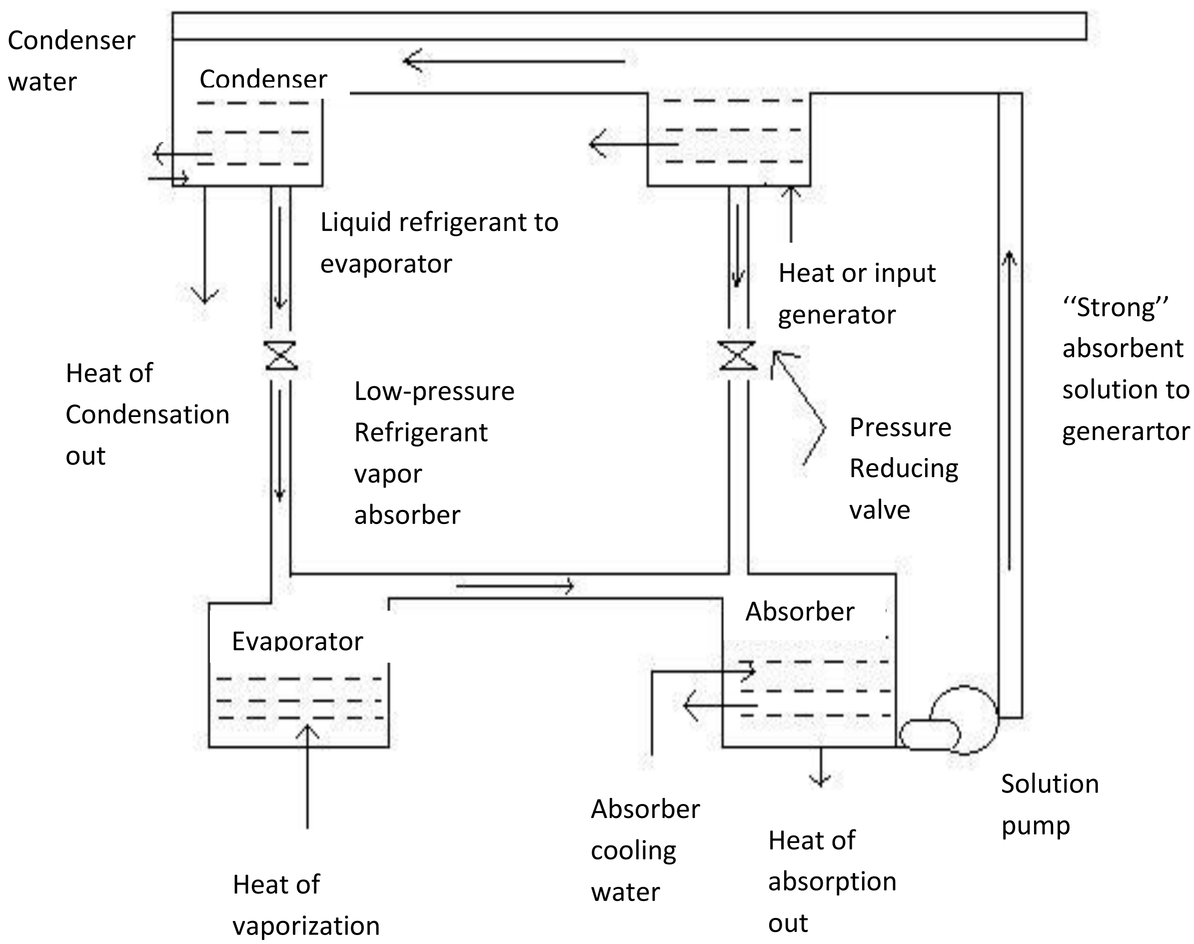


Fig.2-Basic Absorption Refrigeration Cycle (5)

1.2.1 Refrigerant-absorbent solution

To be suitable in an absorption system, there are certain criteria that the refrigerant- absorbent combination should meet, at least to some degree.

Obviously, the absorbent must have a strong affinity for the refrigerant vapor, and the two must be mutually soluble over the desired range of operating conditions. The two fluids should be safe, stable and non-corrosive both individually and in combination. Ideally, the absorbent should have a low volatile so that the refrigerant vapor leaving the generator will contain little or no absorbent and working pressure should be reasonably low are preferably near atmospheric pressure to minimize equipment weight and leakage into and out of the system. The refrigerant should have a reasonably high latent heat value so that the required refrigerant flow rate is not excessive.

For this project, ammonia-water combination was selected because of the following advantages:

Ammonia-water combination is widely used in domestic refrigerator and in commercial and industrial systems where the evaporator temperature is maintained close to or below 32˚F.

The water (absorbent) has a very strong affinity for ammonia vapor and the two are mutually soluble over a wide range of operating condition.

Both fluids are highly stable and are compatible with most materials found in refrigeration system.

The major disadvantage of the ammonia water system is the fact that the absorbent (water) is reasonably volatile so that the refrigerant (ammonia) vapor leaving the generator will usually contain appreciable amount of water vapor which is allowed to pass through the condenser and go into the evaporator will raise the evaporator temperature and reduce the refrigerating effect by carrying on vaporized refrigerant out of the evaporator.

For this reason, the efficiency of the ammonia-water system can be improved by the use of an analyzer and rectifier which function to remove the water vapor from the mixture leaving the generator before it reaches the condenser (Arora,2004).

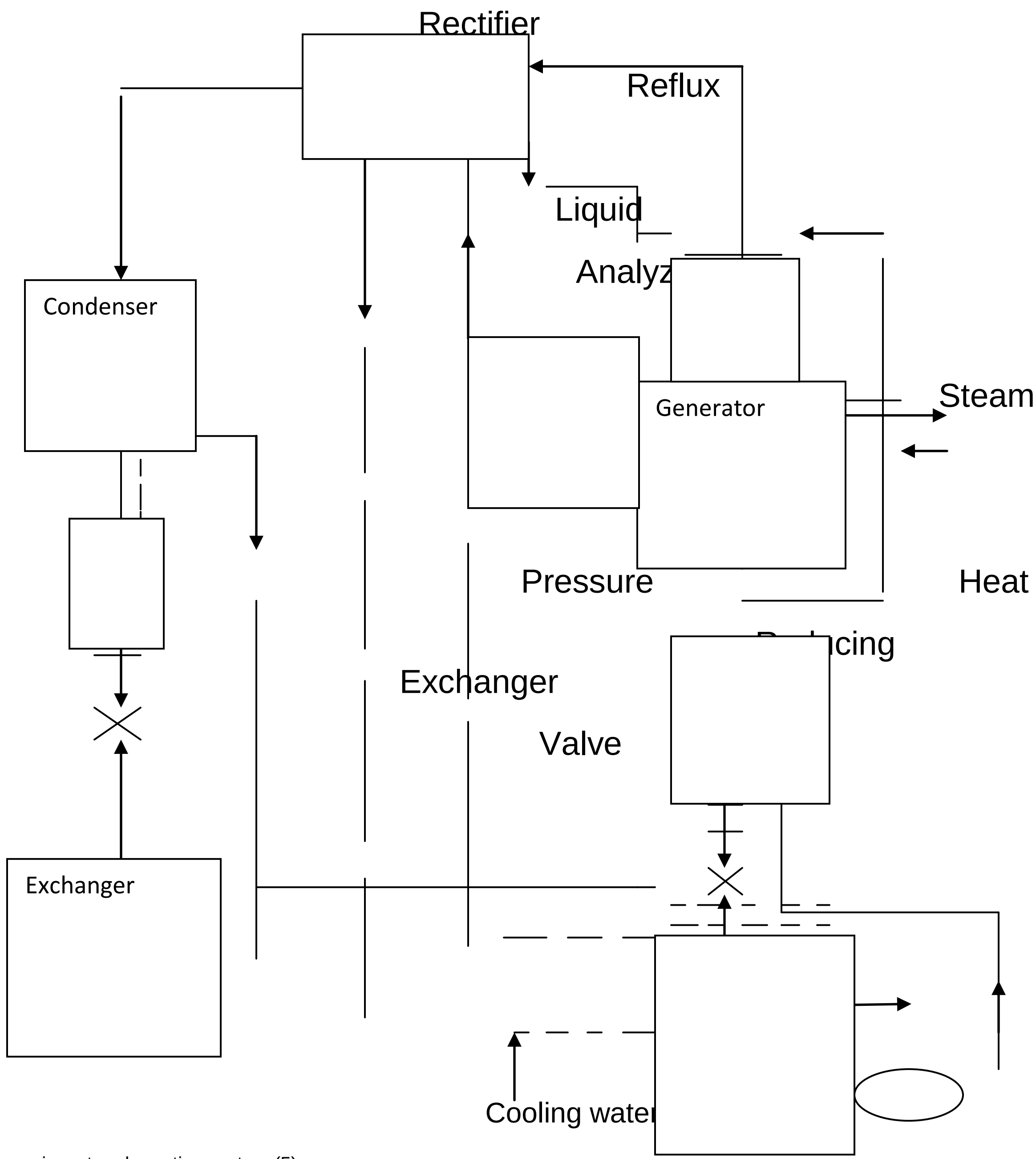


Fig.3 Ammonia-water absorption system (5)

In comparison, the followings are the advantages of vapor absorption refrigeration over compression system:

1) There is no moving part in the system except the aqua-pump motor. The pump motor is quite small as compared to the compressor motor in compression system.

2) Absorption system is quite in operation, very little wear and it has low maintenance cost

3) It can use processed steam, hot exhaust from the furnace, burner (kerosene or gas) solar energy as source of thermal energy in the generator.

4) The capacitor of the absorption system is controlled and maintained by adjusting the steam generator`s temperature even if the evaporator pressure falls.
5) The absorption system can be operated at the designed COP (Coefficient of performance) by the appropriate control of generator`s temperature.

6) It is much more compact and less bulky than the compression (Arora,2004).

2.0 Design specifications and assumptions

The design was based mainly on household refrigeration for both rural and urban Dwellers. The following assumptions were made:

APPLICATION: The system was designed for food preservation, mixed vegetables

AMBIENT TEMPERATURE: The maximum dry bulb and minimum wet bulb temperatures were taken to be 30˚C and 26˚C respectively. The temperature of the evaporator is assumed to be 0˚C while the product freezing temperature was considered to be 2˚C and the generator`s temperature was 100˚C.

2.1 Dimension of the Refrigerator

The unit consists of condenser, evaporator, expansion device, absorber and generator. The compartment formed serves as storage area for the products meant for freezing, therefore the following dimensions were considered

i) External Dimension of the Refrigeration

Length=breadth=800mm=0.8m each

Height=1200mm=1.2m high

ii) Internal Dimensions

Length=breadth=600mm=0.6

Height=1000mm=1m high

2.2 Insulation and Thickness

The insulation of the refrigerator prevents heat loses from compartment. The contents could thus be adequately preserved. Polyurethane was selected as the insulating material because of the following properties:

(i)High insulating strength

(ii)It has very low thermal conductivity when compared with other insulating materials

(iii)Water absorption rate is 0

For the thermal insulating material, polyurethane of 100mm thick was selected and used (Boast,1992).

2.3 Cooling Load Calculation

Cooling loads is the amount of heat that must be removed from the refrigerator space to provide and maintain the desired temperature. The total cooling loads of the refrigerator was determined by all the heat gained from various sources that must be removed from the refrigerated space. These include:

I. Wall, floor and cooling heat gained due to conduction

II. Air change load due to ingress outside, and the air from infiltration and door opening.

III. Product load from incoming goods to be reduced to storage temperature including freezing duties.

IV. Heat of respiration from stored product for food, drugs and farm produces.

V. Miscellaneous loads.

2.3.1 Wall, Floor and Ceiling Conduction Heat Gained

$Q_{gained} = AU\Delta T$

Where:

A = total external surface area of the refrigerated area in squared meters, obtained from:

[(length + width) x 2 x height] + [(change x width) x 2]

U = Overall rate of heat transfer for the well panel in W/m^2°C

ΔT = the temperature difference between the internal temperature and ambient temperature.

2.3.2 Infiltration Load

This occur when the warmth air from outside enters the refrigerated space to replace the more dense cold air while opening and closing the door of the unit.

The warmth air load becomes part of the total of the experiment that must be removed along with other sources of load. It is calculated thus,

I. Air change load (w) = (room volume x heat to be removed x number of air change per day i.e. 86,400)
II. Number of seconds in a day = 60 x 60 x 24

=

86,400secs

III. The room volume (m^3) is the internal volume of the refrigerated space, obtained by multiplying the internal length by the width and height dimensions of the room.
IV. The heat to be removed (j) is that of the infiltration air.
V. The number of air change depends on the temperature i.e. above or below 0°C

2.3.3 Product Load

This is the load that is required to be removed from the contents of the refrigerator in other to reduce their temperature

2.3.3.1 Product Cooling above Freezing

Product load (w) = weight of product x specific heat x temperature difference/86,400. The weight product (kg) loaded per day (24 hours)

Freezing

Product load (w) = (weight of product x latent heat/86,400)

The height of the product is the weight load per day. The freezing process of the product starts from outside when the outside surface has frozen a barrier is formed to the transfer of heat from the outside of the product which will prolong the freezing time.

2.3.3.2 Product Cooling below Freezing

Product load (w) = (weight of product x specific heat/86,400). The parameters are for cooling above freezing because the application is a continuous freezing process; it is normally calculated on the basis of 16 to 18 hours running time per 24hours.

Respiration

Product load (w) = (weight of product x heat of respiration/86,400). The weight of product is the total height in store (not that it was brought into the store par day) (Boast,1992).

2.4 Adopted Specifications

Cold room temperature = 0°C

Ambient temperature = 35°C

Room diameter (internal) = 0.6m long x 0.6m wide

Insulation – polyurethane foamed panel insulation thickness = 100mm

External diameter = 0.8 x 0.8 x 1.2

Product load = 40kg per day entering at 25°C and reduce to 0°C

Product specific heat = 3.97 x 10^3 kJ/kg above freezing and 2.03kJ/kg below freezing

Product latent heat = 31.4kJ/kg

Product freezing temperature = 0°C

2.4.1 Wall Load

Surface area = [(0.8 x 0.8) x 2 x1.2] x [(0.8 x 0.8) x 2]

= 3.84 + 1.28

= 5.12

≈ $5m^2$

Heat gained = Surface area x thermal transmittance x temperature difference

= 5 x 0.14 x (25-0)

= 5 x 0.14 x 25

= 17.5W

2.4.2 Air Infiltration Load

Load = volume x heat factor x air change/86,400

$= (0.36 \; x \; 109{,}000 \; x \; 0.8)/86{,}400$

$= 31{,}392/86{,}400$

= 0.36W

2.4.3 Product Load

Load = product height x specific heat x temperature difference/86,400

$= 40 \; x \; 39{,}700 \; x \; (25 - 0)/86{,}400$

$= 39{,}700{,}000 \,/86{,}400$

= 459.5W

2.4.4 Heat of Respiration

Load = product height x heat of respiration/86,400

$= 1200 \; x \; 3200/86{,}400$

= 44.6W

Total Load = 16.1 + 0.36 + 459.5 + 44.6

= 520.4W

Allowing 16 hours running time per day, then:

Load = 520.4 x (24/16)

= 780.6W or 0.781KW

Total load = wall load + infiltration load + product load + respiration

= 780.6W

Q_E = 0.781KW

2.5 Determination of Mass Flow Rate

The flow of the refrigerant through the generator, the condenser is assumed to be in continuous system. For analytical evaluation of the ammonia- water cycle, the following simplifying assumptions were made.

The refrigerant and absorbent phases at evaporator inlet, absorbent outlet, bottom of the generator, the condenser inlet and vapor from the rectifier are all in equilibrium.

The total pressure of the fluid solution is constant through the system except in the expansion device.

From energy flow equation, the mass flow rate of refrigeration in the equation (m) is given as:

$Q_E = m\Delta H \quad (m = m_i)$

Where:

Q_E = cooling load capacity in kJ/s or kW

M = mass flow rate of the evaporator in kg/s

ΔH = change in enthalpy (refrigerating effect (RE))

$$M = Q / \Delta H \text{ (RE)} \qquad kg/s$$

2.5.1 Determination of Mass Flow Rate of Weak Solution

To determine the mass flow rate of weak solution, the following equations were employed.

From overall mass balance equation

= $m + m_2^1 = m_1^1$.. (1)

= $m_1 x_1 + m_2^1 x_2 = m_1^1 x_1 = (m_1 + m_2^1) x_1^1$ 2

Where:

m^i_1 = mass flow rate for strong solution

m^i_2 = mass flow rate of weak solution

m_1 = mass flow rate of weak solution in evaporator

x^i_1 = mass concentration of ammonic solution

From equation 1 and 2 above

= $m_1 x_1 + m_2^1 x_2^1 = m_1^1 x_1^1 + m_2^1 x_1^1$................. 3

= $m_1 x_1 + m_1 x_1^1 = m_2^1 x_1^1 + m_2^1 x_2^1$

= $\dfrac{m_1 x_1 - m_1 x_1^1}{x_1^1 - x_2^1}$ = $\dfrac{m_1 (x_1^1 - x_2^1)}{x_1^1 - x_2^1}$

Therefore $m_2^1 = \dfrac{m_1 x_1 - m_1 x_1^1}{x_1^1 - x_2^1}$ 4

2.5.2 Determination of Mass Flow Rate of Strong Solution

From the overall energy balance equation

$m_1^1 = m + m_2^1$ 5

Where:

m^i_1 = mass flow rate for strong solution

m = mass flow rate in evaporator

m^i_2 = mass flow rate of weak solution

Determination of Mass Flow Rate and Rate of Heat in the Absorber, Generator and Condenser

2.5.3. Determination of Absorber Heat Ejection (Q_a)

From energy balance equation for absorber

= $Q_A + m_1^1 h_1^1 = m_1 h_1 + m_2^1 h_2^1$

Q_A = $m_1 h_1 + m_2^1 h_2^1 - m_1^1 h_1^1$................. 6

Where:

Q_A = absorber heat ejection

$m_1 \, or \, m$ = mass flow rate

m_1^1 = mass flow rate for strong solution

m_2^1 = mass flow rate of weak solution

h_1 = enthalpy

h_1^1 = state point

h_2^1 = state point

2.5.4 Determination of Generator Heat Transfer

From energy balance equation for generator (Q_G)

$Q_G + m_1^1 h_1^1 = m_2^1 h_2^1 + m_2 h_2$ = $m_2^1 h_2^1 + m_1 h_2$

$= m_2^1 h_2^1$

$+ m_1 h_2 - m_1 h_1$....................... 7

Where:

$(m_2 = m_1 = m)$

2.5.5 Determination of Condenser Heat Ejection (Q_c)

$Q_C = m\,(h_2 - h_3)$ 8

Where:

Q_C = condenser heat ejection

m = mass of flow rate

h_2 - h_3 = change in enthalpy refrigerating effect

2.5.6 Determination of Overall Energy

Overall energy balance = $Q_G + Q_E - Q_A - Q_C$

2.6 Coefficient of Overall Energy Balance by Coefficient of Performance

C.O.P = $\dfrac{Q_E}{Q_A}$ 9

2.7 Determination Of Coefficient Of Performance Of An Ideal Vapour Absorption System

From the first law of thermodynamics neglecting Q_P, we have;

$Q_C = Q_G + Q_E$ 10

As the vapour absorption system can be considered as a perfectly reversible systems, therefore the initial enthalpy of the system = enthalpy of the system after the change of its condition.

i.e.

$$\frac{Q_G}{T_G} + \frac{Q_E}{T_E} + \frac{Q_C}{T_C} \quad \ldots \ldots \ldots \ldots 11$$

From equation (2)

$$\frac{Q_G + Q_E}{T_C} = \frac{Q_G}{T_C} = \frac{Q_E}{T_C}$$

Or

$$\frac{Q_G}{T_G} - \frac{Q_G}{T_C} = \frac{Q_E}{T_C} - \frac{Q_E}{T_C}$$

Or

$$Q_G\left(\frac{T_C - T_G}{T_G\ X\ T_C}\right) = Q_E\left(\frac{T_E - T_C}{T_C\ X\ T_E}\right)$$

Or

$$Q_G = Q_E\left(\frac{T_E - T_C}{T_C\ X\ T_E}\right)\left(\frac{T_C - T_G}{T_G\ X\ T_C}\right)$$

$$= Q_E\left(\frac{T_C - T_E}{T_C\ X\ T_E}\right)\left(\frac{T_G - T_C}{T_G\ X\ T_C}\right) = Q_E\left(\frac{T_E - T_C}{T_E}\right)\left(\frac{T_G}{T_C - T_G}\right)$$

.... 12

Maximum coefficient of performance of the system

$$C.O.P_{max} = \frac{Q_E}{Q_G} = \frac{Q_E}{\left(\frac{T_E - T_C}{T_E}\right)\left(\frac{T_G}{T_C - T_G}\right)} = \left(\frac{T_E}{T_C \, X \, T_E}\right)\left(\frac{T_G - T_C}{T_G}\right)$$

…………………………………………………………………………… (13)

3.0 RESULTS AND DISCUSSION

In this study, mass flow rate and heat transfer in the evaporator, absorber, generator and condenser of designed and developed vapour absorption refrigeration system for rural dwellers were determined. The following specifications were used to determine the mass flow rate and heat transfer in the above components.

Generator temperature = 100°C

Condenser temperature = 45°C

Evaporator temperature = 0°C

Absorber temperature = 30°C

Evaporator - the main aim of refrigeration is achieved in the evaporator, so in designing the system, care was taken to ensure enough surface area so as to enhance the design refrigeration capacity.

Determination of Refrigerating Effect

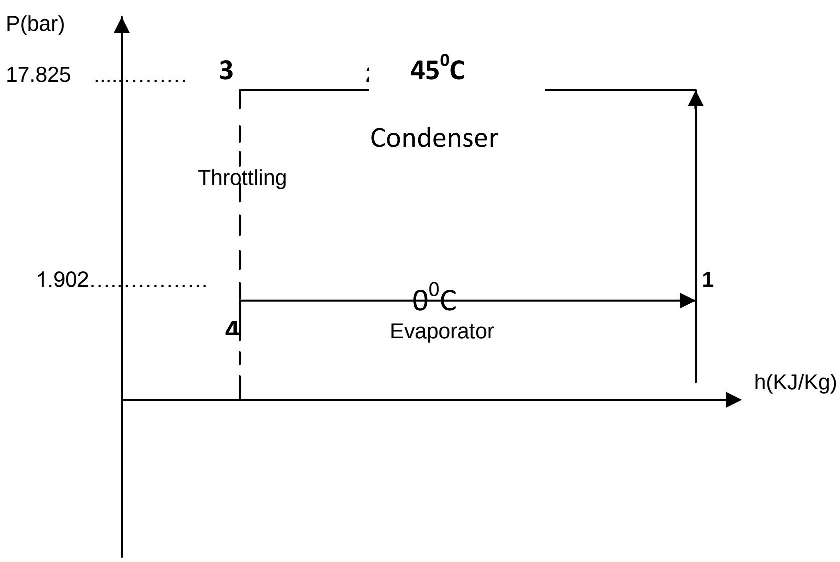


Fig 4: Vapour Absorption Cycle

From Ammonia (NH_3) tables, T_2= 45°C

P (cond) = 17.85 bar from steam table

At 0°C, P_{evap} = 1.902 bar (steam table)

From steam table,

$h_1 = h_g$ = 1444.4kJ/kg (saturated vapor 0°C)

$h_2 = h_g$ = 1474.4kJ/kg (saturated vapor 40°C)

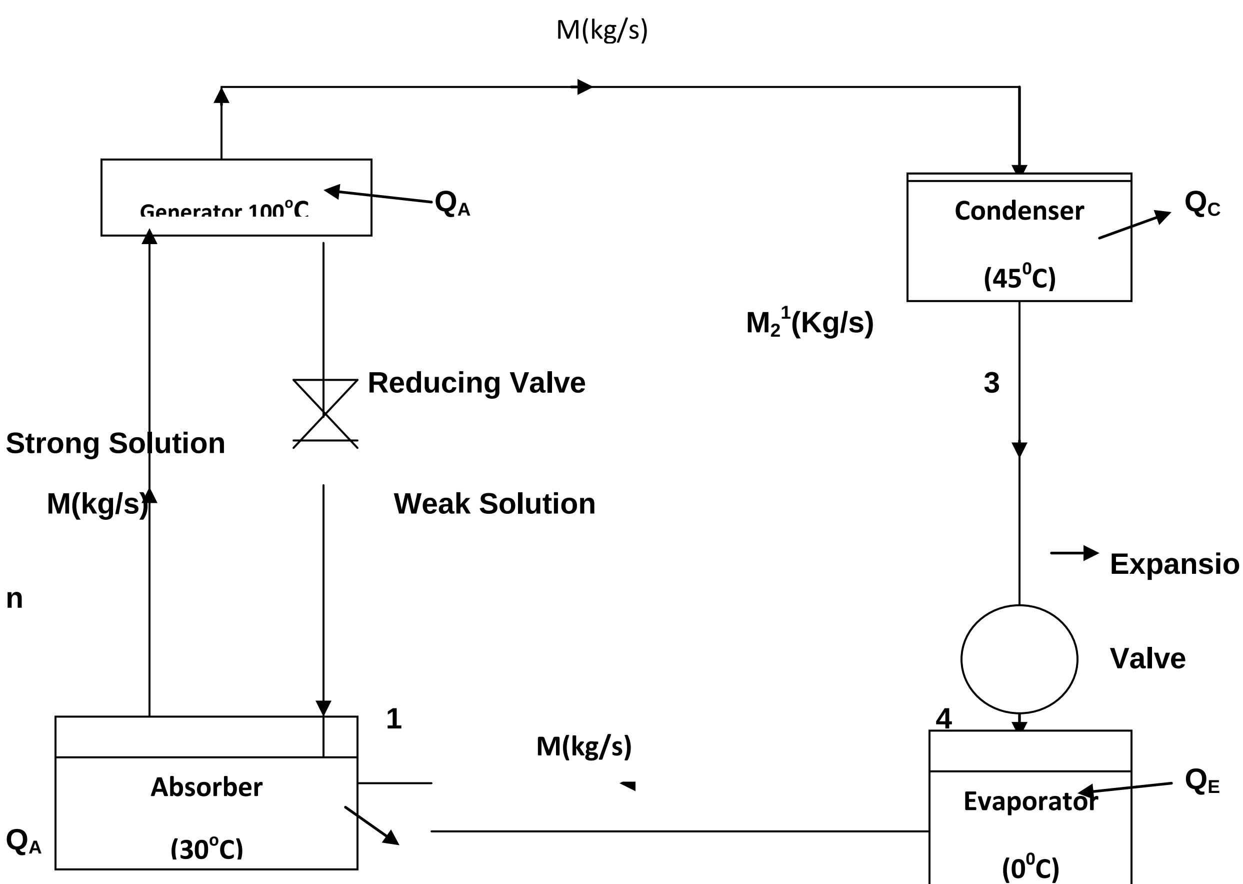
$h_3 = h_g$ = 396.8kJ/kg (saturated vapor -45°C)

But $h_3 = h_4$ (throttling process)

Refrigerating effect (RE) = $h_1 - h_4$

= 1444.4 – 396.8

= 1047.6kJ/kg


M(kg/s)
Generator 100°C
QA
Condenser
(45°C)
QC
M2 1(Kg/s)
3
Reducing Valve
Strong Solution
M(kg/s)
Weak Solution
Expansio
n
Valve
1
4
M(kg/s)
Absorber
(30°C)
QA
Evaporator
(0°C)
QE

Assumed Saturated Vapour

Fig 5 Schematic arrangement of vapour absorption System

Table 1 : showing the enthalpy, ammonic concentration x weight fraction at some selected points along the flow above.

Point in the figure	Ammonic concentration	Enthalpy h (kJ/kg)
1^1	0.35	22
2^1	0.1	695
1	0.945	1420
2	0.945	1474.45
3,4	0.945	396.8

Based on the specifications given above, mass flow rate of refrigerant through the generator and absorber is 0.000745kg/s, mass flow rate of weak solution from generator is 0,00193kg/s and mass flow rate of strong solution from the absorber is 0,002682kg/s. the heat supplied to the generator is 2.39kJ/s, heat rejected in the condenser is 0.8028kJ/s, while heat rejected from the absorber is 2.345kJ/s. the coefficient of performance (COP) of the designed and developed vapour absorption system is 0.32.

4.0 CONCLUSION

In this work, an experimental absorption refrigeration system powered by a gas burner was designed and developed and the system was tested successfully.

A lot has been reviewed and analyzed about vapor Absorption Refrigeration System in this work such as its applications, material selection with their properties, various components used and their sizes, their functions, selection of suitable refrigeration-absorbent combinations with their characteristics, cooling loads and other mathematical analysis.

It was seen that if higher cooling capacity and also lower evaporator temperatures are desired from the system, the generator `s temperature should be increased considerably. In order to improve the COP of the system, combination of analyzer, rectifier and heat exchanger should be incorporated to the system and a better source of energy should be employed.

References

Adekeye, T (2006), Design and Construction of Vapor Absorption Refrigeration System, (Unpublished PGD Project) LAUTECH, Ogbomosho, Nigeria

Althouse,T.B (1935), Modern Refrigeration and Air Conditioning, Good-Heart Wilcox Company, USA

Arora, C.P (2004), Refrigeration and Air Conditioning, McGrawhill, New Delhi (India)

Bacon,D.H and Stephen,R.C (1991), Basic Thermodynamic and Heat transfer, Butter Worth and Co. Ltd ,London.

Boast, M.F.G (1992), Cooling Loads Capacity, 2nd Edition, CBS, Delhi (India)

Eastop, T.D and Conkey,M.C (1993), Applied Thermodynamic, 5th edition, Longman, London.

Enibe, S.O (1997),“Solar Refrigeration for Rural Application energy” Elselver Science Ltd., Britain, Vol 12, No2, Page 157-158.

Omoniyi E B (2003), Design and Construction of Vapor Absorption System for Rural Dwellers,(Unpublished B.Tech Project), LAUTECH, Ogbomosho, Nigeria

Rajput, R. K (2006), Refrigeration and Air Conditioning, Kateria and Son, Delhi(India),page 438-446.

Williams, D.A (1955), “Intermittent Absorption Cooling System”, General Refrigeration, Engineering Journal, pg 132.