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## Development of 112W Vapour Absorption Refrigeration System for Cars

By

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### ABSTRACT

*Twentieth century vehicles and some twenty first century vehicles were not manufactured with the provision of a car refrigeration system. Incorporating vapour compression refrigeration system in these vehicles will pose problems because their engines were not designed to carry the load which the present car refrigeration system operating on the vapour-compression cycle employs. So a car refrigerating unit which operates on the principle of vapour absorption refrigerating system, using exhaust heat as its energy for operation, was designed and constructed. The major parts of the refrigerating system are the evaporator, condenser, generator and absorber and their capacities were determined to be 112W, 111W, 210W and 440W respectively. On performance evaluation, using water and ammonia as absorbent and refrigerant respectively as the working fluid, the refrigerator was able to achieve an evaporator temperature of 21° C and had actual Coefficient of Performance (COP) of 0.32. Although the actual coefficient of performance of the refrigerating system is low, it is still useful for keeping passenger consumables fresh and cool throughout a journey.*

**Keywords:** Vapour, absorption refrigeration system, heat, load, capacity, COP.

### INTRODUCTION

Refrigeration is the process of cooling a space or substance which encompasses humans and non-humans below environmental temperature (Krasner-Khait, 2016). Vapour compression refrigeration system is one of the most popular refrigeration systems widely use today. It finds its wide application in air conditioning of buildings and automobiles inhabited by human, machines and equipment that requires to be cooled for comfort in case of human, preservation or proper functioning in case of machines and equipment. It also finds its application in the cooling of foods and beverages, either for immediate consumption or preservation in our homes, offices, automobiles etc.

Twentieth century vehicles and some twenty first century vehicles were not manufactured with the provision of a car refrigeration system. Incorporating vapour compression refrigeration system in these vehicles will pose problems because their engines were not designed to carry the load which the present car refrigeration system operating on the vapour-compression cycle employs. In order to circumvent the problem of retrofitting vapour compression system refrigerating system into these cars, vapour absorption refrigeration system is being considered appropriate for the cars.

There are two main types of absorption refrigeration system design namely the two fluid and the three fluid absorption system. The differences between them is the way the liquid refrigerant is made to evaporate. In a two fluid system, a large pressure drop, which

causes the liquid refrigerant to evaporate, is achieved by the use of expansion valve. In a three fluid system, the expansion by means of partial pressures is facilitated by the use of a third fluid. The three fluid absorption system mainly use ammonia as their refrigerant and hydrogen as the third fluid. Figure 1 shows a flow diagram of a three fluid absorption system that uses ammonia and hydrogen as the third fluid.

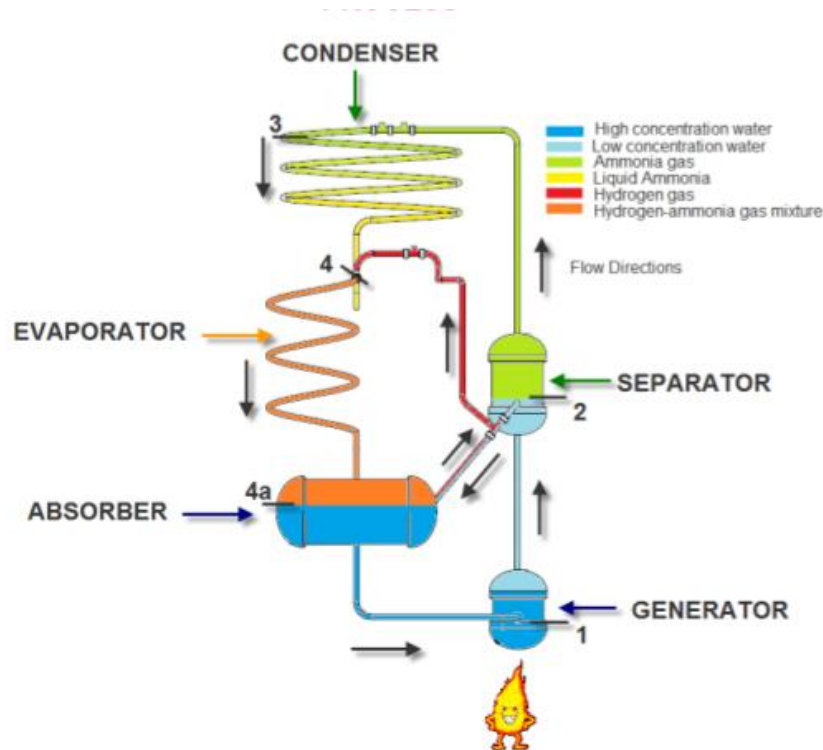


Figure 1. Flow diagram of three fluid absorption system.

Source: Kajano *et al.* (2012)

In line with Kajano *et al.* (2012) description of basic three fluid absorption system showing state points, at (1) the heat applied to the generator causes the strong aqua-ammonia solution to change phase that is vaporize up. At (2) the solution and gas gets into the separator where the water vapor condenses to liquid and gets back to the absorber. Superheated ammonia vapour now rises to the condenser while a small amount of weak aqua-ammonia solution is left in the separator. At (3) the ammonia running through the condenser, releases heat to the surroundings and undergoes change of phase that is condenses to liquid. At (4) liquid ammonia come together with hydrogen and enters the evaporator. At this point, the partial pressure of the hydrogen and ammonia reduces the saturation temperature of the liquid ammonia, thereby causing the ammonia to evaporate. This expansion reduces the temperature of the hydrogen-ammonia mixture, thereby allowing it to absorb heat from the refrigerator compartment and cooling is provided. At (4a) the hydrogen-ammonia mixture leaves the evaporator for the absorber in a gaseous state. In the absorber, the gaseous ammonia and hydrogen come together with liquid water. The density of ammonia is less than that of hydrogen, so ammonia sinks and accumulate under the hydrogen. This lead to an increase its partial pressure and a phase change that is back to liquid. Since ammonia is

soluble in water, ammonia and water are capable of forming a solution together again. Hydrogen is not capable of mixing and therefore continues to circulate back to the evaporator top as a pure gaseous substance. The ammonia and water solution exit the absorber and move to the generator and then starts the cycle again.

The energy or heat required by the generator for its operation in the vapour absorption refrigerating system can be obtained from about 65% to 60% or 60% to 70% estimated by Jadhao and Thombare(2013) of total energy in the form of heat which is wasted into the atmosphere from IC engine through the exhaust gases. Mittal et al (2015) remarked that there is an obvious demand to utilize this waste heat from the IC engine of a vehicle in particular, for useful purpose. In view of this fact and the feasibility of using waste heat from internal combustion engines for vapour absorption refrigeration system shown by Jianbo and Shiming(2013), Thakre et al(2014), Vazhappilly et al(2013), Maurya et al(2014), Manzela et al(2010) and Mageshwaran et al(2016), part of the heat that is wasted through the exhaust of a car is conceived in this research work to be used by the generator for the generation of alumina vapour in the vapour absorption refrigeration system.

So the aim of this research is to develop vapour absorption refrigeration systems for cars using the waste heat from the exhaust for the operation of the generator.

## MATERIALS AND METHOD

### Materials

The materials and equipment used included: insulators (polyurethane and rubber gasket), aqua ammonia solution, hydrogen gas, non-return valve, galvanized metal sheets, aluminum metal sheet, a Honda civic 1994 model made in Japan, mild steel pipes and Type K-thermocouples (manufactured in China by Nigbo Taisuo Tech. Ltd),

### Design Theory and calculations

The following parts of the vapour absorption refrigerating system were considered for the design: Refrigerator cabinet, Generator, Condenser and Evaporator. For the purpose of the design the following parameters and assumptions were adopted and used based on literatures. Length (l), width (w) and height (h) of the refrigerator cabinet = 0.3, 0.25 and 0.25m respectively. Lagging thickness,  $t = 0.05\text{m}$ , external casing thickness,  $t_e = 0.0015\text{m}$  and internal thickness,  $t_i = 0.0004\text{m}$ . Ambient temperature =  $30^\circ\text{C}$ , condenser inlet temperature,  $T_c = 40^\circ\text{C}$ , evaporator temperature,  $T_e = 4^\circ\text{C}$ , absorber temperature,  $T_a = 40^\circ\text{C}$  and generator temperature,  $T_g = 90^\circ\text{C}$

### Determination of volume of the refrigerator cabinet

The refrigerator cabinet is cuboid in shape and the volume  $V_C$ , is given as

$$V_C = l \times w \times h \quad (1)$$

$$V_C = 0.3 \times 0.25 \times 0.25 = 0.019\text{m}^3$$

### Determination of wall heat gained load

The heat flow by conduction, through the insulated walls into the refrigerated space is the wall heat gained load and it was determined using the heat transfer equation stated as

$$Q_w = AU\Delta T \quad (2)$$

Where  $Q_w$  = rate of heat transfer in (W), which is the wall heat gained load,  $A$  = outside surface area of the wall =  $0.81 \text{ m}^2$

$U$  = overall coefficient of heat transfer =  $0.52 \text{ W/m}^2\text{K}$ ,  $\Delta T$  = temperature difference across the wall =  $26 \text{ }^\circ\text{C}$

Therefore, the wall heat gained load,

$$Q_w = 0.81 \times 0.52 \times 26 = 10.95 \text{ W}$$

#### **Determination of air change load**

This is the heat flow from outside warm air into the refrigerated space due to door opening. It was calculated using the equation stated by (Bolaji *et al*, 2012) as

$$Q_a = m'(h_o - h_i) \quad (3)$$

$Q_a$  = air change load,  $m'$  = mass of air entering the space  $\text{kg}$ ,  $h_o$  = enthalpy of outside air,  $h_i$  = enthalpy of inside air

$$m' = \rho v; \quad (4)$$

$$\rho = \text{density of air at } 30^\circ \text{C} = 1.167 \text{ kg/m}^3$$

The volume of air around the refrigerator,  $v$  was determined using the dimensions of the booth of the car the refrigerator will be fitted.

where:

Length, width and height of the car booth are 1.29, 0.9 and 0.69m respectively.

$$V = 1.29 \times 0.90 \times 0.69 = 0.8011 \text{ m}^3$$

Specific heat of air  $C_p = 1.005 \text{ KJ/Kg K}$

$$m' = 1.167 \times 0.8013 = 0.934 \text{ kg}$$

$$h_o - h_i = C_p \Delta T = 1.005(30 - 4) = 26.13 \text{ KJ/Kg}$$

$$Q_a = 0.934 \times 26.13 = 24.41 \text{ kJ}$$

When it is expressed per day, it becomes

$$Q_a = (24.41/24 \times 60 \times 60) = 0.000283 \text{ kW or } 0.283 \text{ W}$$

#### **Determination of Product Sensible Heat Load**

The heat that is removed from the product stored in the refrigerated space is called the product sensible heat load and it was determined using the convectational relation

$$Q_p = \rho_w \times V C_p \Delta T \quad (5)$$

$$Q_p = 1000 \times 0.019 \times 4.2 \times (30 - 0) = 2394 \text{ kJ}$$

Where,  $Q_p$  is heat to be removed from product,  $\rho_w$  = density of water =  $1000 \text{ Kg/m}^3$ ,  $V$  = volume occupied by the product =  $0.019 \text{ m}^3$ ,  $C_p = 4.2 \text{ kJ/kgK}$  which is the specific heat capacity of product above freezing, and water was considered here as the product.  $\Delta T$  is change in temperature of water from  $30^\circ\text{C}$  to  $0^\circ\text{C}$  the ice point.

#### **Determination of Product Load due to Latent Heat of Fusion**

The heat that is further removed from the product in the refrigerated space at constant freezing temperature of the product,  $Q_f$  is termed product load due to latent heat of fusion and was determined using the convectational relation stated as

$$Q_f = m_w \times L = \rho_w \times V \times L \quad (6)$$

Where  $m_w$  is the mass of water and  $L$  is the latent heat of fusion of ice =  $334 \text{ KJ/Kg}$

$$Q_f = 1000 \times 0.019 \times 334 = 6346 \text{ kJ}$$

Therefore total product load per day,  $Q_{PT}$  was computed as

$$Q_{PT} = \frac{Q_P + Q_F}{24 \times 60 \times 60} \quad (7)$$

$$Q_{PT} = \frac{2394 + 6346}{86400} = 0.101 \text{ kW} = 101 \text{ W.}$$

Total cooling load,  $T_c$  is the sum of all the cooling loads which is given as:

$$T_c = \text{Wall heat gained load} + \text{air change load} + \text{product load} = 10.95 + 0.283 + 101$$

$$= 112.23 \text{ W} = 0.112 \text{ kW}$$

This total cooling load is the refrigerating effect,  $R_f$  of the refrigerant

### **Determination of mass flow rate of the refrigerant**

Refrigerating effect,  $R_f$  of the refrigerant is given in line with Kaushik and Singh (2014) as

$$R_f = \dot{m}_R (h_4 - h_3) \quad (8)$$

Where  $R_f = 0.112 \text{ kW}$

$\dot{m}_R$  = mass flow rate of the refrigerant.

Enthalpy of refrigerant saturated vapour,  $h_4$  at  $4^\circ\text{C} = 1466.40 \text{ kJ/Kg}$

Enthalpy of refrigerant saturated liquid,  $h_3$  at  $40^\circ\text{C} = 390.64 \text{ kJ/Kg}$

$$\dot{m}_R = \frac{0.112}{1466.40 - 390.64} = 1.041 \times 10^{-4} \text{ Kg/s}$$

### **Evaluation of Coefficient of Performance (COP)**

The performance of Vapour Absorption Refrigeration System was evaluated in terms of COP,

The theoretical COP is given as

$$(\text{COP})_T = \frac{\text{Heat absorbed in the evaporator}}{\text{Heat supplied in the generator}} = \left( \frac{Q_e}{Q_g} \right) \quad (9)$$

Where  $Q_g$  is the heat supplied in the generator and  $Q_e$  is the heat absorbed in the evaporator.

$$(\text{COP})_T = \frac{0.112}{0.21} = 0.53$$

For a vapour absorption refrigeration system the real or actual COP in line with Lesson (2018) is expressed as

$$(\text{COP})_R = \frac{T_e}{T_c - T_e} \times \frac{T_g - T_a}{T_g} \quad (10)$$

where,  $T_e$  is the evaporator temperature,  $T_c$  is the condenser temperature,  $T_g$  is the generator temperature,  $T_a$  is the absorber temperature.

### **Determination of capacity of the Evaporator**

The evaporator is a heat exchanger in the refrigerator where the liquid refrigerant (ammonia) extracts heat from the refrigerating compartment and evaporates. The capacity of the evaporator is the amount of heat absorbed over a given period. It is also known as the refrigerating capacity of the system which has been determined to be  $0.112 \text{ kW}$

### **Determination of Length of the Evaporator Pipe**

The evaporator capacity,  $Q_{ec}$  can be expressed as

$$Q_{ec} = AU\Delta T \quad (11)$$

The overall heat transfer coefficient,  $U$  For mild steel with water and air/gas passing through both sides respectively is  $11.3 \text{ W}/(\text{m}^2\text{K})$  (The Engineering tool box, 2016).

$$A = \frac{Q_{ec}}{U\Delta T} = 2\pi rl + 2\pi r^2 \quad (12)$$

where

The Evaporator capacity,  $Q_{ec} = 0.112 \text{ kW}$

$A$  is cross sectional area of the pipe

$\Delta T$  or LMTD =  $289.7 \text{ K}$

$U$  is Overall heat transfer coefficient

where  $r$  = radius of evaporator pipe =  $0.003 \text{ m}$

The length of the evaporator pipe was determined using equation 12 as

$$L = \frac{0.0342 - 2 \times 3.142 \times 0.003^2}{2 \times 3.142 \times 0.003} \quad L = 1.8 \text{ m}$$

### **Evaluation of capacity of the Condenser**

The fin and tube condenser was used and the cooling medium was air in this work.

The capacity of the condenser was determined in line with Matta et al (2017) by using equation 13

$$Q_{cc} = \dot{m}_R(h_2 - h_3) \quad (13)$$

where:

$Q_{cc}$  is the condenser capacity

mass flow rate of refrigerant,  $\dot{m}_R = 1.041 \times 10^{-4} \text{ Kg/s}$

From thermodynamics table,

Enthalpy of refrigerant vapour entering the condenser,  $h_2$  at  $90^\circ\text{C} = 1459.19 \text{ kJ/kg}$

Enthalpy of refrigerant liquid leaving the condenser,  $h_3$  at  $40^\circ\text{C} = 390.64 \text{ kJ/kg}$

$$Q_{cc} = 1.041 \times 10^{-4} (1459.19 - 390.64) \\ = 0.111 \text{ kW}$$

### **Determination of Length of the Condenser Pipe**

The condenser capacity can also be expressed as

$$Q_{cc} = AU\Delta T \quad (14)$$

To determine the cross sectional area of the condenser pipe

$$A = \frac{Q_{cc}}{U\Delta T} = 2\pi rl + 2\pi r^2 \quad (15)$$

where

$Q_{cc}$  is the condenser capacity =  $0.111 \text{ kW}$

$A$  is the cross sectional area of the pipe

$\Delta T = 297.85 \text{ K}$

$U$  = Overall heat transfer coefficient

Where  $r$  is radius of condenser pipe =  $0.004 \text{ m}$

So the length of the condenser pipe was determined using equation 15 as

$$L = \frac{0.033 - 2 \times 3.142 \times 0.004^2}{2 \times 3.142 \times 0.004} = 1.3 \text{ m}$$

### **Determination of capacity of the Absorber**

The absorber absorbs low pressure ammonia vapour from the evaporator and weak ammonia solution from the generator, makes this mixture into a strong solution, before it gets back to the generator.

The Absorber Capacity was obtained in line with Kaushik and Singh (2014), using equation

$$Q_{ab} = m_{ws} \times h_5 + \dot{m}_R \times h_4 - m_{ss} \times h_6 \quad (16)$$

where:

$Q_{ab}$  is absorber capacity,

From thermodynamics table,

Enthalpy of weak Solution,  $h_5$  at  $90^\circ\text{C}$  at =  $175\text{ kJ/kg}$

Enthalpy of refrigerant vapour,  $h_4$  at  $4^\circ\text{C}$  =  $1466.40\text{ kJ/kg}$

$m_{ss}$  is mass flow rate of strong solution and  $m_{ws}$  is mass flow rate of weak solution =  $1.187 \times 10^{-3}\text{ Kg/s}$  and  $1.291 \times 10^{-3}\text{ Kg/s}$  respectively as determined by Winze(2017)

$h_6$  is enthalpy of strong solution at  $40^\circ\text{C}$  =  $-50\text{ kJ/kg}$

$$Q_{ab} = 1.291 \times 10^{-3} \times 175 + 1.041 \times 10^{-4} \times 1466.40 - 1.187 \times 10^{-3} \times (-50) \\ = 0.44\text{ kW}$$

### **Determination of capacity of the Generator**

The generator is where solution of aqua-ammonia received from the absorber is heated by car exhaust and this heating causes ammonia solution to separate into ammonia vapour at high pressure and hot weak ammonia which mainly consists of water.

The generator capacity was obtained using the equation in line with Kaushik and Singh(2014), which is stated as

$$Q_g = \dot{m}_R \times h_1 - m_{ss} \times h_6 \quad (17)$$

Where from thermodynamics table,

Enthalpy of refrigerant vapour,  $h_1$  at  $90^\circ\text{C}$  =  $1459.19\text{ kJ/kg}$

Enthalpy of strong solution,  $h_6$  at  $40^\circ\text{C}$  =  $-50\text{ kJ/kg}$

Therefore

$$Q_g = 1.041 \times 10^{-4} \times 1459.19 - 1.187 \times 10^{-3} \times (-50) \\ = 0.21\text{ kW}$$

### **Determination of the pressure of Hydrogen gas Required**

In order to determine the pressure of hydrogen gas,  $P_H$  required in the system, the overall system pressure was taken as the saturation pressure of ammonia at ambient temperature this is in line with the work of Kajano et al (2012).

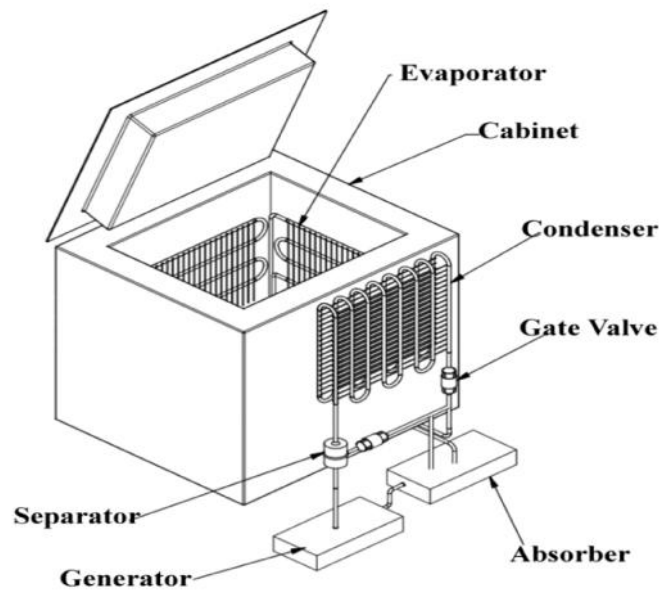
Saturation pressure of ammonia,  $P_T$  at  $30^\circ\text{C}$  is 11.6 bar

Since the required evaporator temperature is  $4^\circ\text{C}$ , the saturation pressure of ammonia,  $P_A$  at that temperature is 4.9 bar. From Daltons law of partial pressure;

$$P_T = P_H + P_A \quad (18)$$

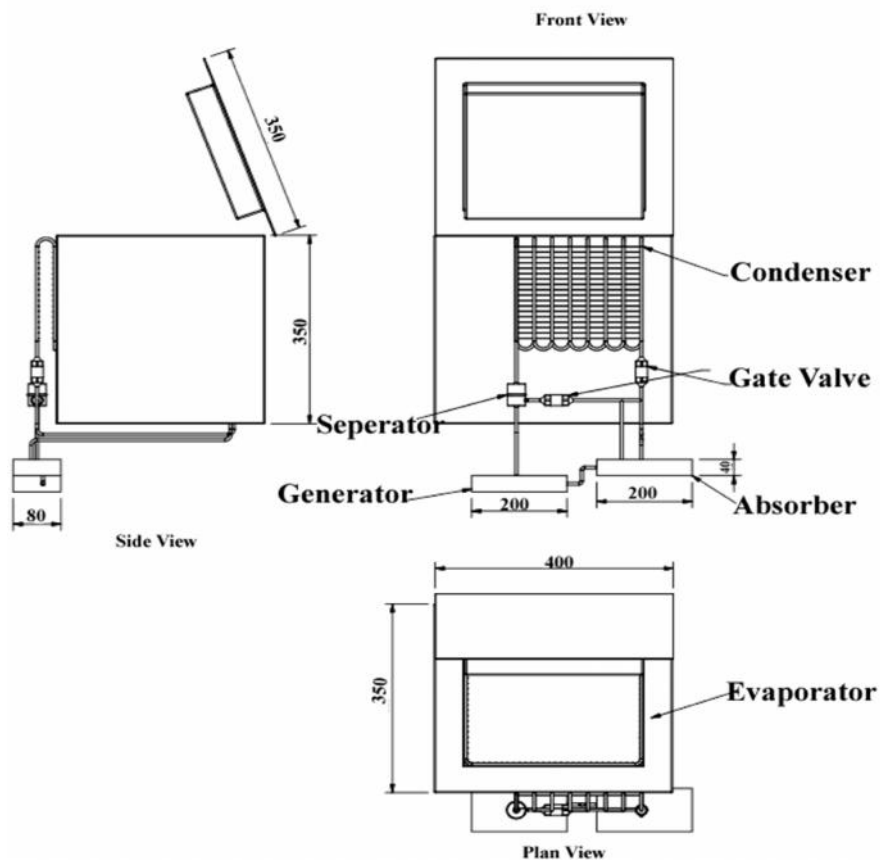
$$P_H = 11.6 - 4.9 = 6.7\text{ bar}$$

Based on the design theory and calculations, the vapour absorption refrigeration system for car was developed. The assembly and orthographic drawings are shown in figures 2 and 3



Scale 1:6

Figure 2: The refrigerator assembly



Scale 1:6



Figure 3. Orthographic diagram of the refrigerator

### **Installation and testing**

The refrigerator cabinet which has the evaporator and condenser was placed in the boot of the car as seen in plate 1. One liter of aqua- ammonia was poured into the generator through the inlet of absorber which was placed under the car with the generator mounted on the exhaust port and securely held in place with a metal strap. . After securely tighten the nuts, hydrogen gas was then compressed in to the system through an inlet located at the pipe connecting the condenser to the evaporator. A gas meter made by MKTA was used while compressing the hydrogen into the system to monitor the pressure so as not to exceed the designed pressure of 6.7bar.



Plate: 1: The vapor absorption refrigerator fitted to the car

To test the system, the car was driven from Minna to Abuja and temperatures of the components were taken in an interval of fifteen minutes, using a K-type thermocouple and the duration of the test was two hours.

### **RESULTS AND DISCUSSION**

The variation of the measured temperatures of the components viz a viz evaporator, condenser, generator and absorber of the vapour absorption refrigerator and the car exhaust with time is shown in figure 4. The average ambient temperature during the test day was 30°.

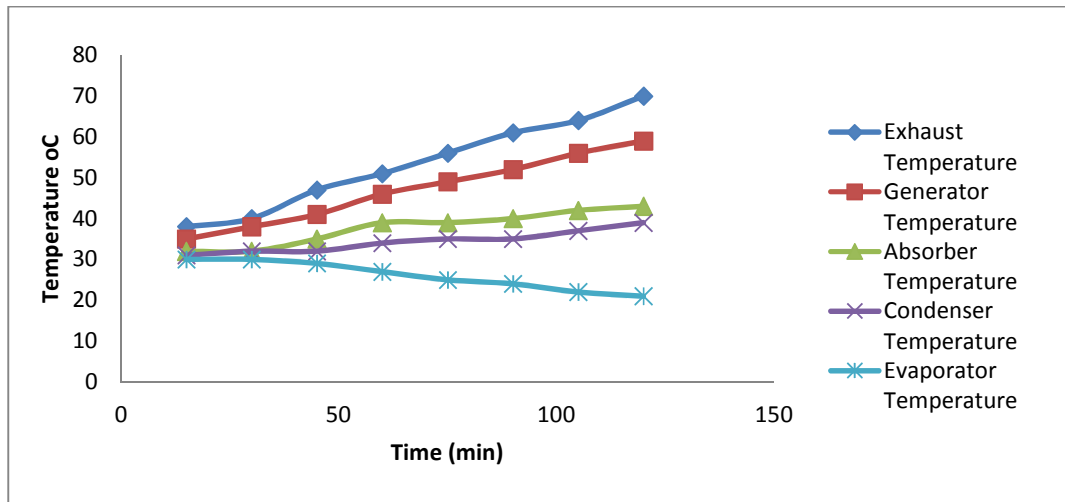


Figure 4: Variation of Components temperature with time

It can be seen from figure 4, that as the car exhaust temperature increased with time. That is, as the exhaust heat increased with time the generator temperature also increased with time. This increment is also observed in the absorber and condenser temperature while, the evaporator temperature decreased with increase in exhaust temperature with time.

The variation of evaporator temperature with actual COP is shown in figure 5.

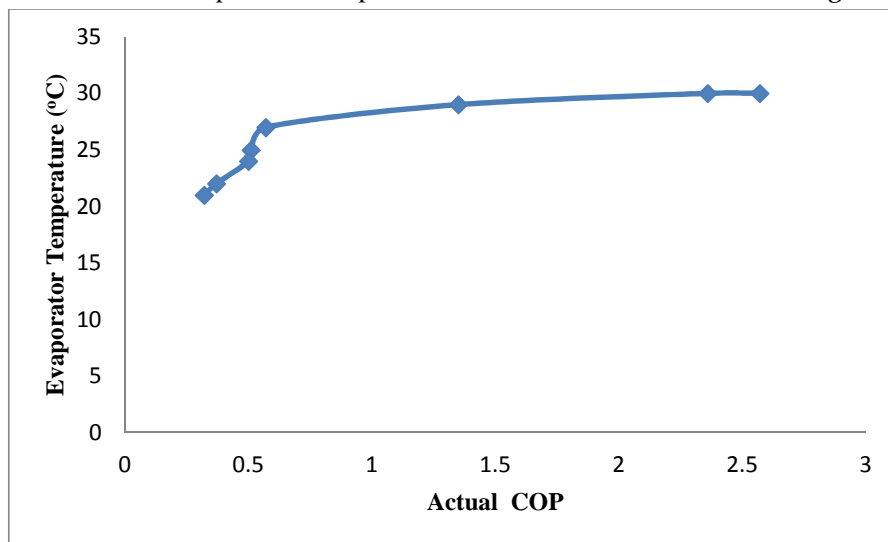


Figure 5: Variation of evaporator Temperature with COP

It can be seen from figure 5 that the actual COP of the system increased with increase in evaporator temperature. But decreased as the temperature of the evaporator reduces. This is due to the increased work done by the system in removing the heat from the refrigerated compartment. This is in line with the results of Crepinsek *et al.* (2009), Salim and Rajesh (2016) and they noted that high evaporator temperature has an impact on the overall COP of the system. More so in line with Mageshwaran *et al.* (2016) that obtained an evaporator temperature of 23 degrees after about an hour of operation. Also with Agra *et al.* (2014) who achieved evaporator temperature of 28 degrees with a maximum COP of 0.36 and attributed this to the low concentration of aqua- ammonia used. However at the end of two hours period

of the test, the actual COP of the system was found to be 0.32 as evident in figure 5. This value is less than that of theoretical COP calculated to be 0.53.

## CONCLUSION

Vapour absorption refrigeration system for cars have been developed in this work and fitted to a car without initial provision of refrigerating system. It can however be concluded that: emission of waste heat from car exhaust to the environment is minimized by the use of this developed refrigeration system. The performance of the car being fitted with the refrigerating system is not affected, because there are no moving parts that would need a power source for drive, which would have been sourced from the car engine. This system can be operated with a relative low generator temperature to reach low evaporator temperature with an acceptable system COP, which is the main advantage of this refrigeration system as it can utilize waste heat from cars. Although the coefficient of performance of the refrigerating system obtained in this research is as low as 0.32, it is still useful for keeping passenger consumables fresh and cool throughout a journey

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