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Development of vapour absorption A/C system using waste heat from an Engine exhaust

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ABSTRACT

The price of fuel is sky rocketing and so is the pollution in the atmosphere. And with the rise in number of vehicles on road the demand for fuel is also increasing rapidly. A large amount of the fuel is used up to run automobiles. These automobiles which use IC engines have a very low efficiency of about 30% to 35%. The rest of the 65% of heat input is lost in various forms to the surroundings through cooling water, exhaust gases and radiation. Still the 35% of work is not utilised completely due to dissipative effect of friction. The present day customer expects his ride to be comfortable. And with rising temperatures automobile air conditioning is a necessity. However, this comfort cooling comes at the cost of higher fuel expenses. The unit needs constant power which is obtained from the engine, thus resulting in loss of engine power as well as mileage. This project aims at reducing the fuel consumption of the vehicle and thus increase fuel economy by reducing the load on the engine by using the excess heat available in the exhaust gases and implementing the vapor absorption system for air conditioning.

KEY WORDS: vapour, engine, exhaust, pollution.

1. INTRODUCTION

The conventional ac system in automobiles uses vapour compression refrigeration system, where the compressor handles refrigerant in vapour form under adiabatic condition, according to adiabatic work relation for open system W = - V. dP (where V is volume in m3, dP pressure rise, W is work required to attain dP) if the fluid require pressure rise is vapour compared to liquid absorb more work, since volume V is much higher for vapour compared to liquid. Thus vapour absorption system is best alternative which handles liquid instead of vapour, but it require high temperature heat source for separation at high pressure end, luckily the availability of high temperature exhaust gas provides solution to this requirement with free of cost. This reduces heat loss to atmosphere and act as Waste Heat Recovery system.

2. EXPERIMENTAL SETUP AND METHODOLOGY

2.1. Proposed A/C Systems in Vehicles: The alternative air conditioning unit runs on the excess waste heat available in the exhaust gases. The condenser, expansion valve and the evaporator remain unchanged while the compressor is replaced by an absorber, pump, regenerator and generator, this is the absorption refrigeration system which involves the absorption of a refrigerant (NH_3) by a transport medium (H_2O). The saturated ammonia vapors from the evaporator enter the absorber where they react with water to form $NH_3.H_2O$. This reaction is exothermic, thus heat is released during this process. To maximize the concentration of NH_3 in H_2O it is necessary to cool the absorber. This cool $NH_3.H_2O$, rich in NH_3, is then pumped to the generator. Heat is transferred to the solution from the source to vaporize the solution. This vapour, which is rich in NH_3, passes through a rectifier, which separates the water and returns it to the generator. The high pressure NH_3 vapor then continues its journey through rest of the cycle. The hot NH_3+H_2O solution, which is weak in NH_3, then passes through a regenerator, where it passes some heat to the rich solution leaving the pump, and is throttled to the absorber pressure.

2.2. Where Do We Save Energy……….??: First, we have a very good source of heat i.e. the engine exhaust, which provides a temperature range between 90°C – 170°C, which is suitable to run this system. Second, the pump work required is almost negligible when compared to compressor work in VCR system.

Table. Engine Specifications

Make	Ruston	**Cooling system**	Water
Loading	Mechanical Brake	**No. of stroke**	Two
Type	Horizontal, Manual governing	**Rated Speed**	Variable
Brake Power	3.7 KW	**Radius of brake Drum**	24 cm
No. of cylinders	one	**Bore**	10.8 cm
Stroke	20cm		

2.3. Methodology: Calculation of the amount of waste heat available in the exhaust gases and cooling water. Calculation of flow rates, pressures and temperatures of various components of the proposed system. Corresponding COP of the proposed refrigeration system required to provide the desired cooling effect. Fabrication of the system and tabulating the temperatures at various points of the system at certain time intervals.

Fig.1.Experimental setup

2.4. Experimental observations and calculations:

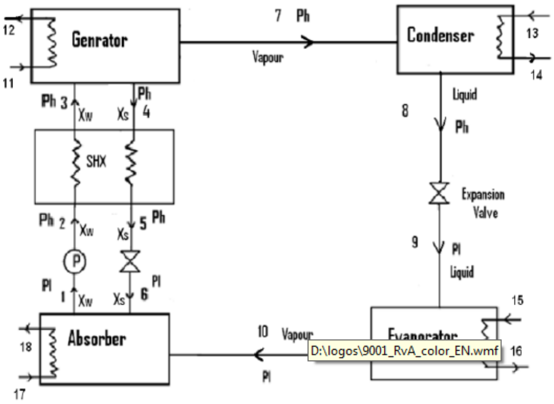

Fig.2.Layout of vapour absorption refrigeration system

Observations: The following readings are taken for each loading of the engine.
a) Time t sec for 20 cc of fuel consumption.
b) Engine speed (N) RPM
c) Load on brake drum in Kg.
d) Spring balance load in Kg
e) Manometer reading (h2-h1) in cm
f) Engine cooling water inlet temp T1°C
g) Engine cooling water outlet temperature T2°C
h) Engine cooling water mass flow rate m1 kg/sec
i) Exhaust gas temp, at calorimeter inlet T3°C
j) Exhaust gas temp, at calorimeter outlet T4°C
k) Water temp, at calorimeter inlet T5°C
l) Water temp, at calorimeter outlet T6°C
m) Engine Exhaust Gas Inlet Temperature (room temperature) T7°C
n) Engine Exhaust Gas Outlet Temperature T8°C

Table.1.Tabular column for air flow measurement

Loaded on hanger (w1) (kg)	0	10	23	33
Spring balance load (w2) (kg)	0	2	6	9
Net load (w1-w2)	0	8	17	24
Time for 20 cc of fuel cons. (sec)	204	144	96	80
Engine mass flow rate (kg/sec)	13	15	22	37
Engine exhaust gas temp. T3 (oc)	100	134	182	230
Manometer head (cm)	8.3	7.9	7.4	7
Engine jacket water temp				
Inlet (T1) (oc)	36	37	36	37
Outlet (T2) (oc)	41	44	50	53

2.5. Experimental Calculation:

a) Torque (T) = (8*9.8*0.24) = 18.8352 N-m

b) Brake Power = 2*3.14* N *T/ 60000 = 2*3.14*650 *18.835 60000 = 1.28 kW

c) Heat Input (HI) = 20*Specific Gravity *CV /t*1000 = HI = 20* 0.83 *43000/144*1000 = 4.95 kW

(C.V = Calorific value of diesel = 43000 kJ/kg, SP. Gravity of diesel = 0.83)

d) Heat equivalent of useful work done = HUE = B.P kW = 1.28 kW
e) Heat carried away by water = HW = m1*CPW* (T2-T1) kW = 0.067*4.187*7 = 1.95 kW
f) Volume of air at R.T.P (Va) = A*Cd*√ gH

Air head (H) = (h2 –h1)*10-2 *ρw/ ρa

ρa (RTP) = ρa (NTP) *273 /273+ room temperature(*C) = 1.293*273/273 +35= 1.146kg/m3

H= 0.079*1000 /1.1 46= 68.93 m

Va = 3.14*(0.02)^2 *0.6 √ 2*9.81 *68.93/ 4 = 0.007043 = 7.043*10^3 m3/sec.

g) Mass of air consumed = m2 = P Va /RT = 1.01325*10^5*7.043*103/287*308 =0.00807kg/sec = 8.073*103kg/sec

h) Mass of fuel consumed = (m3) = 20* Specific Gravity/ t*1000 kg/sec. = 20* 0 0.83/144*1000 = 0.1153*10-3 kg/sec

i) Mass of exhaust gas, m4 = (m2 + m3) Kg/Sec = 8.073*10^-3 + 0.1153*10^-3 = 8.1883*10^-3 kg/sec

j) Heat carried away by exhaust gas, (HG) = m4 *CPG * (T3-Ta) KJ / sec. = 8.1883*10^-3*1.005*(134-35) = 0.8146 KW.

K) Heat unaccounted loss, (HuN)

(HuN) air flow = HI – (HU + HW + (HG) airflow) kW = 4.95 – (0.8146+ 1.95 + 1.28) = 0.9054 kW.

Table.2.Heat Balance Sheet

Load	H.I.		H.U.		H.W.		(Hg) air flow		Hun air flow	
(kg)	kW %	%	kW %	%	kW %	%	kW %	%	kW %	%
0	3.49	100	0	0	1.612	46.189	0.545	15.61	1.33	38.108
¼	4.95	100	1.28	25.85	1.95	39.39	0.8146	16.450	0.9054	18.29
½	7.435	100	2.72	36.58	2.6378	35.47	1.073	14.43	1.0042	13.51
¾	8.9225	100	3.84	43.04	2.487	27.87	1.5298	17.14	1.0657	11.94

2.6. Model Calculation:

Te = Evaporator Temp. = 2°C
Tc = Condenser Temp. = 40°C
TG = Generator Temp. = 60°C
Ta = Absorber Temp. = 40°C
Pe = Pa = P9 = P10 = P1 = P6 = Saturation pressure of NH3 at 2°C = 4.62 bar
Pc = PG = P2 = P3 = P4 =P5 = P7 = P8 = Saturation pressure of NH3 at 40°C = 15.54 bar
Assuming capacity of air conditioner = 0.25 TR
Heat removed = Qe = mr*(h10-h9)
mr = 0. 5 *3.5/ (h 0-h9)
From refrigeration table for NH3 (R-717)
h10 = hg at 2°C = 1446.54 kJ/kg
h9 = hf at 40°C = 371.93 kJ/kg
mr = 8.14*10-4 kg/sec

2.6.1. Generator: Balancing energy across the generator

Qg + Q3 = Q7 + Q4 (1)
Q3 =m3rh3
Q4 = m4rh4
Q7 = m7rh7
Qg = m7rh7 + m4rh4 - m3rh3
Balancing Concentration
m3rX3 = m4rX4
m3r = m4 *X4/X3
m1r = m2r = m3r and m4r = m5r = m6r
m1r = m6 *Xw/Xs (2)
Also, across absorber- balancing mass of NH3
m6r + m10r = m1r (3)
m6r = m1r -m10r
from (2) m6r = m6* Xw/X s = m10r
m6r(1- Xw/X6) = -m10r
m6r = -m 10r /(1 – Xw/X6) (4)
Qg = m3rh3 + m4rh4 – m7rh7

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From P-T-ξ diagram for aqua-ammonia solution:
Xw = 0.42 at absorber outlet conditions
Xs = 0.60 at generator outlet conditions
m10r = m7r = m8r = m9r = mr = mass flow rate of refrigerant across evaporator, condenser, expansion valve respectively.
m10r = m7r = m8r = m9r = mr = 8.14*10-4 kg/sec.
using (4): m6r = 1.89*10-3 kg/sec
using (3) : m1r = 2.704*10-3 kg/sec

2.6.2. Condenser

Qc + Q8 = Q7
Q7 = m7rh7
Q8 =m8rh8
From refrigeration table for NH3:
h7 = hg at 40°C
h8 = hf at 40°C
Qc = Q7 –Q8 = m10r*(h7-h8)
Qc = 0.814*10-3*(1473.30-371.93) = 0.897 kW

2.6.3. Expansion Valve

h8 = h9 = 371.93 kJ/kg
m8r = m9r

2.6.4. Evaporator: By energy conservation across evaporator:
Qe + Q9 = Q10
Qe = Q10-Q9
Q9 = m9rh9
Q10 = m10rh10
h9 = h8
h10 = hg at 2°C
Qe = m10rh10-m9rh9= 0.814*10-3*(1446.54-371.93) = 0.875kW

2.6.5. Absorber: By energy conservation across absorber: Qa + Q1 = Q6 + Q10
Q1 =m1rh1
Q6 = m6rh6
Q10 = m10rh10
Qa = m6rh6 + m10rh10 - m1rh1

2.6.6. Heat Exchanger: We use counter flow heat exchangers.
T1r = T2r = 40°C T4r = 60°C
Assuming Effectiveness (ε) = Q act/Q max = T4r - T5r /T4r – T2r = 0.7
Now, 0.7= 60-T5/ 60-40
T5 = 46°C
T6 = T5 = 46°C
Further, amount of heat gained by cold fluid = amount of heat lost by hot fluid
Qh = mh*Cph*(T4r – T5r)
Qc = mc*Cpc*(T3r – T2r)
Where, mh = mass flow rate of hot fluid.
mc = mass flow rate of cold fluid.
Cph = heat capacity of hot fluid.
Cpc = heat capacity of cold fluid.
mh*Cph*(T4r – T5r) = mc*Cpc*(T3r – T2r)
mh /mc= T3r –T2r /T4r -T5r
1.89 10^-3/2 .704 10^-3 = (T3r -40)/ (60-46)
T3r = 49.785°C

Table.3.Conditions at Various States

States	P (bar)	H (kj/kg)	T(*C)	M(kg/sec)	X %
1	4.62	60	40	2.708*10^-3	42
2	15.54	420	40	2.704*10^-3	42
3	15.54	270	49.78	2.704*10^-3	42
4	15.54	240	60	1.89*10^-3	60
5	15.54	240	46	1.89*10^-3	60
6	4.62	240	46	1.89*10^-3	60
7	15.54	1474.33	60	0.814*10^-3	-
8	15.54	396.81	40	0.814*10^-3	-
9	4.62	396.81	2	0.814*10^-3	-
1.	4.62	1446.54	2	0.814*10^-3	-

Table.4.Energy Flow Of various Components

Description	Notations	Calculated values (kw)
Heat load on evapourator	Qe	0.875
Heat load on condenser	Qc	0.897
Heat load on genrator	Qg	1.49
Heat load on absorber	Qa	1.469
Coefficent of performance	COP=Qe/Qg	0.587

3. RESULTS AND DISCUSSION

The experiment is performed on the 0.25TR ammonia-water vapor absorption refrigeration system and various sets of observations were recorded. These set of observations include the temperature readings at the various locations of the ammonia-water VAR system. The temperature readings include the generator temperature, condenser temperature, evaporator temperature, absorber temperature and temperature of the air conditioned space. The temperatures were measured and recorded using a digital thermometer and are shown in Tables below. Initial temperature of space to be cooled is 32°C. As the engine runs continuously the exhaust temperature increases proportionally. Higher the rate of increase of the exhaust temperature, faster would be the rate of fall of refrigerated temperature depending on the heat transfer media.

Table.5.Temperatures noted at various time intervals

Parts	Temperature measured after.....						
	5 min.	10 min.	20 min.	30 min.	40 min.	50 min.	60 min.
Engine Exhaust	60	85	106	150	180	200	210
Generator	35	40	45	50	54	56	56
Condenser	32	32.5	33	32.5	33	33.2	33
Evaporator	31.8	31.2	30.6	27.3	26	24.9	23
Absorber	30	30.8	31.5	32	32	31.8	31.5
Air Conditioned Space	32	32	31.8	30	28.6	26.2	24

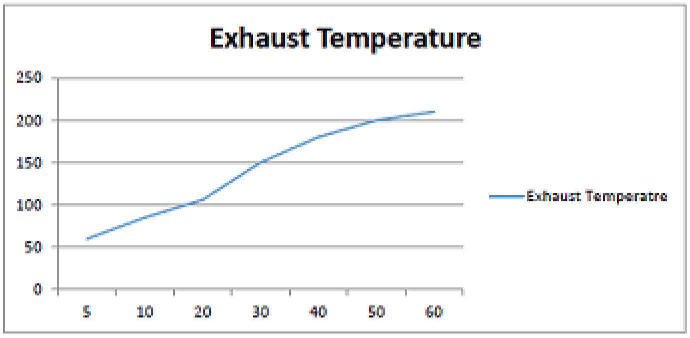

Fig.3.Exhaust temperature vs Time

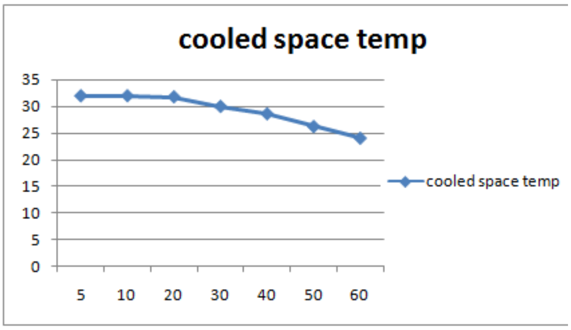

Fig.4.Cold space temperature vs Time

As the engine runs continuously the exhaust temperature increases proportionally. Higher the rate of increase of the exhaust temperature, faster would be the rate of fall of refrigerated temperature depending on the heat transfer media.

4. CONCLUSION

A diffusion vapor absorption system was bought and was attached to the engine exhaust pipe to analyze the performance of the system i.e to cool down the temperature of refrigerating space of the vehicle. The system is tested

with heat input from the exhaust of a 100cc engine and an electric heating element and the following conclusions are made:
a) The amount of waste heat available at the engine exhaust is comparable to the heat required at the generator. Thus, it can be used to provide heat for the purpose of air conditioning. If any extra heat is required, it can be obtained from the engine cooling water.
b) As the absorber temperature increases, the solubility of NH3 in water decreases hence less amount of NH3 vapors would be formed in the generator which would result in decreased cooling effect. Consequently the COP of the system decreases.
c) The COP is found to be 0.587 and this is because of higher generator temperature. A more efficient thermal system should have higher COP and lower total entropy generation.
d) The system uses no special heat transfer equipment i.e the exhaust pipe and the generator pipe are in contact with each other externally and is covered by insulating material. With better gas to gas heat exchangers higher generator temperatures can be obtained.
e) The COP can be increased further by using a heat exchanger between the absorber and generator. As the effectiveness of heat exchanger increases, COP of the system also increases.
f) The required cooling effect is attained by running the engine (100cc) for a minimum time period of 60 minutes. This is relatively higher than the present system and is a disadvantage.
g) As the absorber temperature increases, the solubility of NH3 in water decreases hence less amount of NH3 vapors would be formed in the generator which would result in decreased cooling effect. Consequently the COP of the system decreases.
h) As we increase Absorber Temperatures, the COP of the system decreases. This is because, as temperature of solution increases the solubility of NH3 in water decreases. Thus, the amount of vapors formed in the generator decreases which results in decrease in cooling effect.

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