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A Study on Automobile Air-Conditioning Based on Absorption Refrigeration System Using Exhaust Heat of a Vehicle

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ABSTRACT – Energy from an exhaust of an internal combustion engine is used to power an absorption refrigeration systemto air-condition an ordinary passenger vehicle. Feasibility study has been done to find out the energy available from exhaust gas of a vehicle. Cooling load for the automobile has been estimated. In this paper theoretical evaluation ofLiBr-Water based absorption refrigeration system is presented. Mathematical modeling of system using EES software is done, Alsoeffects on COP of system with change in different parameters has been studied.

Keywords:Automobile Exhaust, Absorption Refrigeration System, Internal Combustion Engine, EES

INTRODUCTION

In vapour absorption refrigeration system, aphysiochemical process replaces themechanical process of the vapour compression system by using energy in the form of heat rather than mechanical ork. Themain advantage of this system lies in possibility of utilizing energy from exhaust a sofvehicle and alsousinganeco-friendly refrigerant such as water. The vapour absorption system has many favorable characteristics; typically a much smaller electrical input is required to drive the solution pump as compared to the power requirement of the compressor in the vapour compression system. Also,fewermovingpartsmeanlower noise level, higher reliability and improved durabilityin vapourabsorptionsystem.

METHODOLOGY

In vapour absorption refrigeration system as shown in FIG1, the compressor is replaced by an absorber, a pump, a generator anda pressure reducing valve. These components in the system perform thes ame function as that of compressor in VCR system. The vapour refrigerated from evaporator is drawn into the absorber where it is absorbed by the weak solution of refrigerant forming astrong solution. This strong solution is pumped to the generator where it is heated utilizing exhaust heat of vehicle. During the heating process the vapour refrigerant is driven off by the solution and enters into the condenser where it is liquefied. The liquid defrigerant then flows into thee vaporator and the cycle is completed.

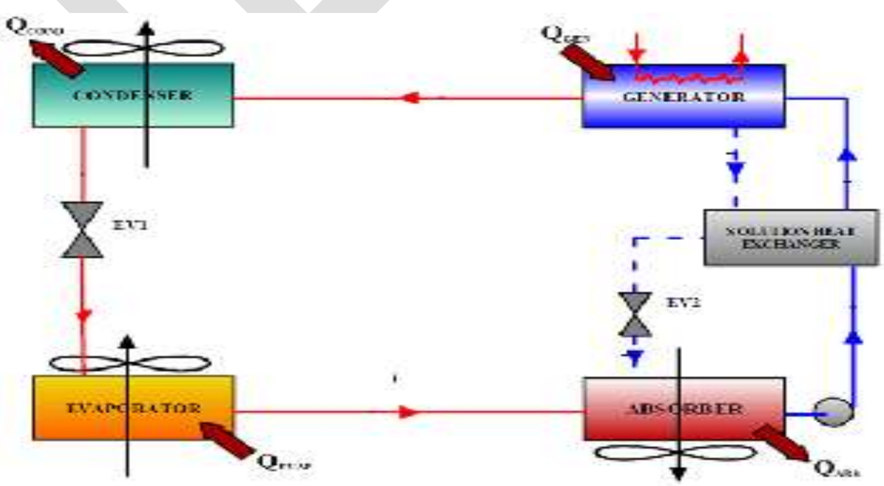


FIG [1]

MEASURED EXHAUST USEFUL HEAT AND HEAT LOAD CALCULATION

To generate base line data, the engine is allowed to run at different throttle position (one-fourth and half) considering engine speed as running parameter. The mass flow rate of air, mass flow rate of fuel and temperature of exhaust gas is measured as given in Table 1 For measuring the required data plenum chamber (1 m^3) with circular orifice of 32 mm diameter, inclined tube manometer, burette for petrol measurement and thermocouple for exhaust temperature measurement installed on engine. The determination of actual load becomes very difficult in vehicle air conditioning because of the variation of the load in the climatic conditions when the vehicle is exposed during the course of long journey. The cooling load of a typical automobile is also considered at steady state conditions. The cooling capacity is affected by outdoor infiltration into vehicle and heat gain through panels, roofs, floors etc. The cooling load considered in this analysis is given in Table 2. The table shows that heat load inside the traveler is 2 kW. Therefore, 2 KW air conditioning unitis sufficient to fulfill the cooling.

Throttle position opening	Eng. Speed (rpm)	Air Pr. (mm of H_2O)	Time for cons. of 25cc of fuel(sec)	Exh. Temp. (^oC)	Mass of fuel (kg/s x 10^{-5})	Mass of air (kg/s x10^{-4})	Exh. useful energy (KW)
	3500	7.4	40	622	46	64	3.98
	3000	7.9	57	605	32	67	3.91
1/4	2500	7.2	48	566	38	64	3.50
	2000	5.6	42	623	44	56	3.49
	1500	4.9	41	502	45	52	3.05
	3500	14.8	34	669	57	91	6.02
	3000	15.9	29	615	63	94	5.74
Half	2500	12.3	24	648	71	83	5.47
	2000	9.4	32	595	57	73	4.51
	1500	6.8	39	508	47	62	3.61

TABLE [1]

Heat load inside the vehicle is calculated as follows:

We have considered passengers in the traveler and calculated the following:-

- Radiation Load
  $Q_{rad} = \sum S*\tau*I_{rad}*\cos\theta$

- Ambient Load
  $Q_{amb} = \sum S*U*(T_s - T_i)$

- Ventilation Load
  $Q_{ven} = m_{ven}* (e_o - e_i)$

- Metabolic Load
  $Q_{meta} = \sum M*A$

- Overall Heat Load
  $Q_{AC} = (Q_{rad} + Q_{amb} + Q_{ven} + Q_{meta})$

Heat Load	Amount of Heat(KJ/hr)
Radiation Load	85.83
Ambient Load	422.83
Ventilation Load	59.54
Metabolic Load	1356.23
Total	1924.43(KJ/hr) or 1.9 Kw

TABLE [2]

MODELLING OF ABSORPTION SYSTEM

Followingassumptionhasbeenmadetomodelthesystem.

1. Generator and condenser as well as evaporator and absorber are under same pressure.
2. There are no pressure changes except through the flowrestrictors and the pump.
3. Refrigerant vapor leaving the evaporator is saturated pure water.
4. Liquid refrigerant leaving the condenser is saturated.
5. Strong solution leaving the generator is boiling.
6. Weak solution leaving the absorber is saturated.
7. No liquid carryover from evaporator.
8. Flow restrictors are adiabatic.
9. Pump is isentropic.
10. No jacket heat losses

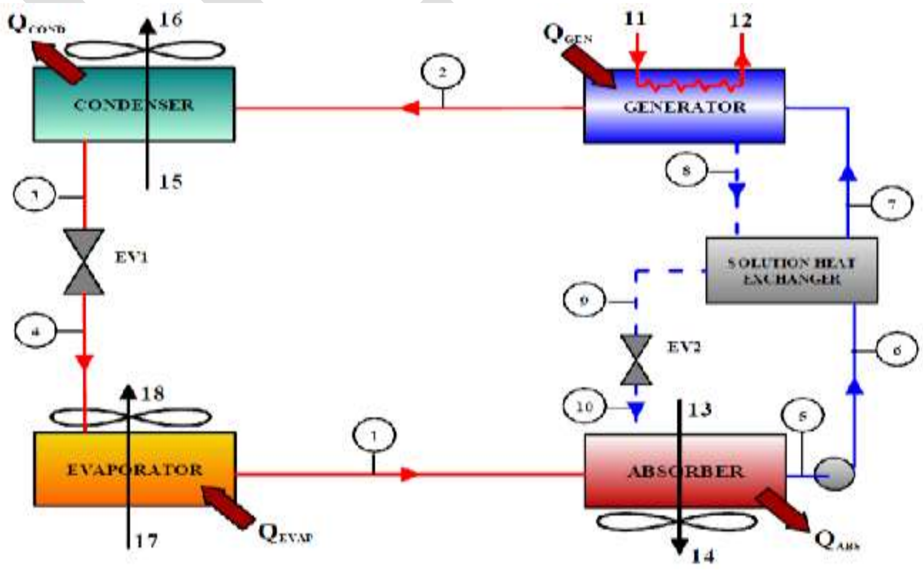


FIG [2]

- 1st point is saturated water vapor;
- 2nd point is superheated water vapor;
- 3rd point is saturated liquid water;
- 4th point is vapor-liquid water state;
- 5th point is saturated liquid solution;
- 6th point is sub-cooled liquid solution (at P_{low});
- 7th point is sub-cooled liquid solution (at P_{high});
- 8th point is saturated liquid solution;
- 9th point is sub-cooled liquid solution;
- 10th point is vapor-liquid solution state.

2 KW Aqueous Lithium Bromide Absorption System

Assumptions Taken:-
Condenser Temperature = 38^o C
Evaporator Temperature = 7^0 C
Absorber Temperature = 37^0 C
Generator Temperature = 85^0 C
Pressure values taken from p-h chart of water as refrigerant for condensing temperature 35^0 C and evaporating temperature 7^0 C
P_E = 1 KPa
P_C = 5.696 KPa

1. For Evaporator

Process Cycle 4-1
Heat load on Evaporator $Q_E = 2KW$
$Q_E = m_R (h_1 – h_4)$
For Defined System
$m_R = m_1 = m_4 = 0.000844$ *Kg/Sec*

2. For Generator

Process Cycle 7-2
Mass Balancing Of Weak and Strong Solution
$m_7 = m_2 + m_8$
$m_7x_7 = m_8x_8$

$m_7 = 0.0101$ *Kg/Sec*
$m_8 = 0.00928$ *Kg/Sec*
$m_2 = 0.000844$ *Kg/Sec*

Heat Input To Generator $Q_g = m_2h_2 + m_8h_8 - m_7h_7$
$Q_g = 0.0909*m_8*h_2 + m_8*h_8 – 1.0909*m_8*h_7$

$\mathbf{Q_g = 2.725\ KW}$

For Defined System
$m_8 = m_9 = m_{10} = 0.00928$ *Kg/Sec*
$m_7 = m_6 = m_5 = 0.01010$ *Kg/Sec*
$m_2 = m_3 = m_4 = m_1 = 0.000844$ *Kg/Sec*

3. For Condenser

Process Cycle 2-3
Heat Rejected by Condenser $Q_c = m_2*(h_2 – h_3)$

Q_c = 2.113 KW

4. For Absorber

Process Cycle 1-5
Heat Rejected by Absorber $Q_a = m_1h_1 + m_{10}h_{10} - m_5h_5$

Q_a = 2.567 KW

5. For Solution Heat Exchanger

Process Cycle 6, 9 - 7, 8

Heat transfer $Q_{SHEX} = m_5*(h_7 - h_6)$

Q_{SHEX} = 0.416 KW

SYSTEM ANALYSIS

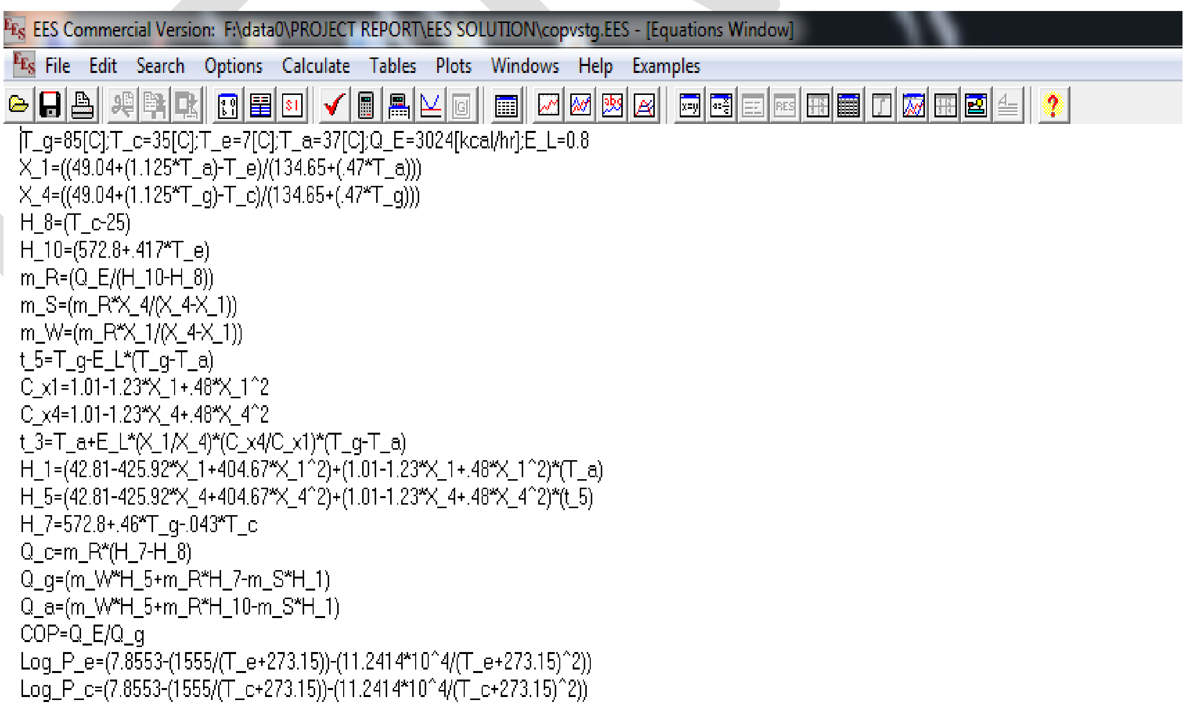
System analysis is based on certain fixed parameters which are shown in **TableNo.3**by using this fixed parameters COP, Mass flow rate of refrigerant, mass flow rate of strong solution, mass flow rate of weak solution, heat transfer in generator, condenser and absorber are found out using EES software and the effect of generator temperature, evaporator temperature, condenser temperature and absorber temperature on system COP is analysed using EES software.

INPUT PARAMETERS

T_g = Generator Temperature (°C)	85°C
T_e = Evaporator Temperature(°C)	7°C
T_c = Condenser Temperature(°C)	35°C
T_a = Absorber Temperature(°C)	37°C
Q_e = Load $(kcal/hr)$	1720$(kcal/hr)$

Table No.3

EES PROGRAMM

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T_g=85[C];T_c=35[C];T_e=7[C];T_a=37[C];Q_E=3024[kcal/hr];E_L=0.8
X_1=((49.04+(1.125*T_a)-T_e)/(134.65+(.47*T_a)))
X_4=((49.04+(1.125*T_g)-T_c)/(134.65+(.47*T_g)))
H_8=(T_c-25)
H_10=(572.8+.417*T_e)
m_R=(Q_E/(H_10-H_8))
m_S=(m_R*X_4/(X_4-X_1))
m_W=(m_R*X_1/(X_4-X_1))
t_5=T_g-E_L*(T_g-T_a)
C_x1=1.01-1.23*X_1+.48*X_1^2
C_x4=1.01-1.23*X_4+.48*X_4^2
t_3=T_a+E_L*(X_1/X_4)*(C_x4/C_x1)*(T_g-T_a)
H_1=(42.81-425.92*X_1+404.67*X_1^2)+(1.01-1.23*X_1+.48*X_1^2)*(T_a)
H_5=(42.81-425.92*X_4+404.67*X_4^2)+(1.01-1.23*X_4+.48*X_4^2)*(t_5)
H_7=572.8+.46*T_g-.043*T_c
Q_c=m_R*(H_7-H_8)
Q_g=(m_W*H_5+m_R*H_7-m_S*H_1)
Q_a=(m_W*H_5+m_R*H_10-m_S*H_1)
COP=Q_E/Q_g
Log_P_e=(7.8553-(1555/(T_e+273.15))-(11.2414*10^4/(T_e+273.15)^2))
Log_P_c=(7.8553-(1555/(T_c+273.15))-(11.2414*10^4/(T_c+273.15)^2))
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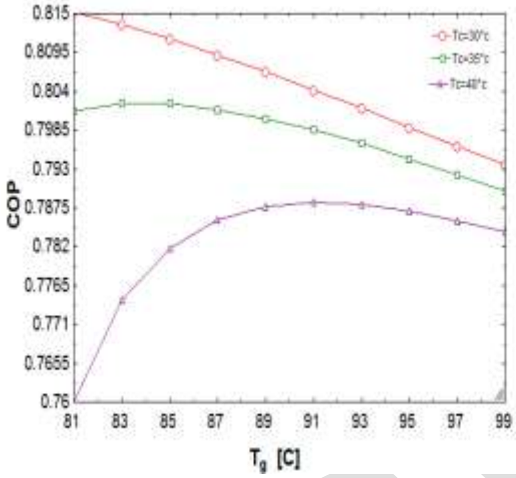
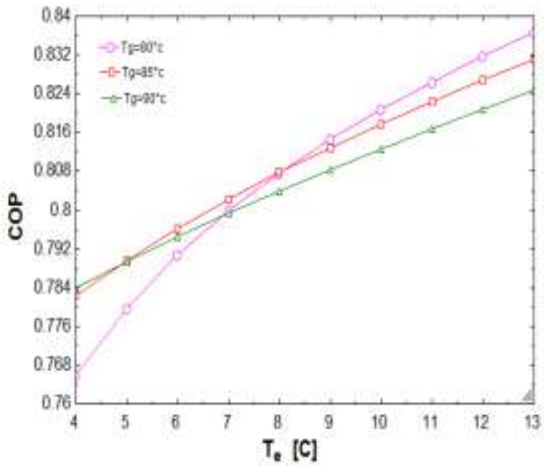
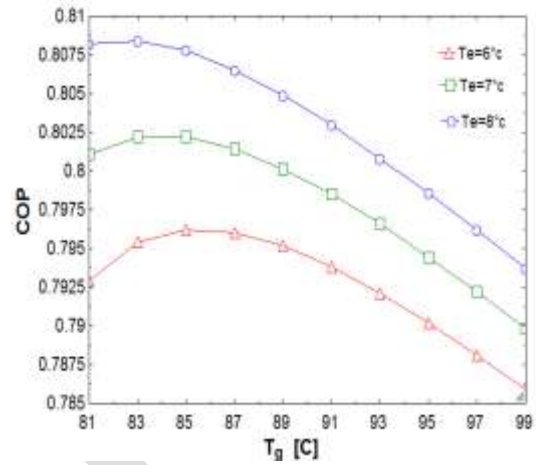
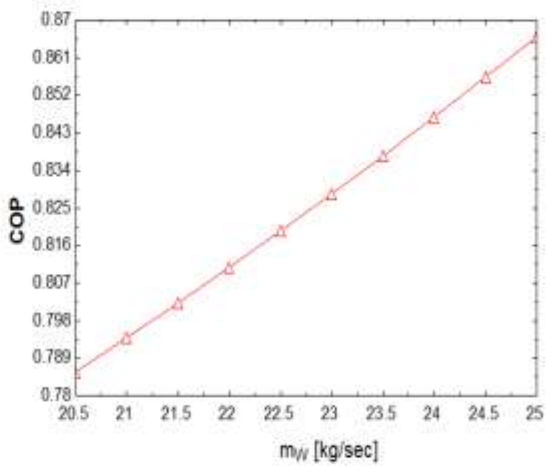
EES Commercial Version: F:\data\?\PROJECT REPORT\EES SOLU

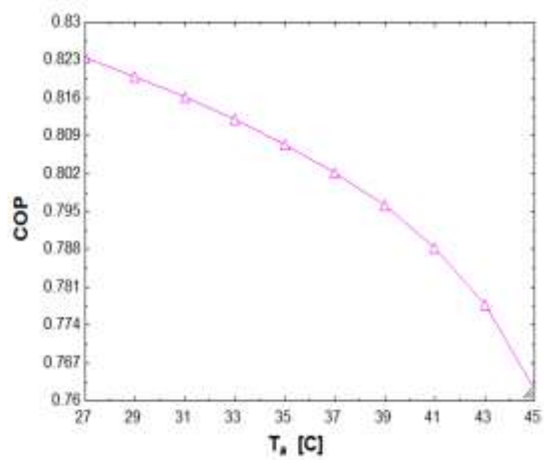
	COP	T_c [C]	T_g [C]
Run 1	0.8152	30	81
Run 2	0.8134	30	83
Run 3	0.8114	30	85
Run 4	0.8091	30	87
Run 5	0.8067	30	89
Run 6	0.8042	30	91
Run 7	0.8016	30	93
Run 8	0.7989	30	95
Run 9	0.7962	30	97
Run 10	0.7935	30	99











CONCLUSION

As per the calculations of heat load and heat availability obtained from a vehicle a 2kW system is feasible to provide air conditioning in a vehicle. From system analysis it is seen that COP of system increases with increase in generator temperature and evaporator temperature but it reduces with increase in condenser and absorber temperature. There is optimum value of generator temperature above which COP reduces also COP increases with increase in mass flow rate of water (m_w).

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