SIMULATION OF ABSORPTION REFRIGERATION SYSTEM FOR AUTOMOBILE APPLICATION

by

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An automotive air-conditioning system based on absorption refrigeration cycle has been simulated. This waste heat driven vapor absorption refrigeration system is one alternate to the currently used vapour compression refrigeration system for automotive air-conditioning. Performance analysis of vapor absorption refrigeration system has been done by developing a steady-state simulation model to find the limitation of the proposed system. The water-lithium bromide pair is used as a working mixture for its favorable thermodynamic and transport properties compared to the conventional refrigerants utilized in vapor compression refrigeration applications. The pump power required for the proposed vapor absorption refrigeration system was found lesser than the power required to operate the compressor used in the conventional vapor compression refrigeration system. A possible arrangement of the absorption system for automobile application is proposed.

Key words: *vapour absorption system, air conditioning, water-lithium bromide*

Introduction

Comfort air-conditioning as we know it today has been growing at such a rate that it would no longer entertain remarks on it as a prohibitive luxury as was considered ever since the first refrigerated automobile air-conditioning came to existence. Conventionally, automobile vehicles use vapor compression refrigeration (VCR) technology to provide comfort air-conditioning, in which the compressor energy required is obtained from the internal combustion (IC) engine of the vehicle or by providing a IC engine dedicated for that purpose. Though automotive air-conditioning system has undergone tremendous improvements in design, performance and efficiency as a result of continuous efforts put for the enhancement of every individual component of the system, there had been no success to find an alternate technology that could reign supreme over the predominantly used VCR system posing serious problems that are harder to address. Also the search for eco friendly refrigerant has not been so successful with compromises had to be made in one or more factors in the choice of a refrigerant for a particular application.

Apart from the need to replace halogenated hydrocarbons that contribute to ozone depletion potential (ODP) and green house warming, a system, that, ensures vehicle performance, reduces noise, is easy to maintain and highly reliable was needed for a very long time. Efforts so far taken to utilize the waste engine heat for cooling shows promise but no one have come with an operative design with no prototypes built for those designs. VCR systems are highly prone to refrigerant leakage issues due to their inherent moving components, whereas a vapor absorption refrigeration (VAR) system would prove itself to operate for long periods of time with minimal operational and maintenance related issues compared to a VCR system. Also the VAR systems operated using water as refrigerants shows extensive adaptability towards zero ODP. Hence a VAR system could well address the issues related to the conservation of environment and global warming.

Investigation on the performance of an IC engine on introducing the VAR system and utilization of exhaust gases of an IC engine to drive a VAR system into the exhaust system had been done by conducting range of experiments on a Ford 150 (Dover) fuel injection, 6 liter, turbo diesel engine using different combinations of speed and torque to obtain a range of engine power output [1]. Several ways and means were suggested to counter the reduction in exhaust gas flow in slow running traffic or stationary situations or when vehicle is parked and cooling is still required, since lesser the heat content possessed by the exhaust gas, lesser would be the cooling effect. The experimental results proved that 6 liter turbo diesel engine was capable of providing enough energy to drive the VAR system via its waste exhaust heat.

A study on VAR based automotive air-conditioning system was done in another work and a proposal for a possible arrangement of the VAR for this purpose had been advocated. Investigation on the suitability of such a system had been done taking operating conditions and other standard requirements in to account. Finally the particular features of the proposed scheme and locates a border line in the power-velocity diagram had been pointed out. Measurements were carried out on a four cylinder spark ignition to estimate the amount of heat recoverable, in particular, when the engine runs at low loads [2, 3]. A breadboard prototype of an absorption system was designed, built and tested for truck refrigeration using heat from the exhaust gases. Further simulation of the system which included cycle analysis and component modeling and used the test data to validate the model was carried out. Analysis of the amount of recoverable energy of the exhaust gases for truck driving conditions on city traffic, mountain roads, and flat roads was also done [4]. In this study, to drive the absorption refrigeration system, the heat from the exhaust gases was considered to cater the heat source for the desorber rather than using the engine cooling water as heat source due to temperature requirements to operate the cycle.

Scope of the present work

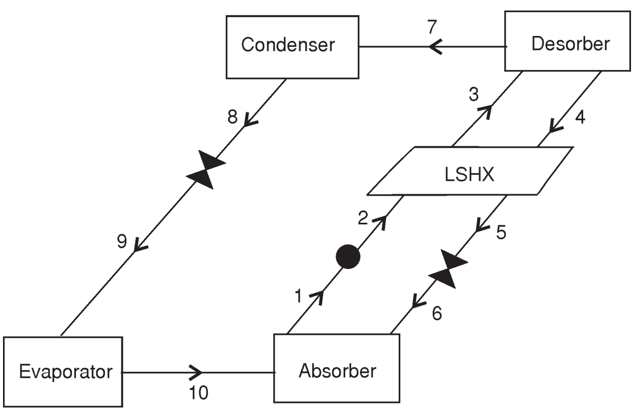

Figure 1. Single effect water-lithium bromide absorption cycle
LSHX – liquid side heat exchanger

Absorption refrigeration technology could address these problems, which, uses eco friendly refrigerants and is driven by waste heat freely available at the exhaust system of the vehicle. The only deciding factor for the success of this proposal could be the amount of power that could be recovered from the vehicle engine, which has to be sufficient enough for providing comfort air-conditioning at all speeds *i. e.* slow speed, cruise speed, and idle conditions of the vehicle.

A single effect VAR cycle using water-lithium bromide as the working fluid is shown in fig. 1. The choice of having water/lithium bromide as the working fluid is based on its high thermal and chemical stability, low toxicity, low heat of mixing, *etc*. In this VAR cycle the compressor of the VCR is replaced by components, absorber and desorber to elevate the pressure level of the binary mixture of refrigerant and absorbent. Here in this case water being the refrigerant and lithium bromide the absorbent. The weak solution entering the desorber is heated by an external source and the refrigerant vapour gets driven off and enters the condenser as saturated vapour and gets condensed, throttles there after and enters the evaporator to facilitate removal of heat from the space to be cooled. The pressure difference between the evaporator and absorber causes the vapour to get transported to the absorber where it gets absorbed into the strong solution entering the absorber from desorber through the flow restrictor.

In the absorber the heat of absorption is released to the ambient. The weak solution is then pumped to the desorber by a mechanical pump, through a heat exchanger where heat exchange between the weak and strong solutions takes place. Thus the cycle continues. The experiment results of Boatto *et al*. [2, 3] has proved the fact that as the speed of the motor increases there is an corresponding increase in fuel and exhaust power at constant engine power and that the same being observed when the engine power increases and with constant motor speed. In an IC engine, the fuel power is seen to be distributed as 30% engine power, 40% exhaust power and 30% coolant power. Table 1 shows that at 1300 rpm the fuel power and exhaust power are at 30 kW and 5 kW, respectively, when the engine power is 10 kW and the coolant power is 15 kW as the fuel power is equal to the engine power, exhaust power, and coolant power. Similarly at 2000 rpm engine power has been found to be 24 kW, with the fuel power, exhaust and coolant power at 70, 17, and 29 kW, respectively. Therefore the availability of heat that is recoverable from the exhaust system of an IC engine is in the range of 5 to 17 kW as the speed of the motor increases from 1300 to 2000 rpm.

Table 1. Summary of the experimental results of work of Boatto *et al.* [2, 3]

Row	Engine speed	Engine power	Exhaust power
1	1300	10	5
2	1500	14	8
3	1700	18	11
4	1900	22	15
5	2000	24	17

The reference cycle data has been given in tab. 3 as the thermodynamic analysis of a VAR for automotive air-air-conditioning. The coefficient of performance (*COP*) of the system is found to be 0.77 which is reasonably good. The results of a thermodynamic analysis of a typical R134a vapor compression system in a mid sized passenger car presented by Boatto *et al.* [2, 3] is shown in tab. 2.

Table 2. Thermodynamic analysis of vapor compression automotive air-conditioning system presented by Boatto *et al.* [2, 3]

P_c [kPa]	P_e [kPa]	T_c [°C]	T_e [°C]	rpm	m_{ref} [kg/s]	Disp. [m^3/rev]	W_c [kW]	Q_{con} [kW]	Q_e [kW]	*COP*
1574	314	57	0	1000	0.027	$1.37 \cdot 10^{-4}$	1.1	4.21	3.1	2.8
1574	314	57	0	2000	0.054	$1.37 \cdot 10^{-4}$	2.2	8.42	6.21	2.8

The VCR system installed in the mid sized passenger has designed to provide the cooling power of 2 kW at both idle and cruise conditions. The result shows that VCR system requires 1.1 to 2.2 kW power during idle to cruise speed conditions. On comparing the results of the thermodynamic analysis of both cycles, requirement of power at all conditions of the vehicle engine is negligible in case of a VAR.

Mathematical model and thermodynamic analysis for the VAR system

The simulation procedure involves mathematical modeling of each component of the H_2O-LiBr absorption refrigeration system. The overall system performance is evaluated by combining these models under usual sequence of operation of the simulated system. Generally made conditions and assumptions, that the system is simulated under steady-state conditions, there is negligible pressure drop in the pipes, there is no heat transfer except at the four major components, the expansion process of the expansion device is at constant enthalpy and that the components effectiveness is at predetermined value were incorporated in the model to simplify analysis. To perform a thermodynamic analysis of an absorption refrigeration system, the conservation of mass and the first law of thermodynamics were applied to the individual components of the system. Both mass balance of the total mass and mass balance for each material species were included in the mass conservation study. Table 3 shows the mass and energy balances for the absorption cycle. The subscript in the equations denotes the various state points of the cycle.

Table 3. Thermodynamic analysis of vapor absorption cycle for automotive air-conditioning

P_c [kPa]	P_e [kPa]	T_c [°C]	T_e [°C]	T_d [°C]	m_{ref} [kg/s]	m_{ws} [kg/s]	m_{ss} [kg/s]	Q_d [kW]	Q_c [kW]	Q_e [kW]	Q_a [kW]	*COP*
7.4	1.2	39	10	93	0.0008	0.005	0.0049	2.188	2.04	2	2.48	0.77

The *COP* of the absorption refrigeration system can be written as Q_e/Q_d. Here in this work, MATLAB common software has been used to do the simulation. It is easy to implement control strategies in this MATLAB software. For property data base EES software was used for the appropriate data for various state points and conditions encountered in the simulation procedure [5, 6].

Mass and energy balances for the vapour absorption cycle

Absorber

– Mass balance

$$m_{10} + m_6 = m_3 \tag{1}$$

$$m_{10}x_{10} + m_6x_6 = m_3x_3 \tag{2}$$

– Energy balance

$$m_{ref}\,h_{10} + m_{ss}h_6 = m_{ws} + h_1 + Q_a \tag{3}$$

Desorber

– Mass balance

$$m_3 = m_7 + m_4 \tag{4}$$

$$m_3x_3 = m_7x_7 + m_4x_4 \tag{5}$$

– Energy balance

$$m_{ws}h_3 + Q_d = m_4h_4 + m_7h_7 \tag{6}$$

Condenser

– Mass balance

$$m_7 = m_8 \tag{7}$$

– Energy balance

$$Q_c = m_{ref}(h_7 - h_8) \tag{8}$$

Evaporator

– Mass balance

$$m_{10} = m_9 \tag{9}$$

– Energy balance

$$Q_e = m_{ref}(h_{10} - h_9) \tag{10}$$

Solution heat exchanger

– Mass balance

$$m_4 = m_5 \tag{11}$$

$$x_4 = x_5 \tag{12}$$

$$m_2 = m_3 \tag{13}$$

$$x_2 = x_3 \tag{14}$$

– Energy balance

$$m_{ws}(h_3 - h_2) = m_{ss}(h_4 - h_5) \tag{15}$$

Pump

– Mass balance

$$m_1 = m_2 \tag{16}$$

$$x_1 = x_2 \tag{17}$$

– Energy balance

$$W = m_{ass}(h_1 - h_2) \tag{18}$$

Solution expansion valve

– Mass balance

$$m_5 = m_6 \tag{19}$$

$$x_5 = x_6 \tag{20}$$

– Energy balance

$$h_5 = h_6 \tag{21}$$

Refrigerant expansion valve

– Mass balance

$$m_8 = m_9 \tag{22}$$

– Energy balance

$$h_8 = h_9 \tag{23}$$

Proposal for the prototype and possible arrangement of the system in a vehicle

The schematic of the proposed setup for the absorption automotive air-conditioning is given in fig. 2. Direct heat recovery mode as suggested by Boatto *et al.* [2, 3] has been opted for in this scheme. A plenum is included in the scheme as suggested by Horuz (1998) to minimize

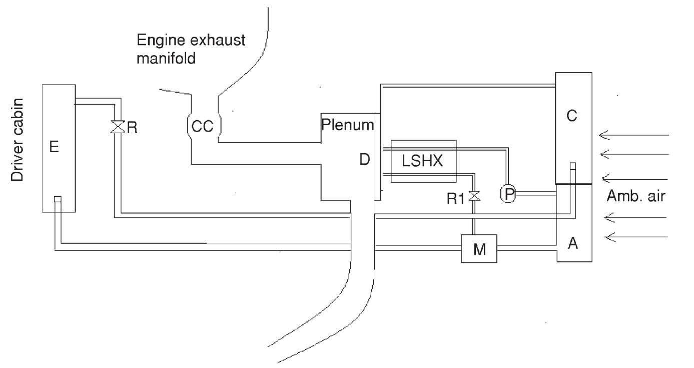

Figure 2. Schematic diagram of the proposed possible arrangement of the automotive absorption air-conditioning system
CC – catalytic converter, P – pump, D – desorber, M – mixer, A – absorber, E – evaporator, LSHX – liquid side heat exchanger, R and R1 – restrictors, C – condenser

the back pressure imposed on the vehicle. The heat exchanger types selected are fin and tube type for evaporator, condenser and absorber.

The internal heat exchanger and desorber will be of plate and tube heat exchanger type. Desorber which is a plate and tube heat exchanger is proposed to be kept inside the plenum to be constructed next to the catalytic converter of the engine exhaust system. The heat input to the desorber would driven off the refrigerant vapor to the condenser, which has to be placed in the position as is normally found in the case of conventional VCR system *i. e.* in front of the radiator. As it would ensure rejection of heat from the condenser to take place effectively. The condensed liquid refrigerant condenser passes through a flow restrictor R and enters the evaporator cooling coil placed near the driver's cabin. On taking the cooling load from the car's utility space the refrigerant vapour enters the absorber, which is a fin and tube heat exchange, to be also placed in front of the radiator, after getting mixed with the strong solution in the mixer.

After rejecting the heat of absorption to the ambient the weak solution is pumped to the desorber for the cycle to continue. The strong solution leaving the desorber enters the mixer after passing through a flow restrictor R1. The power required for the pump would be very meager which could be supplied from any source available in the vehicle.

Results and discussion

With the available lower generator heat, an analysis was performed to find the limitations of the proposed system at 1300 rpm engine speed condition. Using the available exhaust power range of 5 kW at 1300 rpm to 17 kW at 2000 rpm as observed by Boatto *et al.* [2, 3] the effect of operating conditions on the system was analyzed.

Effect of operating conditions on the system

The influence of generator heat on the performance of the other system components heat exchange rate is first studied. Figure 3 illustrates the heat exchange rates for evaporator, condenser and absorber for the range of varying generator heat input from 5 to 17 kW. The evaporator heat rate varies from 3.7 to 11 kW from 1300 to 2000 rpm. The condenser heat transfer rates also increase with engine speed from 3.8

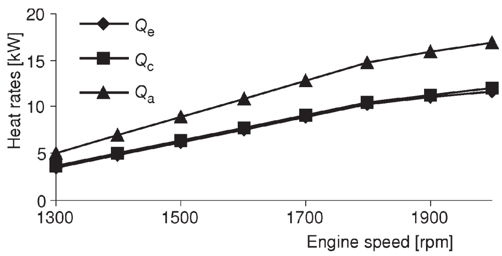

Figure 3. Variation of heat exchange rates of various components with the changing engine speed

to 11.2 kW and the absorber heat transfer rate varies from 4.3 to 16 kW. The increase in desorber heat increases the refrigerant flow rate hence this increase in heat transfer rates of components is observed.

Figure 4 illustrates the effects of varying absorber temperature on the other system components at a constant generator heat rate of 5 KW. From the figure it is evident that evaporator and condenser rates decrease when the absorber temperature is increased. The absorber heat rate shows slight increase when the absorber temperature is increased. The evaporator capacity decreases though at a slow rate.

The impact that the varying condenser temperature has over the system's other component's heat exchange rate was analyzed by varying the condenser temperature at the 1300 rpm engine condition *i. e.* desorber heat rate being at 5 kW. The results are illustrated in fig. 5. The evaporator heat varies from 4.3 to 4.15 kW and condenser heat rate from 4.4 to 4.15 kW whereas a slight increase in condenser heat rate is observed.

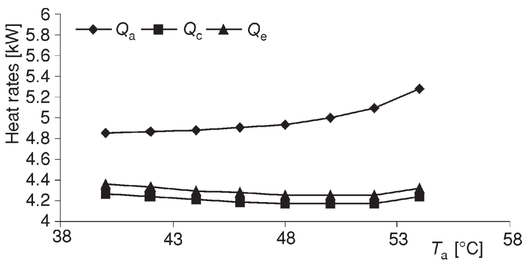

Figure 4. Variation of heat exchange rates of various components with the changing absorber temperature

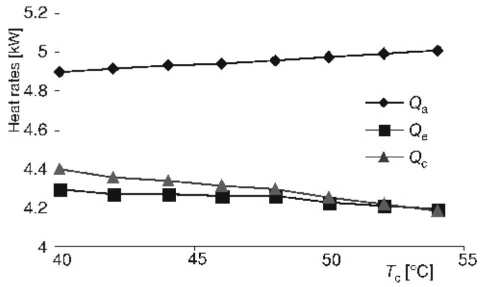

Figure 5. Variation of heat exchange rates of various components with the changing condenser temperature

Finally, the desorber temperature was varied to observe the corresponding changes in heat exchange rates of other system components. Figure 6 shows the heat exchange rates of each component as function of changing generator temperature. The results show that Evaporator capacity varies from 4.0 to 4.3 kW when the desorber temperature varies from 86 to 112 °C.

Figure 7 illustrates the coefficient of performance of the system at varying temperatures at absorber and desorber. From the results it is found that the temperature at which the heat is supplied has a significant influence over the system performance. Also as the desorber temperature is increased, the evaporator heat transfer rate declines and that greatly influence the performance of the system.

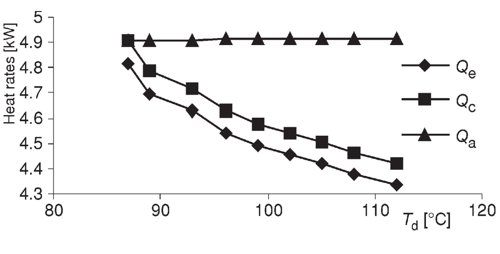

Figure 6. Variation of heat exchange rates of various components with the changing desorber temperature

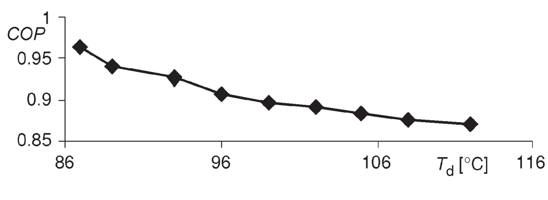

Figure 7. Variation in system performance for varying desorber temperature

Effect of heat exchangers on performance of the system

The influence of heat exchanger size on system performance is shown in figs. 8 to 11. In each figure the *COP* and the heat rate are plotted against the over all heat transfer coefficient *UA* of that particular heat exchanger. Similar trend is observed in all the cases. The *COP* decreases as *UA* decreases for small values of *UA* above which it is observed to be almost insensitive to the changes in *UA* values. Thus the results infer that the heat transfer rates of the system components are interrelated with each other and being significantly influenced by the operating temperature of the system and the heat exchanger performance.

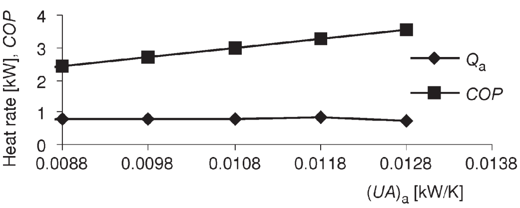

Figure 8. Variation in system performance and absorber capacity for varying absorber size

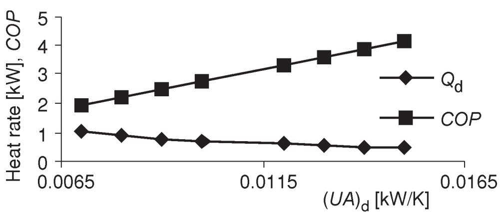

Figure 9. Variation in system performance and desorber capacity for varying desorber size

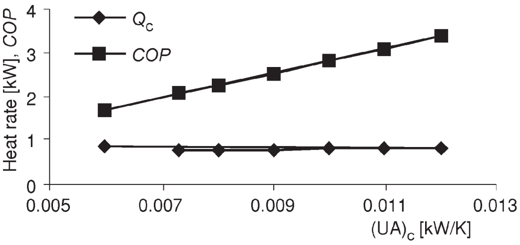

Figure 10. Variation in system performance and condenser capacity for varying condenser size

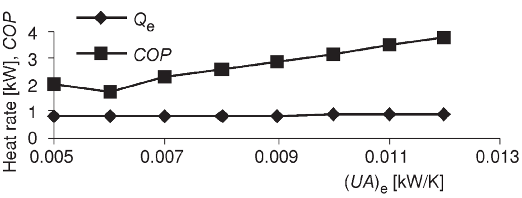

Figure 11. Variation in system performance and absorber capacity for varying evaporator size

Conclusions

With the thermodynamic analysis of VAR system for automotive air-conditioning a reference design data was arrived at for providing 2 kW at both idle and cruise speed conditions of the vehicle motor. The pump power required by the absorption refrigeration cycle was found to be meager when compared to the vapor compression cycle's power requirement to operate its compressor. Performance analysis has been done on the VAR for automotive air-conditioning at 1300 rpm (slow running conditions).

The results infer that the system performance varies with varying operating temperatures of the desorber. Also the heat transfer rates of the system components are interrelated with each other and being significantly influenced by the operating temperature of the system and the heat exchanger performance. Thus a similar analysis at idle condition of the vehicle would be necessary to predict the suitability of such a system to the automotive air-conditioning.

Nomenclature

COP	– coefficient of performance, [–]
Disp.	– compressor displace ment volume, [m^3/rev]
h	– enthalpy, [$kJkg^{-1}$]
m	– mass flow rate, [kgs^{-1}]
p	– pressure, [Pa]
Q	– rate of heat transfer, [kW]
T	– temperature, [°C]
UA	– over all heat transfer coefficient, [WK^{-1}]
W	– power, [kW]
x	– mass concentrations, [$kg_{LiBr}/kg_{solution}$]
x_1, x_2, x_3	– weak solution
x_4, x_5, x_6	– strong solution
x_7, x_8, x_9, x_{10}	– refrigerant concentration

Subscripts

a	– absorber
c	– condenser
comp	– compressor
d	– desorber
e	– evaporator
LiBr	– lithium bromide
ref	– refrigerator
ss	– strong solution
ws	– weak solution

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