A car air-conditioning system based on an absorption refrigeration cycle using energy from exhaust gas of an internal combustion engine

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Abstract

Energy from the exhaust gas of an internal combustion engine is used to power an absorption refrigeration system to air-condition an ordinary passenger car. The theoretical design is verified by a unit that is tested under both laboratory and road-test conditions. For the latter, the unit is installed in a Nissan 1400 truck and the results indicate a successful prototype and encouraging prospects for future development.

Keywords: car air-conditioning, absorption refrigeration, renewable energy

Introduction

Since 1987 the Montreal Protocol controls the use and release of CFCs and has set a time-scale schedule for eliminating their production. This agreement is an historic step in the ongoing process of building consensus regarding environmental impacts of CFCs (Epstein and Manwell, 1992).

One of the HCFCs, R-22, and one of the HFCs, R-134a, are utilised as substitutes for CFCs, but the HCFCs and HFCs will face similar restriction for their high GWP 9 the parameter of Global Warming Potential). For comparison, some of the working fluids' ODP (the parameter of Ozone Depletion Potential) and GWP are listed in Table 1 (Epstein and Manwell, 1992).

To date, almost all car air-conditioning systems are charged with R-134a. However, alternatives with lower GWP than R-134a are desirable.

Some new systems are being developed in order to revitalise the use of ecologically safe refrigerants. For example, a system for car air-conditioning using CO_2 as the refrigerant has been developed by Lorentzen and Pettersen (1993). The testing of a laboratory prototype has shown that CO_2 is an acceptable refrigerant for car air-conditioners.

Due to the international attempt to find alternative energies, absorption refrigeration has become a prime system for many cooling applications. Where thermal energy is available the absorption refrigerator can very well substitute the vapour compression system.

Table 1: Environmental impact

Ozone Depleting and Global Warming Potentials			
Chemical	*ODP*	*GWP*	*Estimated atmospheric life (years)*
CFC-12 CCl_2F_2	0.93	3 700	122
HCFC-22 $CHClF_2$	0.05	510	18
HFC-134a CF_3CH_2F	0	400	18
Carbon dioxide CO_2	0	10	230
Ammonia NH_3	0	0	-
Water H_2O	0	0	-

It is a well-known fact that a large amount of heat energy associated with the exhaust gases from an engine is wasted. A rough energy balance of the available energy in the combustion of fuel in a motor car engine shows that one third is converted into shaft work, one third is lost at the radiator and one third is wasted as heat at the exhaust system (Greene and Lucas, 1969). Even for a relative small car-engine, such as for the Nissan1400, 15 kW of heat energy can be utilised from the exhaust gas (Wang, 1997). This heat is enough to power an absorption refrigeration system to produce a refrigeration capacity of 5 kW.

The standard working fluids for absorption refrigeration plants are water and ammonia, Lithium-Bromide and water, and Tetra-Ethylene Glycol Dimethyl-Ether (TEG-DME) and R-22. Of these combinations, water and ammonia is no threat to the environment and is preferable for this application for reasons listed below:

- Water has the highest latent heat of vaporisation at 0° C but its combination with LiBr can cause crystallisation due to unstable temperature conditions caused by fluctuations of exhaust gases flow rates.
- Freon-22 is a well known refrigerant, but in combination with its absorbent the plant becomes uneconomical.
- Ammonia is highly soluble in water and this ensures low solution circulation rates. Both constituents are obtainable at minimal cost.

The choice of Ammonia-water combination is not made without considering certain disadvantages: ammonia attacks copper and its alloys when it has been hydrated. Therefore, all components are made from mild steel or stainless steel.

The American National Standards Institute (ANSI) (King, 1977) classified refrigerants into three groups as to their safety in use. Ammonia due to its toxicity falls into group 2, which means that it cannot be used in air-conditioning systems in direct expansion in the evaporator coil. Equipment must be installed outside of the inhabitant space.

In order to circumvent the toxicity problem, water or glycol is used as a secondary fluid to transfer the heat from the passenger space to the evaporator. In this manner, the chance of ammonia contact with the passengers is minimised.

Design and testing of the prototype

Preliminary analysis (Wang, 1997) showed that an absorption refrigeration plant with a 2 kW cooling load at 0°C and with water as a secondary fluid, is more than sufficient to air-condition the passenger space of the NISSAN 1400 truck.

This cooling load is calculated from three difference heat sources:

1. Transmission of heat through the car body structure $Q_T = 0.19$ kW
2. Solar heat gain through the wind-screen and side windows $Q_R = 0.44$ kW
3. Internal heat gains $Q_I = 0.50$ kW

Total $Q_{TOT} = 1.13$ kW

This prototype unit shown in Figure 1 consists of the generator and analyser (1); reflux condenser (2); condenser (3); accumulator (4); evaporator (5); absorber (6); solution pump (8); expansion valve (9); solution expansion valve (10) and heat exchanger (11). The fan coil (12) serves for both heating and cooling of the passenger's space.

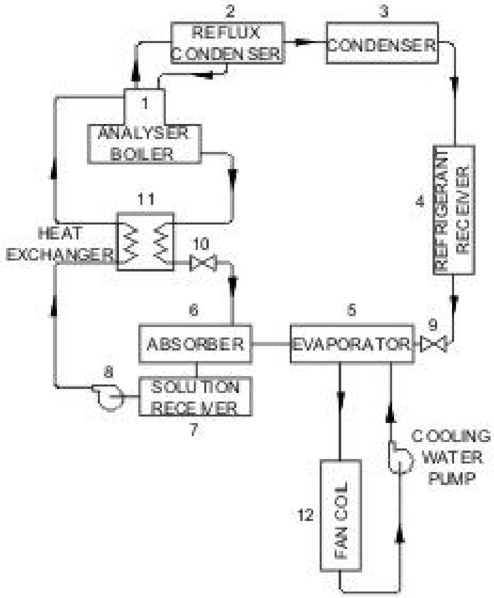

Figure 1: Schematic arrangement of the absorption system

Some of the above mentioned components are shown in the display of Figure 2 and in schematic arrangement in Figure 3.

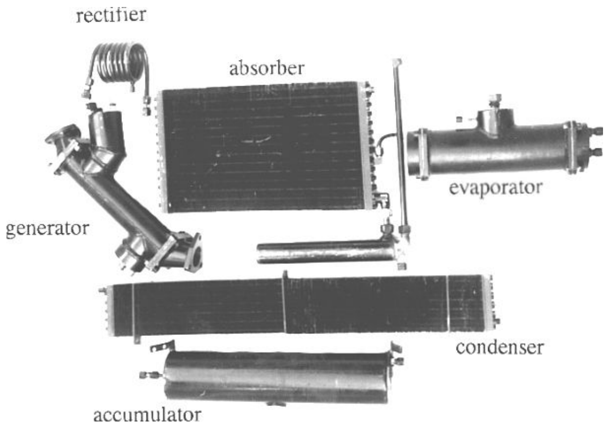

Figure 2: Components of the car air-conditioning plant

Results

Laboratory test

Prior to installing the components in the car, the absorption plant is assembled for a laboratory test. The exhaust gas from the engine is simulated by the combustion of propane through a gas burner. The air-cooled condenser and absorber are subjected to an air draft created by air-blowers maintaining the condensate and the strong solution in the absorber at an average temperature of 35^{0}C.

The plant is subjected to a variation of expansion valve settings[1] and its cooling capacity is established by measuring the temperature drop and rate of water circulation through the evaporator. The inlet to the evaporator is between -2°C and 0°C.

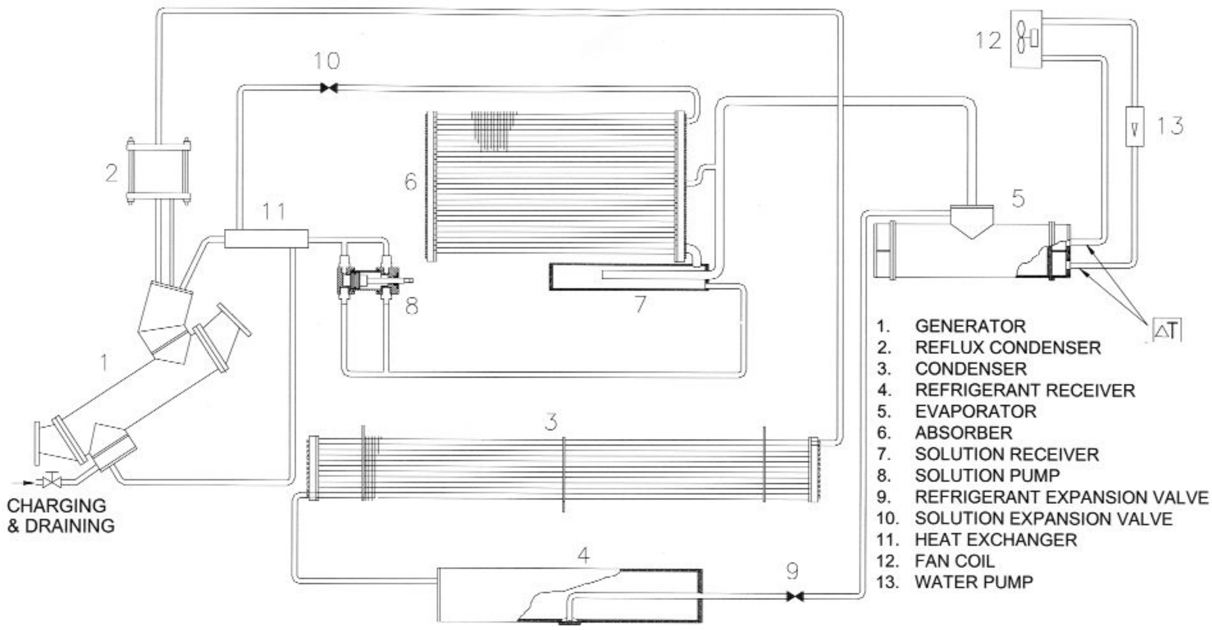

Figure 3: Schematic diagram of the assembly of the absorption air-conditioning system

Under these conditions, the load variation is shown in Figure 4, where a maximum of 2 kW cooling is obtained.

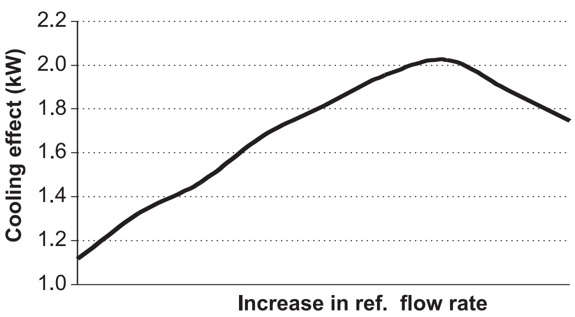

Figure 4: Load variation

Road-test

The road-test results are collected in two groups, (vehicle in motion):

1. Before opening the expansion valve
2. After opening the expansion valve.

These results are depicted in the graph of Figure 5, where a drop in the temperature in the passenger's space is clearly indicated.

Table 2 contains the averaged results collected from the absorption plant during the entire duration of the road tests. These averaged values serve to plot the enthalpy-concentration performance chart for the plant as shown in Figure 6. The construction of the absorption cycle on the enthalpy-concentration diagram is according to established theory of absorption refrigeration machines (Vicatos, 1995).

With reference to the data from Figure 5 and Table 2, the diagram of Figure 6 is constructed and analysed as follows:

The 3bar low pressure and the 5°C evaporator's temperature, establish points '5L' and '5V'.

At the generator and condenser, the data is not sufficient for a full evaluation. However, with the simplification that there is only a reflux condenser, instead of a distillation column to purify the generator's vapours, point '1' can be considered as the vapour in equilibrium with the generator's solution, point 'A'. This solution boils at 120°C (Table 2) and the line A-1 is the isotherm between the liquid and its vapour. This vapour becomes the condensate at 8bar pressure and this is depicted by point 'E'. The temperature curve through point 'E' is 28.4°C, which is in agreement with the average ambient cooling temperature, about 25°C (see Figure 5).

The vertical line 1-E indicates the concentration of the refrigerant, y= 0.737.

The line 1-E intersects the isotherm 5L-5V at point 'D' and the difference in enthalpy h_D-h_E is the Refrigeration Effect, RE=188.7kJ/kg.

The absorber, at point 'B', is at conditions of 3bar pressure and 28.4°C temperature. This temperature is the same as that of the condenser at point 'E', because both are cooled by the same medium.

Table 2

Pump rate of strong solution W^{ST} (g/s)	*Low pressure (bar)*	*High pressure (bar)*	*Generator temperature (°C)*	*Evaporator exit temperature (°C)*
21.5	3	8	120	5

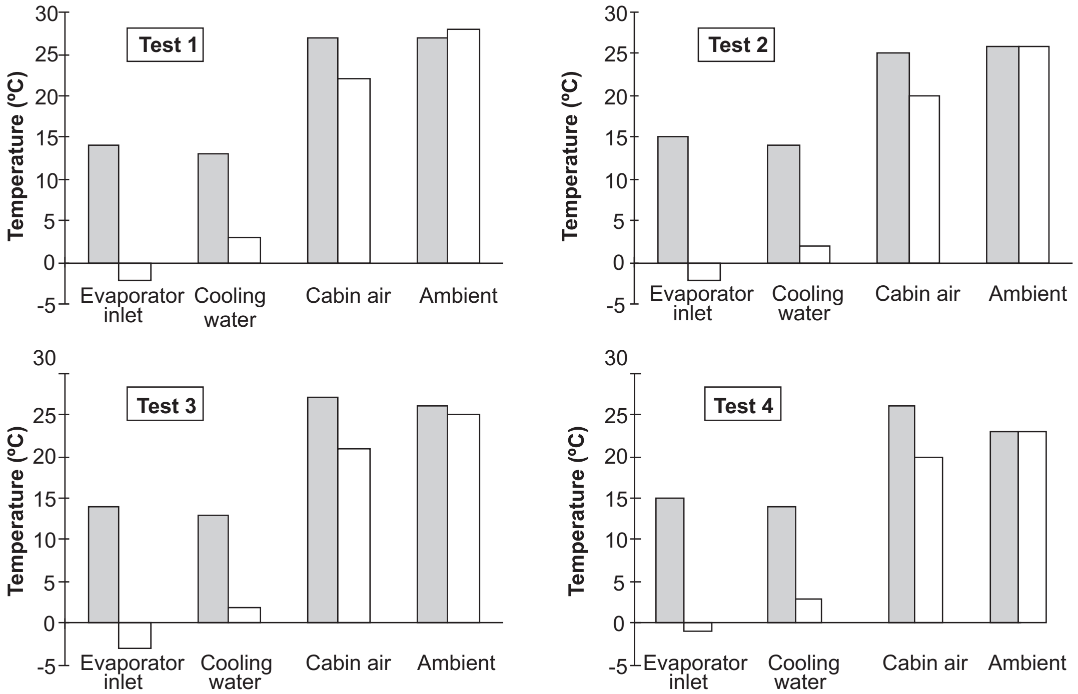

Figure 5: The shaded and the clear areas indicate the temperature conditions before and during the road-tests respectively

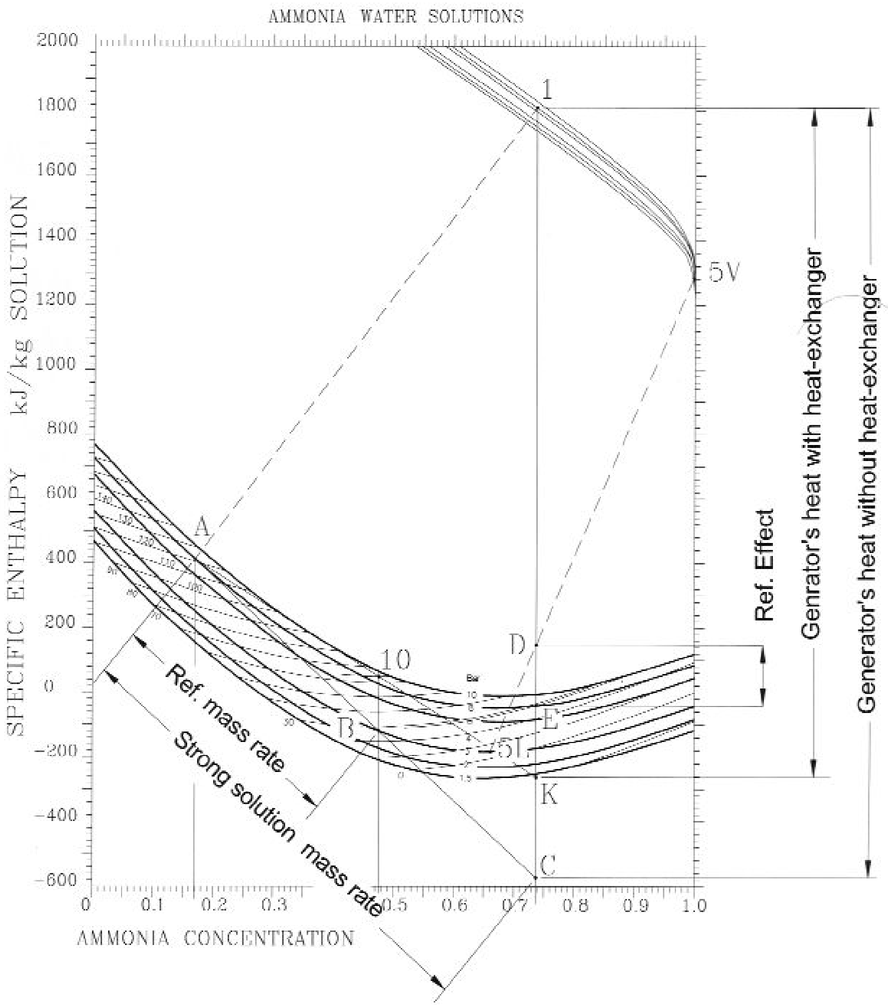

Figure 6: Averaged values for the enthalpy-concentration performance chart of the plant

The vertical line through point 'B' indicates the concentration of the strong solution, $X^{ST} = 0.475$.

The vertical line through point 'A' indicates the concentration of the weak solution, $X^{WE} = 0.170$.

From established theory (Vicatos, 1995), the ratio of the differences of concentrations is equal to the solutions mass rates.

$$\dot{f} = \frac{y - x^{WE}}{x^{ST} - x^{WE}}.\dot{m} = \frac{\overline{AC}}{\overline{AB}}.\dot{m}$$

Where

$\dot{f}$ = the strong solution mass rate = 21.5g/s see Table 2)

$\dot{m}$ is the refrigerant mass rate which is equal to

$$\dot{m} = \frac{\overline{AB}}{\overline{AC}}.\dot{f}$$

The capacity of the plant can be calculated using the information from Figure 6. The ratio of the lengths AB to AC represent the ratio of the mass rates of the refrigerant to the strong solution.

Thus $\frac{\overline{AB}}{\overline{AC}} = \frac{1}{1.854}$

And $Capacity = RE.\left(\dot{m}\right)$

Substituting the values:

$$Capacity = DE.\left(\dot{f}\right).\frac{\overline{AB}}{\overline{AC}}$$

$$= 188.7(0.0215)\frac{1}{1.854} = 2.188kW$$

The COP value is calculated as the ratio of refrigeration effect to the generator's heat.

$$COP = \frac{RE}{Gen.Heat}$$

The generator's heat, however, can be reduced by the use of a regenerator, or a heat exchanger.

During the road-test, there was no provision made to measure the temperature of the solution at the exit of the heat exchanger. Therefore, the following calculation of the COP is only an indication and not the real condition.

From Figure 6:

$$\text{COP} = \frac{h_D - h_E}{h_1 - h_C} = \frac{188.2}{2383.9} = 0.08$$

(Without the use of a solution heat exchanger)

A solution heat exchanger would bring the strong solution to a higher temperature than that of the absorber, (let's say to 54°C, point '10'). This would reflect to a heat saving in the generator, equivalent to the enthalpy difference h_K-h_C. The new COP will be in this case:

$$\text{COP} = \frac{h_D - h_E}{h_1 - h_K} = \frac{188.2}{2071.8} = 0.09$$

(With the use of a solution heat exchanger)

Conclusions

The theoretical analysis, is verified by both laboratory and road tests through the results obtained. This work results from a prototype which will have to be improved for further development. The claim that is made from this work is that it has shown the feasibility of such a system in a positive frame.

It can be concluded that:

1. In the exhaust gases of motor vehicles, there is enough heat energy that can be utilised to power an air-conditioning system. Therefore, if air-conditioning is achieved without using the engine's mechanical output, there will be a net reduction in fuel consumption and emissions.
2. Once a secondary fluid such as water or glycol is used, the aqua-ammonia combination appears to be a good candidate as a working fluid for an absorption car air-conditioning system. This minimises any potential hazard to the passengers.
3. The low COP value is an indication that improvements to the cycle are necessary. A high purity refrigerant would give a higher refrigeration effect, while the incorporation of a solution heat exchanger would reduce the input heat to the generator. The present system has both a reflux condenser and a heat exchanger. However, the reflux condenser is proved inadequate to provide high purity of the refrigerant and needs to be re-addressed. The evaluation of the COP, with and without the heat exchanger also proves that unless there is a high purity refrigerant, the effect of the heat exchanger to the generator's heat is small.

Note

1. The refrigerant mass rate at the expansion valve is supported by a corresponding strong-solution mass rate, by adjusting the 'pump-rate' of the solution-circulating pump. Although there is no quantitative result on the mass rate of the refrigerant, or on the solution circulation, there is only a qualitative result on the cooling effect of the laboratory-test plant.

References

Epstein G.J. and Manwell S.P. (1992) Environmental Trade-offs Between CFCs and Alternative Refrigerants, *ASHRAE Journal.*

Greene A.B. and Lucas G.G. (1969) *The Testing of*

Internal Combustion Engines, The English Universities Press.

King G.R. (1977) *Modern Refrigeration Practice*, McGraw-Hill, Inc.

Lorentzen G. and Pettersen J. (1993) A New, Efficient and Environmentally Benign System for Car air-conditioning, *International Journal Refrigeration,* 16, 1.

Vicatos G. (1995) *Heat and mass transfer characteristics: Design and optimisation of absorption refrigeration machines*, PhD thesis, University of Cape Town South Africa.

Wang S. (1997) *Motor vehicle Air-conditioning utilising the exhaust gas to power an absorption refrigeration cycle*, MSc thesis, University of Cape Town South Africa.

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