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Design and Fabrication of Auto Air Conditioner Generator Utilizing Exhaust Waste Energy from a Diesel Engine

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Abstract

In this investigation; analysis and design of an automobile air conditioner was made by utilizing the available diesel engine exhaust waste energy to provide the required heat for the generator. Because automotive air conditioning is one of the most equipment that heavily uses CFC compounds, and the leakage of CFCs from such air conditioners affect the environment, the absorption cycle was found to be an ideal option. Cooling load for the automobile has been estimated and found to be within acceptable ranges which are about 1.37 TR. The reported results show that the COP values directly proportional with increasing generator and evaporator temperatures. Measured COP values of the proposed model varied between 0.85 and 1.04.

The generator was designed and fabricated for optimal performance and could be rapidly transfer to the industrial applications, The system was found to be applicable and ready to produce the required conditioning effect without any additional load to the engine The proposed system decreases vehicle operating costs and environmental pollution caused by the heating system as well as causing a lower global warming. Simple experiments were carried out to examine the performance of the generator heat shown the benefit of exhaust gases as an alternative heating source.

Keywords: *Generator ; Absorber ; COP; Waste Energy; Cooling Load; Aqua- ammonia*

1. Introduction

Carbon dioxide coming out of every car's tailpipe is a greenhouse gas. The ultimate effects are unknown, but it is a strong possibility that, eventually, there will be dramatic climate changes that affect everyone on the planet – global warming to be exact. For this reason, there are growing efforts to reduce the green house gases. Automobile air conditioning currently is performed by vapor compression refrigeration systems, but the refrigerants in vapor compression refrigeration systems are mainly HCFCs and HFCs, which are not environmentally friendly, and the compressor uses a significant portion of the engine power. According to the statistic information, 65-70% of the combustion energy of the fuel consumed is taken away by the radiators and exhaust gases. If the waste heat can be recovered, it is enough to satisfy the input power need for air-conditioning. The waste heat from exhaust gases constitutes a large percentage of the total waste heat.

Much of an internal combustion engine's heat from combustion is discarded out of the exhaust or carried away via the engine cooling water. All this wasted energy could be useful. The common automobile, truck or bus air conditioner uses shaft work of the engine to turn a mechanical compressor. Operating the mechanical compressor increases the load on the engine and therefore increases fuel consumption, emissions and engine operating temperature. Nowadays all the scientists and car manufacturers in the world search to solve two main problems in vehicles. The first problem is fuel economy and reduces the losses and the second problem is environment pollution. The most available studies that conducted the auto air conditioners using waste energy as thermally driven are Robert. and Frosch, [1]. Investigated the automotive air conditioner utilizing the solar and motor waste heat. Binghadi and Agarwal [2]. proposed a use of lithium bromide Li.Br –Zn Br as a working fluid using a computer aided analysis of the thermal system. Frank et al [3]. Designed a heat generating apparatus and system for an automobile to use for an absorption air conditioning system including temperature control.

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Alhusein Inayatallah, [4]. designed a simple aqua ammonia absorption system for automobile air conditioning utilizing the exhaust waste heat from a spark ignition engine. Gui et al [5].The feasibility of application of a solid-absorption system using ammonia and chlorides as working pair to automobile air-conditioning system is investigated. Masadeh, Sh. [6]. Carried out the analysis and investigation of an automobile aqua- ammonia air conditioner using the available for the engine exhaust gases and the desired cooling load from Alhusein and Inayatallah (1994).A complete system was built and the performance of the system was studied based on the different factors including the temperatures and the exhaust gases flow rate and the engine power. He suggest that the system can be manufactured easily and transferred to the industry and more commercial size recommended to be studied. Al-Aqeeli1and Gandhidasan [7]. The feasibility and design of an air conditioning system for automobiles using the Open Cycle Absorption System, with LiBr-H2O as the working fluid. Shah Alam [8] proposed a model for utilizing the exhaust waste heat to run automobile air-conditioner In his work three fluid (Ammonia–Hydrogen and Water) vapor absorption systems is used for air conditioning of four strokes, four cylinders passenger car.

2. Objective

The main purpose of this investigation is focus lights on the feasibility to utilize the waste energy from the exhaust for a diesel engine and design an suitable size of automobile air conditioning system based on absorption cycle and to explore the advantages of this system over conventional air-conditioning system using vapor compression cycle. In this investigation we will focus on the generator to explore the feasibility of using waste energy to design the generator since this component is the most important component of absorption system and it is directly influence the performance of the whole system.

3. Theoretical Background

Based on estimated cooling load calculations, the cooling capacity or heat must be removed from the evaporator space is 5 kW or (Q_e = 4.8 kW), [9-12]. In order to measure the exhaust waste energy to know the amount of heat that can be utilize by the generator (Q_g). Shell and tube heat exchanger has been used. Water passed through the tube and hot gases from the exhaust flow in the shell and touch the pipe wall as shown in figure1.

The waste heat from the exhaust which can be transferred to the water and the heat gain by water can be estimated from the following [13,14]

$$Q_g = \dot{m} cp \Delta T \qquad (1)$$

Where Q_g is the generator heat (kW) , $\dot{m}$ is Water mass flow rate and C_p: specific heat at constant pressure. To find the coefficient performance of the absorption cycle [15]:

$$COP = \frac{Q_e}{Q_g} \qquad (2)$$

Table 1. Summarized the measured heat gained by water which equal to the available heat for the generator

RPM	T_{in} (°C)	T_{out} (° C)	Q_g (kW)	COP
900	16	٧٠.5	4.548	1.099384
1200	16	73.8	4.823	1.036616
1300	16	77	5.09	0.982236
1400	16	80.2	5.357	0.982236
1500	16	83.9	5.666	0.882421
1600	16	85	5.758	0.868354
1800	16	87.1	5.933	0.842706
2000	16	89	6.092	0.820773

Table 1 represents measured values for waste heat at different temperatures and engine

4. Results and Discussion

Based on the data reported figure 2 shows the variation of the coefficient of performance with the energy available in the exhaust or heat available in the generator. This energy obtained without additional energy input and from equation the COP inversely proportional with heat generated in the exhaust. The main components that controlled COP are evaporator and generator heat, in other words exhaust and evaporator conditions. So, changing the values of these two quantities will change the COP.

From the results obtained, it is clear that the system is feasible to be designed with good reduction in the cost of operation and fuel consumption.

The same trend for the heat generated can be seen for the exhaust temperature shown in figure 3, the heat in the generator proportional directly with the exhaust gas temperature that means low COP has been reported at higher exhaust gas temperature. the temperature in the refrigerator was increased. At such condition there was excessive energy being transferred from the high-temperature exhaust gas to the refrigerant, not allowing its condensation in the condenser due to the elevated sensible heat to be removed

Fig.2. Shows the variation of the exhaust gas temperature with the diesel engine speed at different loads. This temperature increases with the engine speed and the vehicle load. This means that the higher the engine speed and vehicle load, the higher the amount of heat provided to the passenger compartment. For the vehicle loads of 20% and 80%, the maximum exhaust gas temperatures were 230 and 280 °C, respectively. It is seen that this temperature had sufficiently high values when the engine speed was between 1500 and

2500 rpm. Furthermore, exhaust gas temperatures at different vehicle loads diverge with the increasing engine speed.
Consequently, the heat generated or energy available in the exhaust gases variation with the engine speed explored in figure 3. Directly proportional relation between Qg and engine speed as a result of exhaust gas temperature increasing. Therefore, absorption refrigeration system may be able to take advantage of the exhaust gas power availability and provide the cooling capacity required for automotive air conditioning. The heat available in the exhaust gas it is able to evaporate the aqua-ammonia solution to complete this cycle. The waste heat energy available in exhaust gas is directly proportional to the engine speed and exhaust gas flow rates in addition the exhaust gas flow temperature which was indicted by figure 4.
Higher COP for the diesel engine using exhaust waste energy can be achieved at the
From the previous discussion regarding the COP value, it can be seen that this COP depends mainly on the evaporation and regeneration energies. So, changing the values of these two will change the COP produced. This indicates that higher regeneration temperatures are needed in order to allow the system to work efficiently.
From the results obtained in this work, it is clear that the system is feasible to be designed with good reduction in the cost of operation and fuel and power consumption in addition to low engine emissions.

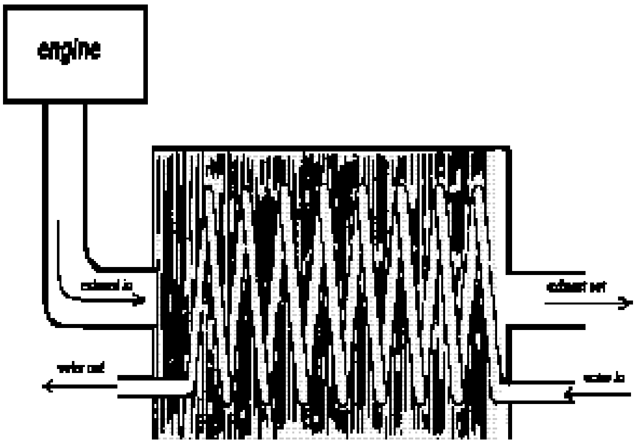

Figure1. Exhaust gas flow through the heat exchanger

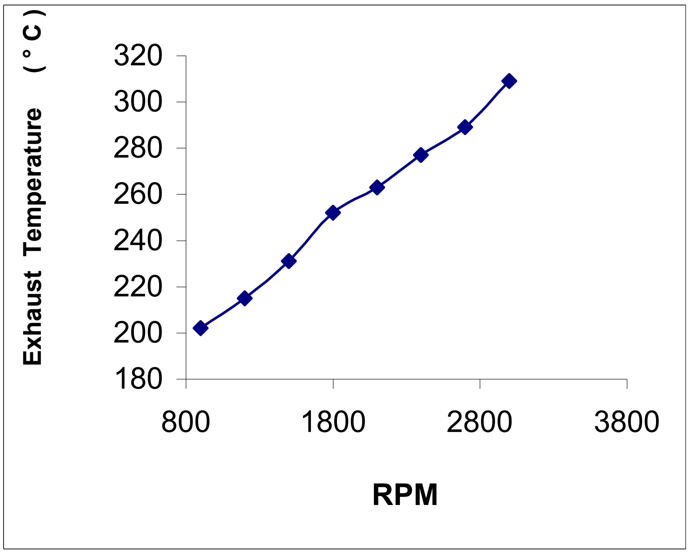

Figure 2. The effect of diesel engine speed on the exhaust gas temperature

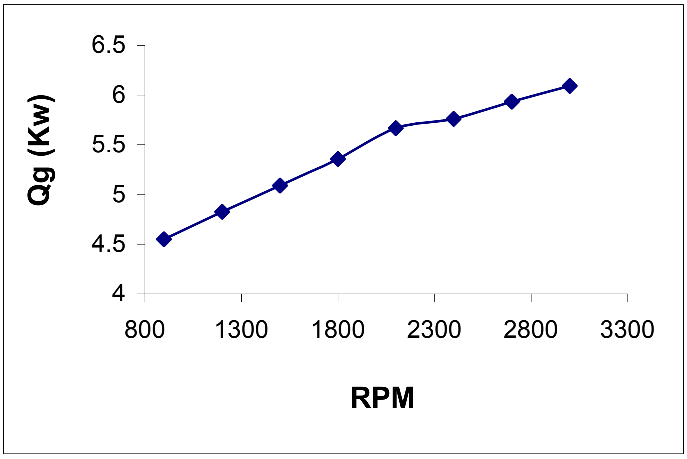

Figure 3. The effect of diesel engine speed on the exhaust heat generation (Qg)

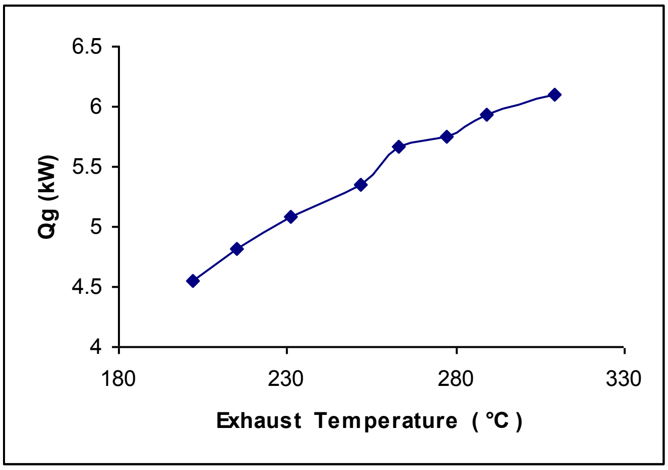

Figure 4. The effect exhaust gas temperature on heat generation (Qg).

5. Design and Analysis of The Proposed Model

The generator is used to create the same task of the compressor in the conventional compression refrigeration cycle. It is located where the heat is available from the exhaust gases, and the important limiting factor the space occupied by generator. The generator used to evaporate the mixture of ammonia that react with water and leaves pure ammonia or mixture with high ammonia concentration.
The generator is design to have a capacity of 4.6 kW with temperature around 90 °C and pressure of 19 bar. The maximum space available in the automobile that this component can be installed is 50 cm long, 25 cm wide and 15 cm high.
Assuming that the overall heat transfer coefficient through the generator, mass flow rate and specific heat are constant, and that there is no heat loss to the atmosphere, the external heat transfer area of the generator can be calculated.
The external heat transfer area required is calculated from [16] :

$$A = \frac{Q}{U \Delta LMTD} \qquad (3)$$

To find ΔTLMTD: log mean temp difference

$$\Delta LMTD = \frac{\left[(T_{hi} - T_{co}) - ((T_{ho} - T_{ci}))\right]}{\ln\left[(T_{hi} - T_{co}) \Big/ ((T_{ho} - T_{ci}))\right]}$$

(4)

T_{hi}= = 543 k and T_{ho}= outlet exhaust temp= 473 k
$\Delta LMTD$ = 182.6 °C

The overall heat transfer coefficient, neglecting the fouling resistance of the tube, is given by the equation [18]:

$$\frac{1}{U} = \frac{1}{h_i} * \frac{d_O}{d_i} + \frac{d_o}{2k_s} * \ln\left(\frac{d_o}{d_i}\right) + \frac{1}{h_o} \qquad (5)$$

To find (h_i) inside heat transfer coefficient from Nusselt number

$$Nu = \left(\frac{h_i D_i}{k_L}\right) \qquad (6)$$

As the flow inside the generator is two-phase flow. The following equation can be used as long as liquid wets the wall [17,20]:

$$Nu = 0.06\left(\frac{\rho_1}{P_v}\right)^{0.28}\left(\frac{D_i G_x}{\mu_1}\right)^{0.87}(\Pr)^{0.4} \qquad (7)$$

$$Nu = 1246.549 = \frac{h_i D_i}{k_l} \qquad (8)$$

$$h_i = 25.4763 \left(kW / m^2 k\right)$$

For flow outside the tubes, the equation for normal to the bank of tubes can be used , and the outside film coefficient can be calculated [19].

$$\left(\frac{h_o * D_o}{k_g}\right) = C(\mathrm{Re})^n (\Pr)^{0.36} \left(\frac{\Pr_\infty}{\Pr_s}\right)^{0.25} \qquad (9)$$

Where:
C and n: constant depending on Re.
K_g: thermal conductivity of exhaust gases.
The staggered design of the cross flow heat exchanger shown in figure 5 was used for the generator, so we used the following procedure[16,19]:

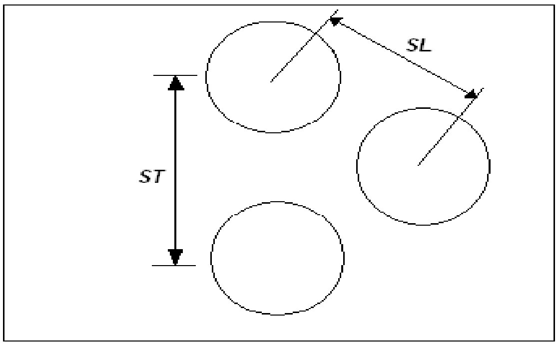

Figure 5. Schematic diagram for staggered design of tubes

S_T =2D
S_L = 1.5 D
$S_D = [(0.5\ S_T)^2 + SL^2]^{0.5}$
If $2(S_D – D) > (S_T – D)$, then

$$V_g = \frac{(V_\infty S_T)}{(S_T - D)} \qquad (10)$$

where:
V_g =velocity of the exhaust gases through the generator.
V_∞= velocity of the exhaust gases entering the generator.
D =outer diameter of the tube.
Then the following correlation will be used in the design [20, 21] :

$$N_{UD} = C(\mathrm{Re})^n\ (\mathrm{pr})^{0.36}\ \left(\frac{\mathrm{pr}_\infty}{\mathrm{pr}_s}\right)^{0.25} \qquad (11)$$

A steel tube can be used for the generator material because the steel does not reacted with the ammonia solution A schedule 40 steel tube of 1.6 cm inner diameter and 2.134 cm outer diameter was used for the generator, then according to this size:
S_T = 4.268 cm, S_L=3.201 cm, S_D =3.847 cm

$2(S_D - D) = 3.42625$

$(S_T - D) = 2.134$

Since $2(S_D - D) = 3.42625 > (S_T - D) = 2.134$, then

V_∞ was measured be 7 m/s. and V_g =14 m/s.

$$C = 0.35\left(\frac{S_T}{S_L}\right)^{0.2} = 0.37073 \qquad (12)$$

Pr_∞= Prandtls at fluid temperature = 0.703

Pr_s= Prandtls at surface temperature = 0.705

n = 0.6, C = 0.37073

T_f = 510 k So we take the properties at T = 510k

Pr = 0.69

From Silva and Costa [22] , the properties for the exhaust gas can be found as :

$$\rho_g = \frac{353}{T_g} = \frac{353}{508} = 0.694\left(kg/m^3\right)$$

$$\mu_g = 1.348*10^{-5} + 2.68*10^{-8} \times \left(T_g\right)$$

$$\mu_g = 2.74*10^{-5}\left(Ns/m^2\right)$$

$$k_g = 8.459\times10^{-3} + 5.7\times10^{-5} \times \left(T_g\right)$$

$$k_g = 0.037915\ (W/mk)$$

V_g= 14 m/sec

Where Tg is the exhaust gas temperature

$$Re = 7567.13$$

Nu= 69.88

$$h_o = 124\left(W/m^3k\right)$$

Substituting values in U

k_{steel}= 42 w/m^2k [16]

U= 122.139 W/m^2k

Q_g= 4.6 kW

$$Q_g = UA_s\Delta LMTD$$

So the Area required A_s= 0.196 m^2 and L=2.932 m

In order to distribute the available length in the specified space we have a space for tubes of long 8 cm and outside diameter 2.134 cm to leave space for the solution to be collected through the other space. Therefore the number of tubes required is 37.

We need 37 tubes of outside diameter 2.134 cm and inside diameter of 1.6 cm

The recommended shape of the proposed generator as shown in figures 6 and 7.

Which is seemed to be in a good agreement for the model proposed by [6] for four stroke spark ignition engine.
The properties of ammonia and water mixture and air are available in literature and from [23-25]

6. Conclusion

Diesel engines can be considered as a potential energy sources for absorption refrigeration systems, because of the energy wasted through the exhaust gas.
the absorption refrigeration system may be able to take advantage of the exhaust gas power availability and provide the cooling capacity required for automotive air conditioning.
The waste heat energy available in exhaust gas is directly proportional to the engine speed and exhaust gas flow rates. The coefficient of performance found to be between 0.85 and 1.045 which has a good agreements with data reported in literature. Reducing the fresh air intake and sealing the automobile body can result in a saving in cooling requirements such as door sealing and tinting the glass.
Feasibility study should made to decide the unit's chances to be produced on commercial scales. Also, applying this project practically in Jordan, because it has many advantages from the pollution and economic point of view.

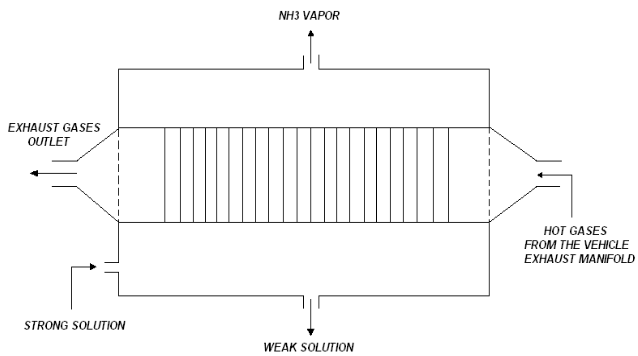

Figure 6. Schematic diagram for fluid flow in the proposed generator

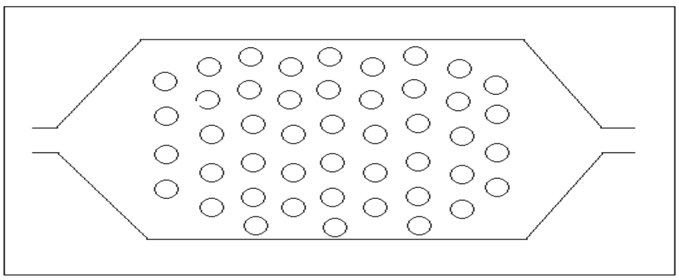

Figure 7. Schematic diagram for the tubes arrangement in the proposed generator From the other side of the generator

Nomenclature

A	External heat transfer area ,m^2
Ao	Outside bare tube area (m^2)
A/C	Air Conditioning
COP	Coefficient of Performance
Cp	constant pressure specific heat ,kJ/kg k
D	Diameter , m
G	Mass flux ,kg/m^2 s
g	gravitational acceleration ,m/s^2
h	Convective heat transfer coefficient of tube, W/m^2k
k	thermal conductivity, W/ m k
LMTD	Log mean temperature difference ,°C
m	mass flow rate , kg/hr
P	Pressure , Pa
Q	Heat gained or lost in a certain device ,kW
Qg	Heat gained in the generator (kW)
Tci	inlet water temperature ,°C
Tco	outlet water temperature, °C
Tf	Mean film temperature ,°C
Thi	Inlet exhaust temperature ,°C
Tho	outlet exhaust temp, °C
U	Overall heat transfer coefficient , W/m^2k

Greek Symbols

μ	Dynamic viscosity, Pa.s
ρ	Mass density, kg/m^3
ß	volumetric thermal expansion coefficient ,k^{-1}
ΔT	Temperature difference ,°C
ν	kinematics viscosity,m^2/s
ℓ	Liter , m^3

Subscripts

a	air
g	gas
i	Inner side
l	liquid
o	Outer side

Non-dimensional Numbers

Gr	Grashoff number
Nu:	Nusselt number
Pr	Prandtls number
Ra	Rayleigh number
Re	Reynolds number

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