Energy and Exergy Analysis of the Diesil-Absorption System

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Abstract- the absorption refrigeration system is attracting increasing research interest, since the system can powered by waste thermal energy, and does not use ozone depletion refrigerants. The purpose of this paper is to present the energy and exergy analysis of a turbocharged Diesel engine combined with an absorption refrigerator unit. The SPICE program is used in the analysis of the energy values at different locations of the Diesel engine and computer program was written for the absorption refrigeration cycle. The second law is applied to perform an exergy analysis in the Diesel-absorption cycle. Results show that the majority of the engine irreversibility occurs within the combustion chamber and the absorber has the highest exergy loss in absorption cycle.

Keywords energy, exergy: Diesel engine, SPICE, absorption refrigeration

NOMENCATURE

COP	Coefficient of Performance
h	Enthalpy (kJ/kg)
$\bar{h}_c$	Enthalpy of combustion (kJ/kmol)
$\bar{h}_f^o$	Enthalpy of formation per mole at Standard state (kJ/kmol)

m	Mass flow rate (kg/s)
mf	mass flow rate
M	Molecular weight
N	Number of moles (kmol)
P	Pressure (Pa), pump
Q	Heat flow rate (kW)
R	Gas constant $\bar{R}/M$
$\bar{R}$	Universal gas constant
s	Entropy (kJ/kg k)
T	Temperature (K)
x	Mass faction of lithium bromide (%)
y	Mole fraction

Subscripts

a	Absorber
c	Cold side, Condenser
C	Compressor
CC	Combustion chamber
ch	Chemical contribution to availability
cw	Chilled water
e	Outlet, evaporator
f	Fuel
g	Generator
h	hot side
i	Inlet, constituent of the mixture
int	Inter-cooler
o	Ambient
p	Products
pr	Pre-cooler
r	Reactants
ph	Physical

T	Turbine
th	Thermomechanical contribution to availability
she	Solution heat exchanger
v1	Refrigeration expansion valve
v2	Solution expansion valve
1-18	Stations in the Absorption Cycle
1,2,3,4,and 5	Different states of a system
$\bar{}$	Bar over symbol denotes property on a molar basis

Greek Symbols

$\Delta\psi$	Availability difference
ψ	Availability (kJ/kg)
Σ	Summation

INTRODUCTION

In recent years, there has been increased interest in integrating systems for combined heat and power, [1], which is defined as a thermal system that produces electrical and heat energy simultaneously from a single source of fuel [2]. The absorption refrigeration system using aqueous lithium bromide as the absorbent and water as a refrigerant has received much attention in the specialised literature. Moreover, most of these studies have dealt with the use of solar energy as source heat. The utilisation of exhaust waste heat is now well known and the forms the basis of many combined Heat and Power installations. Recently, the research and development of absorption refrigerators driven by different heat sources have become very important. Among these refrigerators, this paper investigates theoretically, single-effect absorption refrigeration driven by waste heat from a Diesel engine.

The development of computers has transferred the place where initial tests are carried out from the test bed to the computer screen. This has enabled the designer to test a greater number of alternative options than it could ever be possible using engine prototypes. This saves effort and money. In addition, the development of fast computers has led to the birth of engine cycle simulation programs, which provide a useful tool in engine performance prediction. The availability of low cost high-speed digital computers aided this process significantly. The program which will be used for this engine simulation is SPICE (**S**imulation **P**rogram for **I**nternal **C**ombustion **E**ngines) developed at the University of Bath [3].

One method that can be used to quantify the irreversible nature of a real process and to perform analysis based on the combined first and second laws of thermodynamics is the exergy analysis, [4]. Exergy is defined by [5] as a generic term for a group of concepts that define the maximum potential of a system, a stream of matter or a heat interaction, the state of the conceptual environment being used as the datum state. The exergy method, known as the second law analysis, calculates the exergy loss caused by irreversibility, which is an important thermodynamic property which measures the useful work that can be produced by a substance, or the amount of work needed to complete a process. Unlike energy, exergy is not conserved: analysis of exergy losses provides information as to where the real inefficiencies in a system lie. In this section, the exergy losses inherent in a process are calculated. The concept of exergy is extensively discussed in the books of [5] and [6]. Energy and exergy analyses were carried out for each component in the system, and the results tabulated.

ENERGY ANALYSIS OF THE ENGINE

The salient parameters of the Caterpillar Diesel engine examined are summarised in Appendix (1). It is estimated that the air inlet temperature to the compressor is $25°C$. The engine is represented as a system comprising thermodynamic control volumes and flow junctions obeying the principles of energy and mass conservation. A typical idealised block system representation is shown diagrammatically in Fig. (1), which shows how a typical problem can be set up before preparing the data for the computer. All volumes, junctions and shafts are given numbers, which are used in the structural data to describe how they are linked to form the system. The six control volumes numbered (1) to (7) represent the cylinder, intake and exhaust manifolds respectively. The control volumes are connected flow junctions, which are numbered (1) to (15). Junction (1) connect the intake manifold to the compressor and allows air to enter the system. Junction (2) connects the inter cooler to the inlet manifold. Junctions (3) to (8) are a cam operated poppet intake valve which connects the intake manifold to the cylinder. Junctions (9) to (14) a poppet exhaust valve, which connect the cylinder to the exhaust manifold. Junction (15) connect exhaust manifold to the turbine to allow the products of combustion to leave the system.

Table (1) Energy balance for the engine

Energy analysis	*Contribution (%)*
Useful Work	39.13
Exhaust	37.51
Coolant	19.05
Friction Work	4.07
Inter-cooler	0.21
System	100.00

EXERGY ANALYSIS OF THE ENGINE

A schematic diagram for the Diesel-Absorption system, which shows the exergy flow and the state points, is given in Fig. (1).The pre-cooler and inter-cooler are both supplied with chilled water from the absorption unit in the configuration shown. Referring to Fig. (1), the exergy balance equations and associated irreversibilities for various processes are written on the assumption of adiabatic condition. In this analysis it is assumed that the fuel is Dodecane. The exergy value of a steady stream of fluid entering or leaving part of a process is the maximum amount of energy or work that can be obtained from the stream in bringing it to equilibrium with the environment. It is given by [5]:

$$\psi = \left(h - h_o\right) - T_o\left(s - s_o\right) \tag{1}$$

Pre-cooler heat exchanger:

Heat exchange between the ambient air and the chilled water takes place in the pre-cooler. In this case, the chilled water is supplied from the absorption refrigeration unit in order to reduce the charge air temperature. The amount of heat recovered from the pre-cooler is:

$$Q_{cw} = m_{cw}\left(h_{cwpe} - h_{cwpi}\right) \tag{2}$$

The corresponding exergy recovered from the pre-cooler is:

$$\psi_{pre} = \psi_1 - \psi_o \tag{3}$$

Compressor:

The exergy of the compressed air is given [7]:

$$\Delta\psi_c = \dot{m}\left[\left(h_2 - h_1\right) - T_o\left(s_2 - s_1\right)\right] \tag{4}$$

where:

$$h_2 - h_1 = \int_{T_1}^{T_2} c_p \, dT \qquad (5)$$

and

$$s_2 - s_1 = c_p \ln\left(\frac{T_2}{T_1}\right) - R \ln\left(\frac{P_2}{P_1}\right) \qquad (6)$$

Where the value of c_p was determined from the following equation, [8]:

$$c_p = A + BT + CT^2 + DT^3$$

The internal power requirement of the air compressor is:

$$Q_{2\text{-}1} = \dot{m}c\left(h_2 - h_1\right) \qquad (7)$$

The availability destruction in the compressor has two components, viz. Mechanically and process irreversibilities: the former is as computed by [5]:

$$\Delta\psi_{mc} = \left(\frac{1}{\eta_{mc}} - 1\right) Q_{2\text{-}1} \qquad (8)$$

The gross power input to the air compressor is found by:

$$W_c = Q_{2\text{-}1} + \Delta\psi_{mc} \qquad (9)$$

The process irreversibility (exergy loss) in the compressor is:

$$\Delta\psi_{TC} = W_c - \left(\psi_{io} + \Delta\psi_{mc}\right) \qquad (10)$$

Inter-cooler:

Heat exchange between the compressed air and the chilled water takes place in the inter-cooler. In this case, the chilled water is supplied to the inter-cooler. This is from the absorption refrigeration unit. The amount of heat recovered from the inter-cooler is:

$$Q_{cw} = \dot{m}_{cw}\left(h_{cwie} - h_{cwie}\right) \quad (11)$$

The corresponding exergy recovered from the inter-cooler is:

$$\psi_{cw} = \psi_{cwie} - \psi_{cwii} \quad (12)$$

The exergy loss in the inter-cooler is determined by:

$$\Delta\psi_{ic} = \psi_{cwii} + \psi_2 - \psi_3 - \psi_{cwie} \quad (13)$$

Combustion Process:

In analysing the combustion process the exergy of the air and the fuel must be considered. The exergy of the fuel supplied to the combustion chamber is calculated by using equations for the molar composition of the fuel gas mixture. The result of the calculation is given in Table (3). Assuming the combustion is complete, the physical properties of the exhaust gases are determined from the product of a complete combustion of the fuel. The combustion equation on a mass basic is as follow [9]:

$$C_xH_y + a\,O_2 + 3.76a\,N_2 \rightarrow x\,CO_2 + \frac{y}{2}H_2O + 3.76\,a\,N_2 \quad (14)$$

where,

$$a = x + y/2$$

In this work Diesel fuel is considered to be Dodecane $C_{12}H_{26}$. The molecular weight of Dodecane is 170.337 kg/mole. Therefore for 1 mole of $C_{12}H_{26}$, the balance equation for the combustion process with the theoretical amount of air is as shown below:

$$C_{12}H_{26} + 18.5(O_2 + 3.76N_2) \rightarrow 12\,CO_2 + 13\,H_2O + 3.76\times18.5N_2 \quad (15)$$

Therefore, the total mass of the mixture is the sum of the masses of the individual components and the mole number of the mixture is the sum of the mole numbers of the individual components:

$$m_m = \sum_{i=1}^{k} m_i , \qquad N_m = \sum_{i=1}^{k} N_i$$

The mass fraction mf_i is the ratio of the mass of a component to the mass of the mixture, while, the mole fraction is the ratio of the mole number of a component to the mole number of the mixture y, in:

$$mf_i = \frac{m_i}{m_m} , \qquad y_i = \frac{N_i}{N_m}$$

The mass of the total products per kg of fuel is:

$$m_f = \sum_i y_i \, mf_i$$

The combustion air enters at T_0 and P_0 with the some composition as that listed for the gas phase in the environment Table (3), in this case no availability enters with it. Accordingly, the only availability entering the combustor is just that associated with the fuel. The hydrocarbon fuel (C_xH_y) is in the state (T_0, x_iP_0). The chemical availability of the hydrocarbon fuel at this condition has a different value [10]:

$$\psi_f^{ch}(T_0, x_iP_i) = \psi_f^{ch}(T_0, P_0) + RT_0 \ln x_i \qquad (16)$$

where x_i is the sum of the products of molar fraction, R is the universal gas constant.

The availability of the combustion chamber is the fuel availability plus the availability leaving the inter-cooler:

$$\psi_{cc} = \left(1 - \frac{T_o}{T_{av}}\right) Q_{cc} - W_{cc} + \sum_r \dot{m}_r \psi_r - \sum_p \dot{m}_p \psi_p \qquad (17)$$

Where, the first and second terms of the right-hand side are the exergy associated with the heat transfer between the system and the thermal reservoir at T_{av} and the network available respectively.

The heat transfer for steady-flow combustion process is determined from a steady flow energy balance applied to the combustion chamber per unit mole of the fuel:

$$Q_{out} + \Sigma \dot{N}_p \left(\bar{h}_f^o + \bar{h} - \bar{h}^o\right)_p = \Sigma \dot{N}_r \left(\bar{h}_f^o + \bar{h} - \bar{h}^o\right)_r \qquad (18)$$

For simplicity, when the combustion process is adiabatic and produces no mechanical work, the energy and availability equation will become respectively:

$$\sum_r \dot{m}_r h_r - \sum_p \dot{m}_p h_p = 0 \qquad (19)$$

The first part of the left side of the equation is known. The only unknown is the temperature T of the product. It can be determined in an iteration procedure as follows:

The value of T_p is guessed. This enables the h (T_p) value for each of the products to be determined from the tables provided in the literature [10]. The procedure continues until the value determined approximates the right side to the desired accuracy. Then, the irreversibility of the combustion chamber is:

$$\Delta\psi_{cc} = \sum_r \dot{m}_r \psi_r - \dot{m}_p \psi_p \qquad (20)$$

The total availability of the products where the temperature and pressures are different from the ambient is the sum of the thermomechanical and chemical availabilities, i.e.:

$$\psi_{fi} = \psi_{fi}^{th} + \psi_{fi}^{ch} \tag{21}$$

Where:

$$\psi_{fi}^{th} = \dot{m}\left[\left(h - h_o\right) - \left(s - s_o\right)\right] \tag{22}$$

$$\psi_{fi}^{ch} = \dot{m}\left[T_o \bar{R} \ln \frac{x_i}{x_i^o}\right] \tag{23}$$

$$\psi_4 = \left(h_4 - h_o\right) - T_o\left(s_4 - s_o\right) \tag{24}$$

The irreversibility in the combustion chamber is calculated as:

$$\Delta\psi_{cc} = \psi_{cc} - \psi_4 + \psi_3 \tag{25}$$

Turbine

In the turbine, the working fluid is expected to expand and convert its thermal energy into mechanical work.
where:

$$\psi_4 - \psi_5 = \left(h_4 - h_5\right) - T_o\left(s_4 - s_5\right) \tag{27}$$

The availability destruction for the turbine is determined by:

$$\Delta\psi_t = \psi_q + \psi_w + \psi_f \tag{28}$$

Since the first term of the right hand is zero (adiabatic condition),

$$\psi_w = -m_t\left(h_4 - h_5\right) \tag{29}$$

and

$$\psi_f = m_t(\psi_4 - \psi_5) \tag{30}$$

Table (2) exergy balance for engine:

Characteristic	*Contribution (%)*
Pre-cooler	0.0415
Compressor	21.44
Inter-cooler	1.389
Combustion Chamber	54.976
Turbine	23.15
System	100.00

THERMODYNAMIC ANALYSIS OF AN ABSORPTION REFRIGERAION

Energy and mass balances were written around each of the components and combined with the state equations for the thermodynamic properties of the lithium-bromide and water to yield a set of equations describing the system. In order to determine the heat and mass transfer of the components, each component was taken as a single unit, the balance equations of mass, energy, heat transfer as shown in Appendix B. The state equations were evaluated by the thermodynamic properties of Li-Br and water expressed [12]. Energy balances were imposed to obtain the solution listed in Table (3). The exergy analysis has been used to show the most effective part where the irreversibility can occur. A schematic diagram of the system energy balance is shown in Fig. (3) and the equivalent availability flow balance is shown in Fig. (4). The equations used in this analysis to produce Table (4) are presented in Appendices B and C.

Table (3) Energy difference for the refrigeration cycle

Component	Energy (kW)		
	Input	Output	Loss
Generator	+44.757	+44.755	0.002
Condenser	-32.292	-32.293	0.000
Evaporator	+30.000	+29.970	0.030
Absorber	-42.440	-42.430	0.010
Refrigeration expansion valve	-	-	-
Solution expansion valve	-	-	-
solution pump	1.44	1.44	-
Solution heat exchanger	+10.80	-10.78	0.02

Table (4) Exergy difference for the refrigeration cycle

Component	Exergy (kJ/kg)			Exergy difference $\Delta\psi$ (kJ/kg)
	Input	Output	Contribution %	
Absorber	244.11	228.6100	59.060	15.498
Generator	348.34	341.2500	27.020	7.0900
Evaporator	23.340	21.35000	7.5840	1.9900
Solution heat exchanger	453.755	452.9800	2.9350	0.7700
Condenser	24.668	23.94000	2.7740	0.7280
Refrigeration expansion valve	23.341	23.34100	-	-
Solution expansion valve	223.555	223.5500	-	-
solution pump	227.866	227.6950	0.1640	0.0430
System	1568.9	1542.661	100.00	26.239

RESULTS AND DISCUSSIONS

The complete energy and mass balance is equal to almost unity. For the system, energy and mass balance are 1.0004 and 1.005 respectively, which is acceptable. It can be seen from Table (1) that 39.13% of the input energy can be utilised to produce useful work, whereas 37.51% of the energy input is wasted through the exhaust. It is observed from this analysis, that the exhaust carries a significant amount of available energy, and the possibility of recovering at least a proportion of it merits serious attention.

The computer program for the exergy analysis has been used to show the most effective part where the irreversibility can occur. The exergy losses are spread over the components as shown in Table (2). The study reveals that the majority of system irreversibility occurs within the combustor, wherein lies approximately 54.976% of the destroyed exergy for the system. At outlet temperature, the chemical availability of the combustion products is small relative to the thermomechanical availability as the temperature decreases: the chemical availability will be come more dominant. The exergy destroyed in the remainder of the system such as compressor and turbine is about 21.44% and 23.15 %. The flow availability in the compressor and turbine decrease from the inlet to exit because power is delivered and availability is destroyed due to internal irreversibilities.

Energy balances were imposed to obtain the solution listed in Table (3). Table (4) ranks the order of the components of the system based on the significance of their contribution in the exergy losses. The absorber has the highest exergy loss 59.06%, basically due to the temperature difference between the absorber and the surroundings. This can be reduced by increasing the surface area of the absorber, consequently, increasing the cost of the absorber. In order to improve the performance of the cycle, special attention must be made to reducing the irreversibilities that exists in this component in the overall design. Due to the temperature difference between the heat source and the temperature of the working fluid in the generator, the next largest exergy loss occurred in the generator. To improve the second low efficiency, it must do a better job of matching the heat source with temperature of the working fluid in the generator. The exergy loss in the evaporator results mainly from the temperature difference between the environment and the evaporating refrigerant.

CONCLUSION

An energy and exergy analysis of a turbocharger Diesel engine combined with an absorption refrigerator unit has been presented. The SPICE program was used in the analysis of the energy values at different locations of the Diesel engine and computer program was written for the absorption refrigeration cycle. The second law is applied to perform an exergy analysis in the Diesel-absorption cycle. Results show that the majority of the engine irreversibility occurs within the combustion chamber of the Diesel engine and the absorber has the highest exergy loss in absorption cycle followed by the generator.

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APPENDIX (A) SILENT PARAMETERS FOR THE ENGINE

Engine make and model
CATERPILLER 3116

Type	4 stroke
Range of operation	1500-2800 rpm
Number of cylinders	6
Bore	105.025mm
Stroke	127mm
Displacement	6.6 litre
Firing order	1-5-3-6-2-4

Rotation (viewed from flywheel) Aspiration CCW
(counter clockwise)
Compression ratio 16.0:1
Type of combustion Direct
injection
Valves per cylinder 2
Valve Clearance Setting Intake
0.38mm.
Exhaust
0.64mm
Turbocharger make and model Schwitzer
S2BW
Governor type Mechanical
After cooler type JW

APPENDIX (B) ENERGY ANALYSIS:

$$Q_G = m_7 h_7 + m_1 h_1 - m_6 h_6 = m_{12}(h_{12} - h_{11}) \quad (B.1)$$

$$Q_C = \dot{m}_w (h_7 - h_8) = m_{14}(h_{14} - h_{13}) \quad (B.2)$$

$$Q_E = m_W (h_{10} - h_9) = m_w (h_{10} - h_8) \quad (B.3)$$

$$Q_A = m_s\ h_3 + m_{10} h_{10} - m_4 h_4 = m_{ss} h_2 + m_W h_{10} - m_{WS} h_4 \quad (B.4)$$

$$Q_{she} = \dot{m}_{ss}(h_1 - h_2) = \dot{m}_{ws}(h_6 - h_5) \quad (B.5)$$

Mass flow analysis:

$$m_{ss} = m_w + m_{ws} \quad (B.6)$$

or:

$$m_6 = m_7 + m_1 \quad (B.7)$$

$$m_{ws} = m_1 = m_w\left(\frac{x_{ss}}{x_{w3} - x_{ss}}\right) = m_1\left(\frac{x_6}{x_1 - x_6}\right) \quad (B.8)$$

$$m_{ss} = m_6 = m_w\left(\frac{x_{ws}}{x_{w3} - x_{ss}}\right) = \left(\frac{x_1}{x_1 - x_6}\right) \quad (B.9)$$

$$COP = \frac{\dot{Q}_E}{\dot{Q}_G} = \frac{\dot{m}_w (h_{10} - h_8)}{\dot{m}_w h_7 + \dot{m}_{ss} h_1 - \dot{m}_{ws} h_6} \qquad (B.10)$$

APPENDIX (C) EXERGY ANALYSIS:

Generator:

$$\Delta \psi_{g,ht} = Q_g \left(1 - \frac{T_0}{T_g}\right) - m_{12}(\psi_{12} - \psi_{11}) \qquad (C.1)$$

$$\Delta \psi_{g,in} = m_7 \psi_7 + m_1 \psi_1 - m_6 \psi_6 - Q_g \left(1 - \frac{T_0}{T_g}\right) \qquad (C.2)$$

$$\Delta \psi_{g,hi} = m_{12}(\psi_{12} - \psi_{11}) \qquad (C.3)$$

Condenser

$$\Delta \psi_{cm} = m_{13}(\psi_{14} - \psi_{13}) \qquad (C.4)$$

$$\Delta \psi_{c,ht} = Q_c \left(1 - \frac{T_0}{T_c}\right) - m_{13}(\psi_{14} - \psi_{13}) \qquad (C.5)$$

$$\Delta \psi_{c,in} = m_7 (\psi_7 - \psi_8) - Q_c \left(1 - \frac{T_0}{T_c}\right) \qquad (C.6)$$

Refrigerant expansion valve:

$$\Delta \psi_{v1} = \psi_8 - \psi_9 \qquad (C.7)$$

Evaporator:

$$\Delta \psi_{e,ht} = Q_e \left(1 - \frac{T_0}{T_e}\right) - m_{16}(\psi_{15} - \psi_{16}) \qquad (C.8)$$

$$\Delta \psi_{e,in} = m_{10}(\psi_{10} - \psi_9) - Q_e \left(1 - \frac{T_0}{T_e}\right) \qquad (C.9)$$

$$\Delta \psi_{cm} = m_{16}(\psi_{15} - \psi_{16}) \qquad (C.10)$$

Solution Expansion Valve:

$$\Delta \psi_{sol\text{-}valve} = \psi_3 - \psi_2 \qquad (C.11)$$

Absorber:

$$\Delta \psi_{a,ht} = Q_a \left(1 - \frac{T_0}{T_a}\right) - m_{17}(\psi_{18} - \psi_{17}) \qquad (C.12)$$

$$\Delta\psi_{a,in} = m_3\psi_3 + m_{10}\psi_{10} - m_4\psi_4 - Q_a\left(1 - \frac{T_0}{T_a}\right) \quad (C.13)$$

$$\Delta\psi_{cm} = m_{17}(\psi_{18} - \psi_{17}) \quad (C.14)$$

Solution heat exchanger

$$\Delta\psi_{she,ht} = m_1\psi_1 + m_5\psi_5 - m_6\psi_6 - m_2\psi_2 \quad (C.15)$$

Solution Pump:

$$\Delta\psi_{pump} = \frac{m_4(P_4 - P_5)}{\rho_5} \quad (C.16)$$

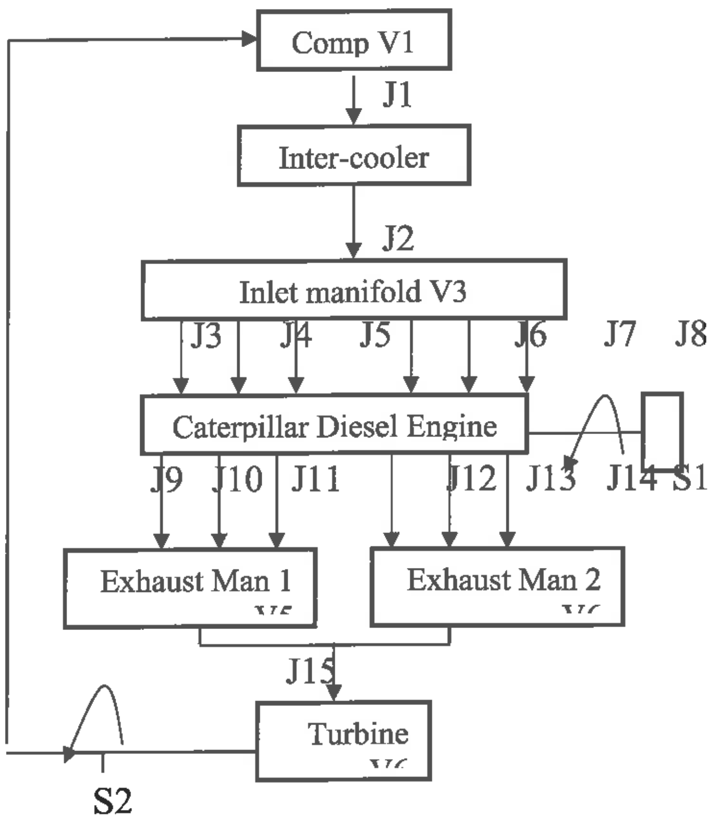

Fig. (1) Thermodynamic Control Volumes of the Engine

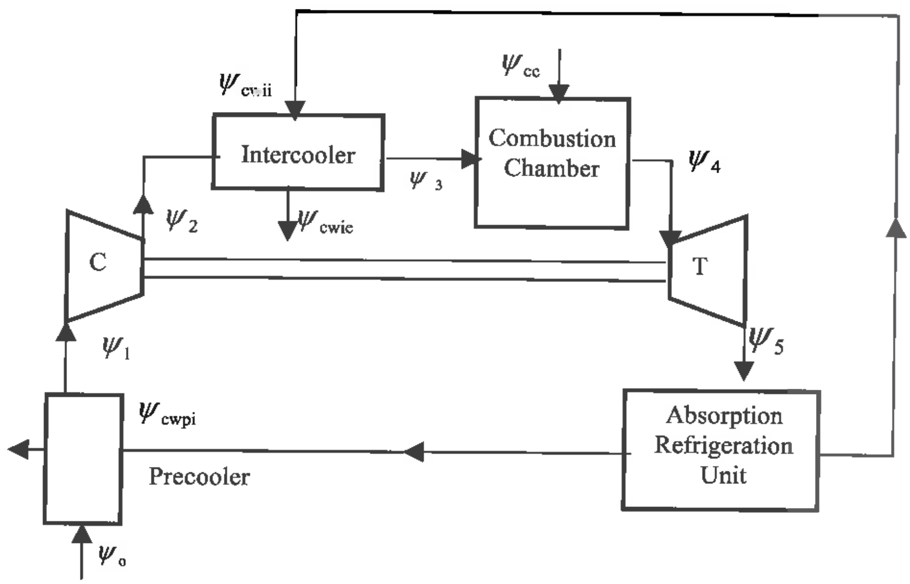

Fig. (2) Exergy flow for the Diesel engine

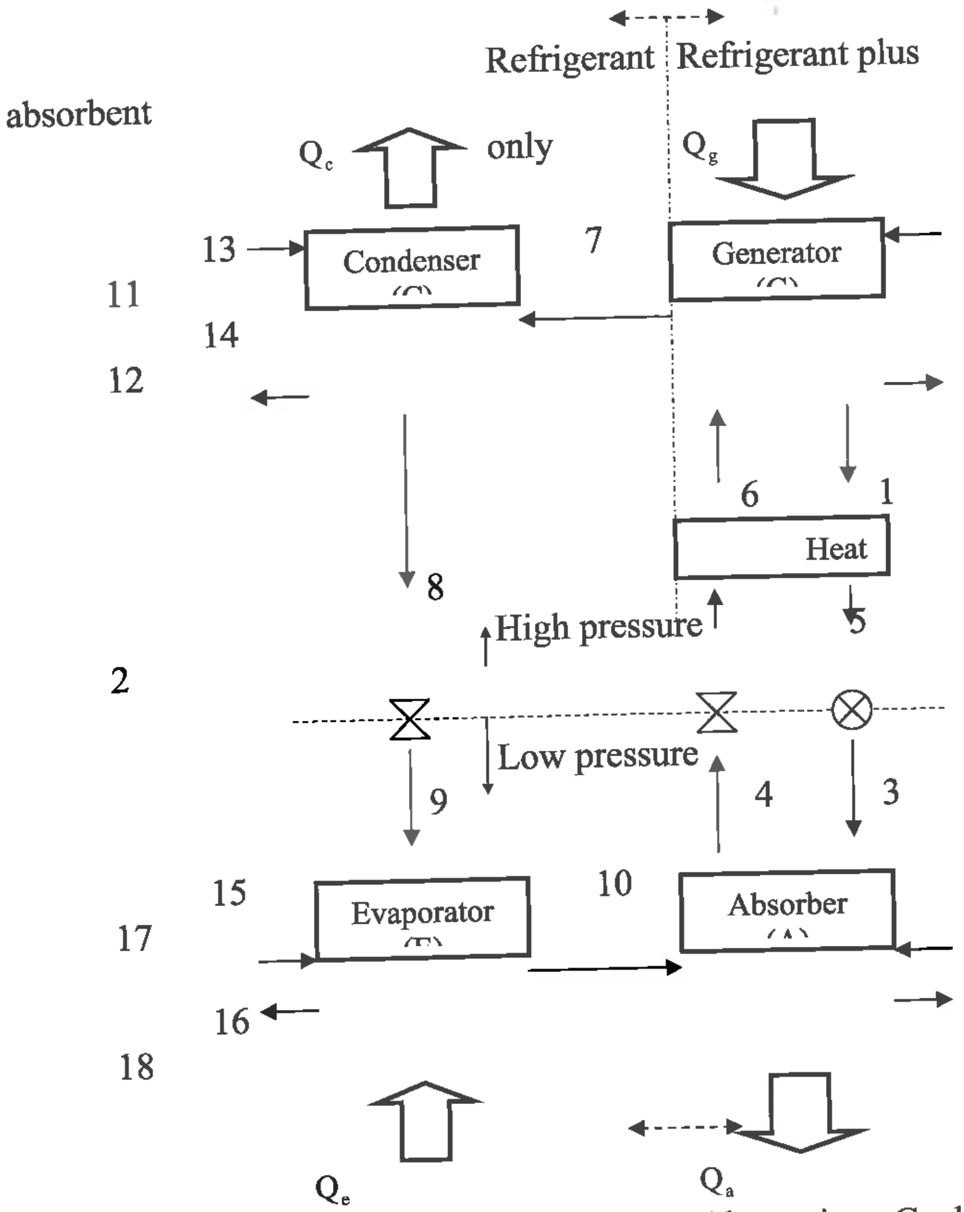

Fig (3) Energy Flow Balance for the Absorption Cycle

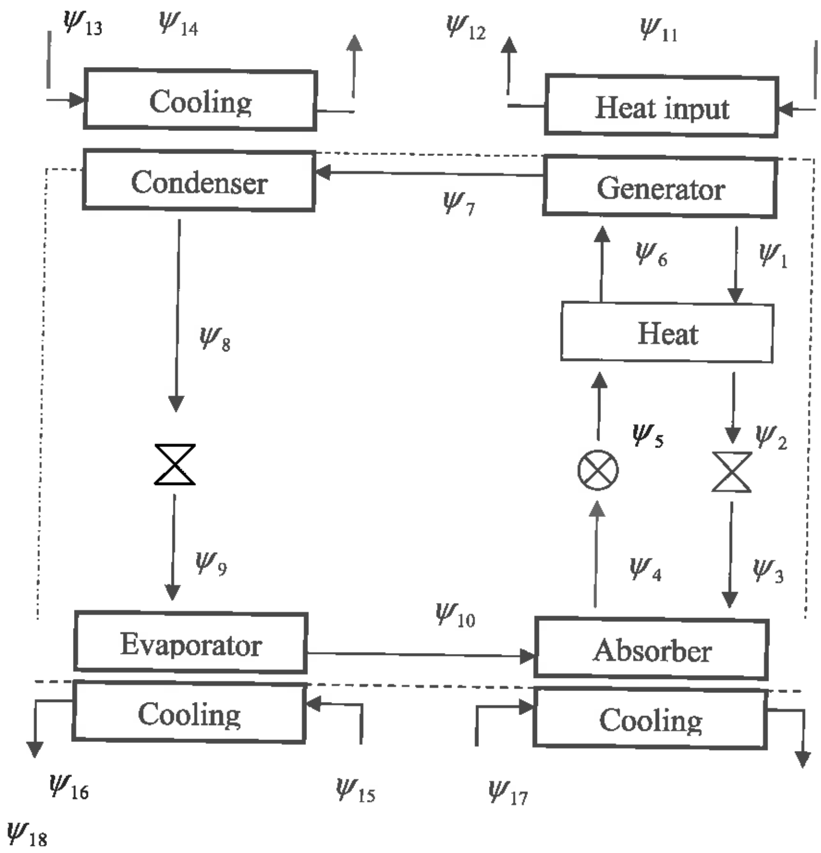

Fig. (4) Availability flow balance for the Absorption Cycle

خلاصـــــــــــــــــــــــــــــــــة :-

أن نظام التبريد الامتصاصي يشجع على زيادة البحث في هذا المجال ، حيث أن هذا النظام يمكن تشغيله بواسطة طاقة الحرارة المفقودة من محركات الديزل ولا يستخدم غازات ضارة بطبقة الأوزون .

أن الغرض من هذه الورقة هو تقديم تحليل الطاقة بواسطة القانون الأول والثاني للديناميكا الحرارية لمحركات الديزل التي تعمل بنظام الشحن التربينى المتوافقة مع وحدة التبريد الامتصاصي .

لقد تم استخدام برنامج (Spice) لتحليل قيم الطاقة عند مواقع مختلفة للمحرك وكذلك تصميم برنامج حاسوب لتحليل دائرة التبريد الامتصاصي .

هذا وأثبتت النتائج أن أعلى درجة فقدان في الطاقة تظهر في حجرة الاحتراق وسجل المكثف أعلى درجة فقدان للطاقة في دورة الامتصاص .

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