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A Performance Analysis of a Spark Ignition Engine Using Gasoline, Methanol and M90 by the Thermodynamic Second Law
가솔린, 메탄올, M90 연료를 사용한 전기점화기관에서의 열역학 제 2법칙적 성능해석

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주요용어 : 가용에너지(Available Energy), 엑서지(Exergy), 전기점화(Spark Ignition), 성능(Performance)

요 약 : 열역학 제 2법칙의 관점의 열역학적 가용에너지인 엑서지 해석법을 적용하여 가솔린, 메탄올, M90 연료를 사용한 전기점화 기관의 성능해석을 수행하였다. 열역학적 사이클 해석을 위하여 사이클을 구성하는 각 과정은 열역학적 모델로 단순화하였고, 크랭크 각도에 따른 실린더의 압력과 작동유체를 구성하는 연료, 공기 및 연소생성물의 열역학적 물성 값들을 이용하여 각 과정에서의 엑서지와 손실 일을 계산하였다. 실험데이터는 단기통 전기점화기관을 가솔린, 메탄올과 M90(메탄올 90%+부탄 10%의 혼합연료)을 연료로 WOT(Wide Open Throttle), MBT(Minimum advanced spark timing for Best Torque), 2500rpm 조건으로 운전하여 측정하였다. 계산에 이용한 자료는 실험으로 측정한 크랭크 각도에 따른 연소실의 압력, 흡입공기와 연료유량, 흡입공기 온도, 냉각수 온도와 배출가스 온도 등이다. 이를 이용하여 각 과정에서의 엑서지와 손실일을 계산하였으며 각 과정에서의 손실 일은 연소과정에서 가장 크며 팽창과정, 배출과정, 압축과정 및 흡입과정 순으로 크게 나타났다.

Nomenclature

A_{ch}	Cylinder head surface area, [m^2]
A_p	Piston face area, [m^2]
B	Cylinder bore, [m]
E_{xi}	Exergy at state i, [kJ]
h^*	Molar enthalpy, [kJ/mole]
h_g	Convective heat transfer coeffeicient, [kW/m^2 · K]
n	Number of moles
n_x	Mole numbers of residual gas
p	In-cylinder gas pressure, [MPa]
p*	Atmospheric pressure, [MPa]
Q	Heat transfer, [kJ]
S_{gn}	Produced entropy, [kJ/K]
S_i	Entropy at state i, [kJ/K]
S_T	Stroke length, [m]
T^*	Atmospheric temperature, [T]
T_w	Wall temperature, [K]
U	Internal energy, [kJ]
V_i	Cylinder volume at state i, [m^3]
V_{TDC}	Cylinder volume at TDC, [m^3]
$W_{12,rev}$	Available work, [kJ]
W_{lost}	Lost Work, [kJ]

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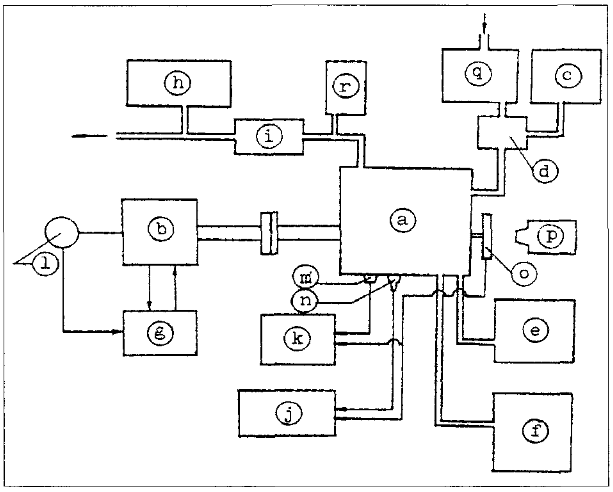
1. Introduction

The engine performance affects directly on the fuel efficiency of the vehicle. The studies for the improvement in the fuel efficiency of vehicles are concentrated on the improvement of engines, such as in-cylinder flow characteristic, operating condition, the geometric shape of the combustion chamber, etc[1-2]. Furthermore, analysis technology using flow or combustion analysis program has been developed and it makes possible the prediction of the engine performance and exhaust emissions. In additions, by using this analysis, it is possible to propose the basic direction of the engine experiments and estimate the engine performance according to the engine variables. However, many analyses have been proceeded in the view of the first law of thermodynamics, which might consider the requisite heat release from the heat engine as energy loss. Therefore, it is necessary to analyze the engine cycle on the base of the second law of thermodynamics, which optimizes the power generating system from the investigation of the available energy at each process. Although there are some studies using the second law analysis in steam turbines, gas turbines, and diesel engines, it is not easy to find the analysis of SI engine by the second law[3-11].

In this study, to examine the lost work due to irreversibility and available work at each process of the cycle, the processes of the Otto cycle are simplified and analyzed using the experimental data by the second law. The test engine is a single cylinder 4-stroke SI engine, which is operated at 2500 rpm, WOT, MBT with 3 kinds of fuel, gasoline, methanol and M90(mixed fuel of 90% methanol and 10% butane).

2. Thermodynamic model and experimental results

The combustion chamber shape of the test engine is simplified as thermodynamic model for the analysis. It is composed of the cylinder, the piston and the environment. The heat transfer of the system and the surround is completed through the system wall. The work-out from the system transfers to the surround through the piston. The system is considered an open system during the intake and the exhaust processes and a closed system during the other processes. The environmental condition is regarded as the standard one, T*=298K, p*=0.101MPa. For the analysis, the basic assumptions are chosen as follows:



ⓐ. Single cylinder engine ⓑ. Engine dynamometer
ⓒ. Fuel supplying system ⓓ. Intake system
ⓔ. Cooling system ⓕ. Lubrication system
ⓖ. Load control system ⓗ. Exhaust gas analyzer
ⓘ. Muffler ⓙ. Combustion analyzer
ⓚ. Spark advance meter ⓛ. Tachometer
ⓜ. Spark plug ⓝ. Pressure transducer
ⓞ. TDC & angle sensor ⓟ. Light source
ⓠ. Surge tank ⓡ. Manometer

Fig. 1 Experimental apparatus

Table 1 Engine specifications

Items	Specifications
Bore × Stroke (mm × mm)	85 x 86
Displacement volume(cc)	488
Compression ratio	8.5
Combustion chamber type	Semi-spherical
Max. power(kW/rpm)	12/4000
Max. torque(Nm/rpm)	33.3/2200

1) The working fluid of the cylinder is perfect gas.

2) The mean temperature of the cylinder wall is considered as constant.

3) The combustion process is perfectly completed and the emissions are composed of CO_2, and H_2O, and N_2.

4) The reservoir is the atmosphere.

5) The valve overlap is neglected.

The schematic diagram of the experimental apparatus is shown in Fig. 1. It is composed of

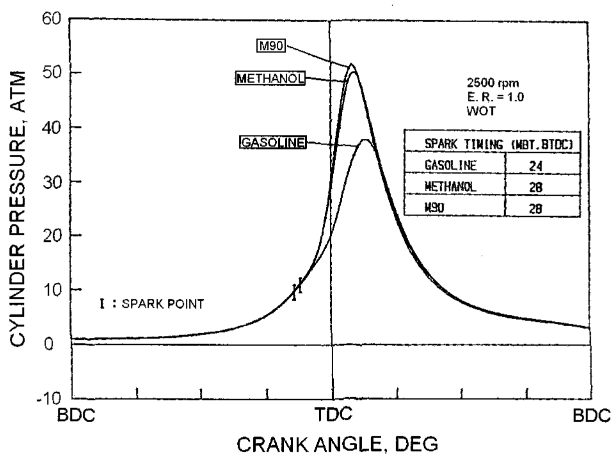


Fig. 2 Averaged cylinder pressure

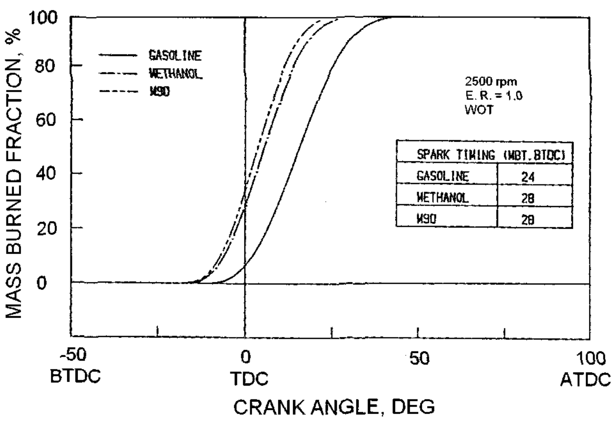


Fig. 3 Averaged mass burned fraction

Table 2 Results of engine test

Fuel	Gasoline	Methanol	M90
IMEP(kPa)	960.4	1,009.4	980.0
$p_{max,mean}$(kPa)	3,802.4	5,086.2	5,233.2
$p_{max,std.}$(kPa)	431.2	323.4	274.4
Spark timing (BTDC)	24°	28°	28°
10% MBF angle (ATDC)	2°	-6°	-7°
90% MBF angle (ATDC)	30°	18°	15°

single cylinder engine, eddy current dynamometer, air-flow meter, exhaust gas analyzer, etc. The experimental engine is a single cylinder 4 stroke SI engine with wedge shape combustion chamber. The displacement is 488cc and its bore and stroke are 85mm and 86mm. The fuel of gasoline and methanol are supplied by the electronic controlled injector, which is installed at the intake manifold. The gas fuel of butane is supplied through the mixer. Specifications of engine are described in Table 1. The cylinder pressure is measured by a piezo transducer(Kistler). The MBT spark timing is determined according to knock limit characteristics at gasoline fueled operating condition and with Methanol, due to its anti-knock characteristics, the MBT spark timing is determined when the maximum indicated mean effective pressure(p_{mi}) is acquired. Larger vaporizing latent heat of methanol reduces the temperature of the methanol-air mixture in intake manifold and causes unstable combustion characteristics. To supply enough heat for vaporizing of methanol, the coolant is circulated around the intake manifold. M90 is supplied by two fuel supplying lines according to the heating value ratio of methanol and butane(90:10). The supplying quantity of butane is controlled using micro-needle valve, gas pressure controller, vaporizer, etc.

The 100 cycle-averaged p-θ values and corresponding mass burned fraction are shown in Fig. 2 and Fig. 3. It is shown that combustion speeds of M90 and methanol are faster than that of gasoline and improve the indicated performance. The major engine test results are described in Table 2.

3. Performance analysis

3.1 Basic equations

Basic equations for the performance analysis by the second law are applied as followings. Cylinder volume and heat transfer area according to crank angle are calculated by following equations.

$$V_{TDC} = \frac{\pi}{4(r-1)} B^2 \cdot S_T \tag{1}$$

$$V(\theta) = \frac{\pi}{4} B^2 \cdot \Delta S_T + V_{TDC} \tag{2}$$

$$A(\theta) = A_{ch} + A_p + \pi B^2 \cdot \Delta S_T \tag{3}$$

Heat transfer rate through the wall is calculated according to following equations and Eichelberg's heat transfer coefficient.

$$\dot{Q}=h_g \cdot A(\theta) \cdot (T-T_w) \tag{4}$$

$$h_g = 2.1 \cdot Z(\theta)^{\frac{1}{3}} \cdot (p \cdot T)^{\frac{1}{2}} \tag{5}$$

Z(Θ) = the mean velocity of in-cylinder fluid.
p = in-cylinder pressure.
T = mean temperature of in-cylinder fluid.

The available energy is defined as the maximum work to be acquired through the reversible process 1-2 in the control volume and described as following equation.

$$W_{12,rev} = (U_1 - T^* S_1 + p^* V_1) - (U_2 - T^* S_2 + p^* V_2) \tag{6}$$

If the process 1-2 is irreversible, the available energy can be described as follows:

$$W_{12,irrev} = W_{12,rev} - T^* S_{gn} \tag{7}$$

The lost work due to the irreversibility can be derived from Eq.(6) and Eq.(7) as follows:

$$W_{lost} = W_{12,rev} - W_{12,irrev} = T^* S_{gn} \tag{8}$$

The exergy can be defined as the available energy when the state 2 of Eq.(6) is the atmospheric condition and the exergy at the state 1 can be described follows:

$$Ex_1 = (U_1 - T^* S_1 + p^* V_1) - (U^* - T^* S^* + p^* V^*) \tag{9}$$

The available energy of the reversible process 1-2 as shown in Eq.(6) can be derived with exergy of the state 1 and the state 2 as following equation.

$$W_{12,rev} = Ex_1 - Ex_2 \tag{10}$$

The above basic concept is applied at each processes of the SI engine cycle to analyze the engine performance according to operating conditions.

3.2 Available energy and lost work

The performance analysis by the second law is proceeded with available energy and lost work at each process. The temperature at each state can be calculated by the equation of the perfect gas and the exergy of the fuel is considered heat of combustion and the produced exergy during the combustion is calculated the difference of the entropy of formation between before and after combustion. The reference state of the thermal property is atmospheric condition. The mole numbers of the residual gas at the end of the exhaust process and at the start of the induction process are the reference values to determine the convergence of the calculation through the cycle and the available energy and the lost work are calculated at each process as follows:

3.2.1 Induction process(1-2)

During the induction process, the fresh mixture comes into the cylinder through the intake valve and the available energy and lost work can be derived as follows:

$$\begin{aligned} W_{12} = Ex_1 - Ex_2 = {} & n_x(u_1-u_2)_x - (n_2-n_x)(u_2-h^*)_m \\ & + p^*(V_1-V_2) - n_x T^*(s_1-s_2)_x - (n_2-n_x)Ex_{fuel} \\ & + (n_2-n_x)T^*(s_2-s^*)_m + T^* S_{2,mix} + (n_2-n_x)T^* s_i \end{aligned} \tag{11}$$

$$\begin{aligned} W_{12,lost} = T^* S_{gn,12} = {} & n_x T^*(s_1-s_2)_x + (n_2-n_x)T^* \cdot \\ & (s_2-s^*)_m - Q_{12}^* + T^* S_{2,mix} + (n_2-n_x)T^* s_i \end{aligned} \tag{12}$$

3.2.2 Compression process(2-3)

During the compression process, the mass of in-cylinder is conserved as the close system and the mole of the working fluid is n_2. The available energy and lost work can be derived as follows:

$$\begin{aligned} W_{23} &= Ex_2 - Ex_3 \\ &= n_2(u_2-u_3) + p^*(V_2-V_3) - n_2 T^*(s_2-s_3) \end{aligned} \tag{13}$$

$$\begin{aligned} W_{23,lost} &= T^* S_{gn,23} \\ &= n_2 T^*(s_3-s_2) - Q_{23}^* \end{aligned} \tag{14}$$

3.2.3 Combustion process(3-4)

During the combustion process, the reactants of the air-fuel mixture convert into the products by the rapid chemical reaction of which the combustion flame is generated from the spark plug and its combustion generates the produced entropy, $\triangle S_{comb}$. The available energy and lost work can be derived as follows:

$$
\begin{aligned}
W_{34} &= Ex_3 - Ex_4 \\
&= n_3(u_3 - h^*) + p^*(V_3 - V_4) + (n_3 - n_x)Ex_{fuel} \\
&\quad - n_4(u_4 - h^*) - n_3 T^*(s_3 - s^*) + n_4 T^*(s_4 - s^*) \\
&\quad + T^* \Delta S_{comb}
\end{aligned} \tag{15}
$$

$$
\begin{aligned}
W_{34,lost} &= T^* S_{gn,34} \\
&= n_4 T^*(s_4 - s^*) - n_3 T^*(s_3 - s^*) - Q^*_{34} + T^* \Delta S_{comb}
\end{aligned} \tag{16}
$$

3.2.4 Expansion process(4-5)

During the expansion process, the mass of the products is conserved after combustion process and the heat transfer from the products is largely occurred through the cylinder wall. The available energy and lost work can be derived as follows:

$$
\begin{aligned}
W_{45} &= Ex_4 - Ex_5 \\
&= n_4(u_4 - u_5) + p^*(V_4 - V_5) - n_4 T^*(s_4 - s_5)
\end{aligned} \tag{17}
$$

$$
\begin{aligned}
W_{45,lost} &= T^* S_{gn,45} \\
&= n_4 T^*(s_5 - s_4) - Q^*_{45}
\end{aligned} \tag{18}
$$

3.2.5 Exhaust process(5-6)

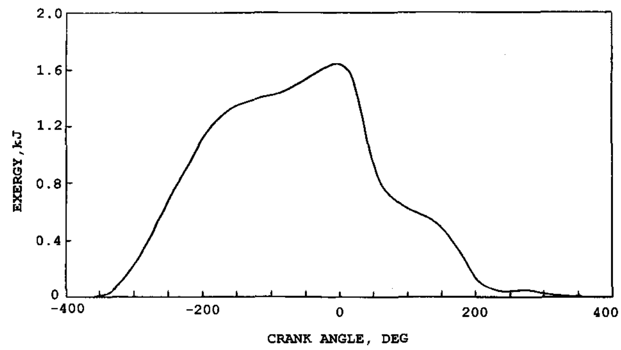
During the exhaust process, the working fluids, the products comes out the cylinder through the exhaust valve and at the end of the process, the residual gas remains in the cylinder. The available energy and lost work can be derived as follows:

$$
\begin{aligned}
W_{56} &= Ex_5 - Ex_6 \\
&= n_5(u_5 - h^*) - n_6(u_6 - h^*) + p^*(V_5 - V_6) \\
&\quad - n_5 T^*(s_5 - s^*) + n_x T^*(s_6 - s^*) + (n_5 - n_x) T^* s_6 \\
&\quad - T^* \Delta S_{comb}
\end{aligned} \tag{19}
$$

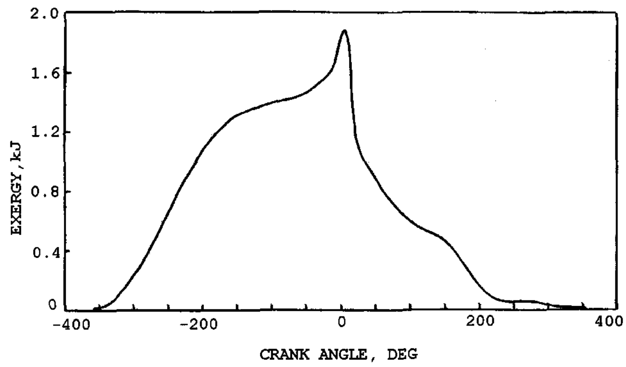
$$
\begin{aligned}
W_{56,lost} &= T^* S_{gn,56} \\
&= n_x T^*(s_6 - s^*) + (n_5 - n_x) T^* s_e - n_5 T^*(s_5 - s^*) \\
&\quad - Q^*_{56} - T^* S_{comb}
\end{aligned} \tag{20}
$$

4. Results and discussion

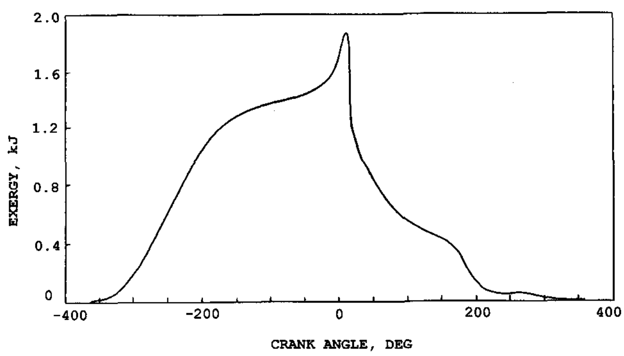
Figure 4 shows the calculated exergy according to engine operating conditions, at 2500rpm, WOT, MBT with 3 kinds of fuel, gasoline, methanol and M90. During the intake process, as fuel flows into the cylinder, in-cylinder, exergy rapidly increases. And during the compression process, in-cylinder exergy increases slowly due to work done by the piston. During the combustion process, the exergy increases and decreases rapidly due to temperature increase and irreversibility of combustion, and the work and the heat transfers with the surround. The exergy of the expansion process continuously decreases due to the heat transfer and the work-out into the surround. During the exhaust process, the working fluid comes out through exhaust valve, which causes decrease of in-cylinder exergy.



(a) gasoline



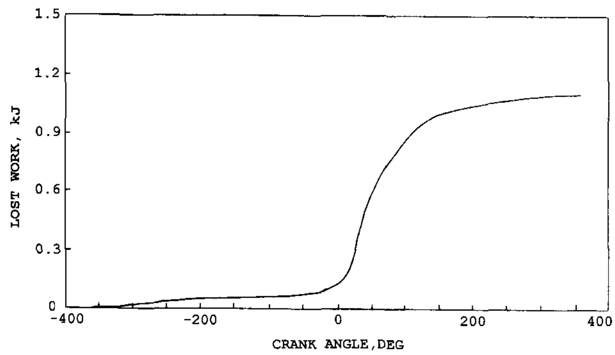
(b) methanol



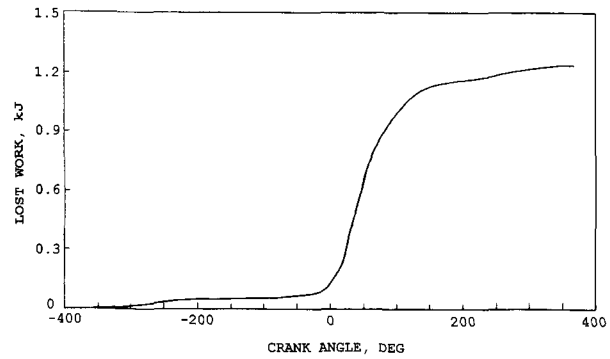
(c) M90

Fig. 4 Exergy according to fuel

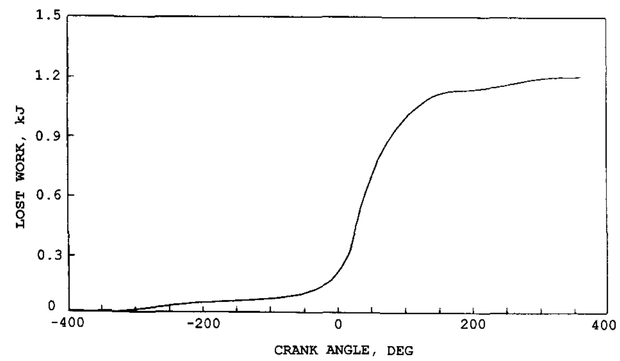
The lost work according to engine operating conditions is shown in Fig. 5. The lost work of the intake and the compression processes gradually increases. But, the lost work of the combustion and the expansion processes rapidly



(a) gasoline



(b) methanol



(c) M90

Fig. 5 Lost work according to fuel

increases, which is mainly affected by irreversibility of combustion itself during the combustion process and the heat transfer to the surround during the expansion process. Table 3 displays the result of the thermodynamic second law analysis using the exergy and the lost work according to crank angles. The compared result of the indicated performance by in-cylinder pressures and the second law analysis is presented in Table 4 and the performance difference range is from 21.8% to 31.1%.

The lost work according to each process can be reduced by the improvement of the operating conditions such as in-cylinder flow characteristics, combustion flame speed, fuel-air mixture distribution, exhaust process, etc. The improved engine performance can be derived from the reduced lost work of each process, which is the key role of the second law analysis.

Table 4 Comparison of the analysis results

Fuel	Net work by the second law	Net work by the pressure	Difference (%)
Gasoline	0.49 kJ	0.40 kJ	22.5
Methanol	0.56 kJ	0.46 kJ	21.8
M90	0.59 kJ	0.45 kJ	31.1

5. Conclusions

condition, at 2500 rpm, WOT, MBT with 3 kinds of fuel, gasoline, methanol and M90. The analyzed result of the engine performance gave the following conclusions:

1) Methanol and M90 showed favorable effects on engine performance such as indicated mean effective pressure, and mass burned fraction, compared with those of gasoline.

2) The validation of the thermodynamic second

Table 3 Analysis result by thermodynamic second law

Fuel	Available work (kJ)	Lost work (kJ) during each process					Net work (kJ)
		Intake	Compression	Combustion	Expansion	Exhaust	
Gasoline	1.67	0.05 (4.2%)	0.07 (5.9%)	0.51 (43.3%)	0.4 (33.9%)	0.15 (12.7%)	0.49
Methanol	1.85	0.06 (4.7%)	0.08 (6.2%)	0.53 (41.1%)	0.43 (33.3%)	0.19 (14.7%)	0.56
M90	1.82	0.07 (5.7%)	0.09 (7.3%)	0.5 (40.7%)	0.43 (34.9%)	0.14 (11.4%)	0.59

law analysis was confirmed with the available energy and the lost work.

3) The lost work according to each process was calculated and the magnitude of the lost work was shown in the order of the combustion process, the expansion process, the exhaust process, the compression process and the intake process.

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