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EFFECTS OF COMPRESSION RATIO ON PERFORMANCES  
OF GASOLINE ENGINE

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**ABSTRACT:** This study investigates how performance of gasoline engine and its characteristic of exhaust gas are influenced by the compression ratio. The experiments were performed in the range of compression ratio between 10 to 16 by use of the single cylinder engine which is able to change it from 5 to 22. Air fuel ratios were given optionally in the possible range which the engine employed was operated normally, where the fuel control was performed with EFI remodeled as it can operate by fake signals. Fuel employed was the regular gasoline on the market. As a result, the compression ratios that the engine performance becomes supreme are between  $\epsilon = 12-14$ . While the trend of thermal efficiency to the compression ratio crosses at  $A/F \doteq 12$ , in which the case of  $\epsilon = 10$  in  $A/F > 12$  gives the maximum, and in  $A/F < 12$  that of  $\epsilon = 14$  becomes maximum.

**Keywords:** Internal Combustion Engine, Gasoline Engine, Compression Ratio

## 1 INTRODUCTION

Although the thermal efficiency of the Otto cycle increases with the raise of the compression ratio, the gasoline engine based on the Otto cycle is restricted by the compression ratio with the problem of the gasoline knock as is well known. However, the recent remarkable progress on knowledge and technologies of engine such as the direct fuel injection in cylinder, clarification of combustion phenomena, also design, processing, control and cooling technology, enable to product the gasoline engine with higher compression ratio of  $\epsilon = 11-12$  [1], the trend of which will be considered to go for a while. On the other hand, in the previous studies on spark-ignition engine with higher compression ratio [2]-[4], there were investigated the features of thermal efficiency under higher compression ratios, and the octane number requirement, etc. However, there have been a few the studies on higher compression ratio of  $\epsilon > 13$ , in addition, their studies were performed by using high octane fuel so as to avoid the knock. Therefore, there would not have been this kind of study for a engine using regular gasoline on the market. Furthermore, it is supposed that working areas of spark-ignition engine at higher compression ratios have not yet clarified. Therefore, it will be still necessary to study how does the increase of compression ratio influence on the performance of engine.

Accordingly, the present study aims to investigate systematically the working characteristics of the gasoline engine under the higher compression ratio from 10 to 16, by using the single cylinder engine which was able to change it arbitrarily from 5 to 22. Also, the fuel employed was regular gasoline on the market.

## 2 EXPERIMENTAL APPARATUS AND METHOD

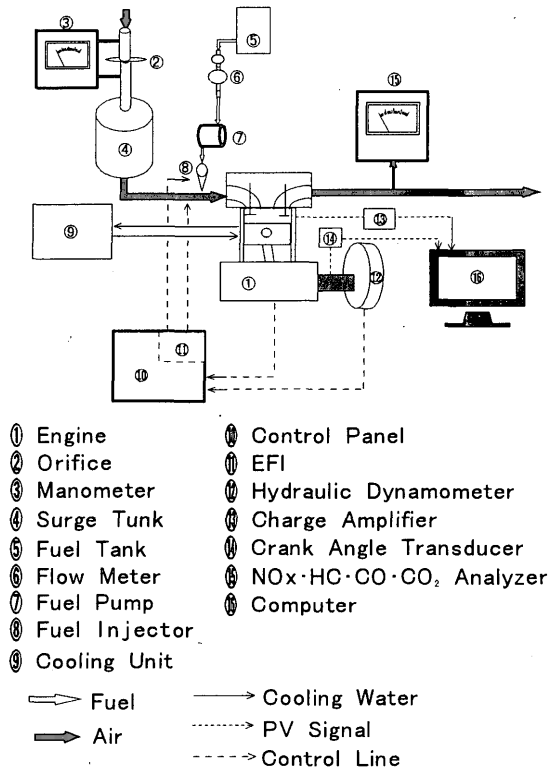


Fig. 1 Outline of Experimental Apparatus

The engine employed is the one for education which is able to change the compression ratio having the specification shown in Table 1. The compression ratio can be changed in the range from 5 to 22, where the range of  $\epsilon = 5-14$  is used as the Otto cycle engine, and the other hand  $\epsilon = 15-22$  is applied for the diesel engine. Furthermore, this is the unique engine which can be operated as the gasoline engine under the higher compression ratio.

**Table 1** Technical feature of the engine employed

<b>Type :</b> 4 cycle (Otto/ Diesel) engine with forced circulation water cooling
<b>Number of cylinder:</b> the single cylinder setting up vertically
<b>Bore and stroke:</b> 92×85 mm
<b>Swept volume:</b> 565 cc
<b>Compression ratio:</b> $\epsilon = 5 \sim 22$
<b>Maximum output:</b> 8 kW (10.9PS)/3000rpm
<b>Maximum torque:</b> 32 Nm (3.3 kg a/the m)
<b>Power supply:</b> DC 12 V
<b>Combustion chamber:</b> Flat top with parallel valve (Heron type)
<b>Fuel supply device:</b> Electric control fuel injection (EFI)
<b>Over head camshaft</b>
<b>Valve timing:</b>
Suction Valve: Open at 15 degrees before TDC. Close at 25 degrees after BDC
Exhaust Valve: Open at 10 degrees before BDC. Close at 3 degrees before TDC

sion ratios of  $\epsilon > 13$ , because it is structurally designed as the diesel engine and also both of the maximum output and the engine speed are comparatively small. Mixture gas was made by injecting gasoline in manifold using EFI (Electric control Fuel Injection), which was controlled so as to be obtained the optional air-fuel ratios by giving fake signals.

An experimental apparatus employed is system as shown in Fig. 1. Namely, this is composed of the air suction line consisting of the orifice ② and the surge tank ④, the fuel line consisting of fuel tank ⑤, flow meter ⑥, fuel pump ⑦ and fuel injector ⑧. In addition, there are the systems of operation and measurement which are consisted of the hydraulic dynamometer ⑫, the control panel ⑩, the exhaust gas analyzer ⑮, the systems measuring the engine pressure indicator and PV diagram ⑬⑭⑯. Also a device ⑨ is the water cooling unit controlled by a thermostat.

In this experiment, there were investigated the various working characteristics of gasoline engine employed under the higher compression ratio like  $\epsilon = 10, 11, 12, 13, 14$  and 16. The fixed spark advance of the engine was 10 degrees before TDC, where the spark advance during operation was varied with a centrifugal device as a function of the engine speed. Also the cooling water temperature was controlled at  $48^\circ\text{C} (\pm 2^\circ\text{C})$ , and the measurement was carried out after sufficient warming-up. The measured items were engine speed, torque, the flow rate of air and fuel, the indicator diagram and an exhaust gas analysis ( $\text{NO}_x$ , HC, CO,  $\text{CO}_2$ ). Finally, there were investigated the various working characteristics of engine employed like modified output, thermal efficiency, air-fuel ratio, fuel consumption rate, characteristics of exhaust gas, and PV diagram by making use of these results obtained.

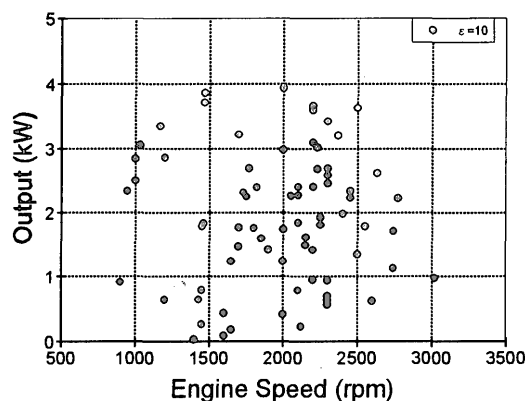
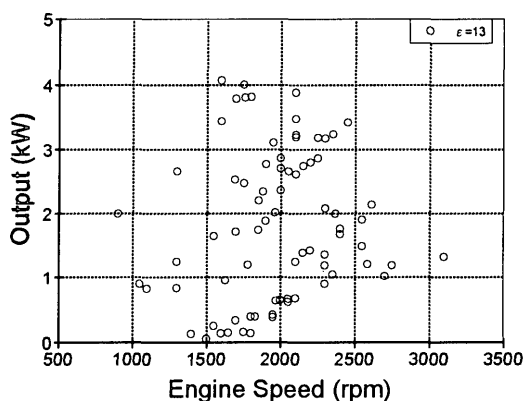
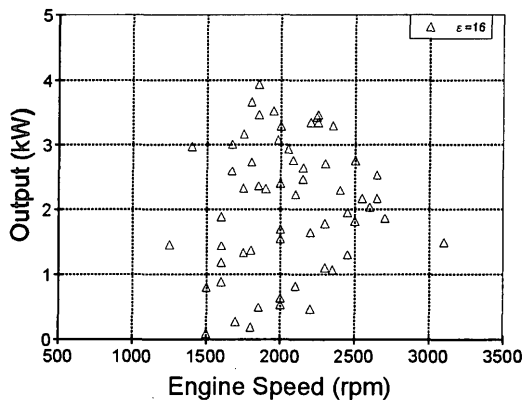
### 3 RESULTS AND CONSIDERATION

#### 3.1 Working Areas

Figs. 2-(a), (b) and (c) display the working areas of the engine employed under the higher compression ratios, taking the modified output in the ordinate and the engine

speed in the abscissa, where each figure corresponds to the case of  $\epsilon = 10, 13$  and 16, respectively. In these figures, the plotted points show the experimented conditions with the engine speed and the output operated, respectively.

Therefore, the areas occupied by these points mean to be the normal working ones without the knock. According to these figures, both the working areas of  $\epsilon = 10$  and 13 are almost same each other, but in that of  $\epsilon = 16$ , there decrease considerably both ranges of the engine speed and the output, comparing with the former cases. Therefore, these results denote that the effects of  $\epsilon$  on the working areas of the operation are not remarkable in  $\epsilon = 10 \sim 14$ , but the working area is reduced clearly in  $\epsilon = 16$ .

(a) the case of  $\epsilon = 10$ (b) the case of  $\epsilon = 13$ (c) the case of  $\epsilon = 16$ **Fig. 2** Working Areas of Engine Employed

While, as regards the actual operation of engine in the higher compression ratios, it was carried out in easy and fine condition under the compression ratio from  $\epsilon = 10$  to 12, but it became gradually difficult by the gasoline knock with increasing of  $\epsilon$ , in  $\epsilon > 13$ . Especially, the knock becomes easy to occur under the higher load operation, thereby decreasing the working areas of the engine. But noting that there are not seen the differences due to  $\epsilon$  under the low load operation.

### 3.2 Working Area For Air-Fuel Ratio

Figs. 3-(a), (b) and (c) display the working areas of engine for the air-fuel ratio (A/F), where there are the air-fuel ratio in the abscissa and the output in the ordinate. These figures correspond to those of (a), (b) and (c) in Fig. 2, respectively. Furthermore, Fig. 4 displays the envelope of each working area for  $\epsilon = 10 - 16$  obtained, insides of which are those shown in Fig. 3, respectively.

In these figures, the upper and the lower limits of A/F move from 16 to 18 and from 9 to 8, respectively, in the range of  $\epsilon$  from 10 to 14. As a result, the working area increases and reaches the maximum around  $\epsilon = 13$ . However, it is noted that, in  $\epsilon = 16$ , only the upper boundary backs to A/F = 16, but the lower one stays around A/F = 8, the working area of which decreases. Therefore, it may suggest that the combustion under  $\epsilon = 16$  becomes difficult in higher A/F like 16.

Regarding the output, the peak value increases gradually with the increase of  $\epsilon$ , reaching the maximum (4.2 kW) at  $\epsilon = 12$ , but it backs to the level of  $\epsilon = 10$  (4 kW) in  $\epsilon = 14$  and 16. Also, in these figures, A/F at the peak of output is about 12 independent of  $\epsilon$ , so that the combustion in the peak load for each  $\epsilon$  seems to be not essentially affected by  $\epsilon$ . On the contrary, the compression ratio gives the complex effects on behavior of the lower and the upper limits of A/F, as displayed by Fig. 4. Therefore, this may mean that there are some differences between the combustion around the peak load and those in the lower and upper limit of A/F. Also, points to which attention should be paid in these results are that the maximum working area of the engine employed is between  $\epsilon = 12$  and 14, and that of  $\epsilon = 10$  is unexpectedly small, being basically same as the case of  $\epsilon = 16$ . It is difficult to understand reasons why the engine employed has the larger working area in such high compression ratios as  $\epsilon = 12-14$  than that of  $\epsilon = 10$ . But, this suggests that the performances of gasoline engine will be possible to be improved, yet, in such higher  $\epsilon$  as  $\epsilon = 12-14$ .

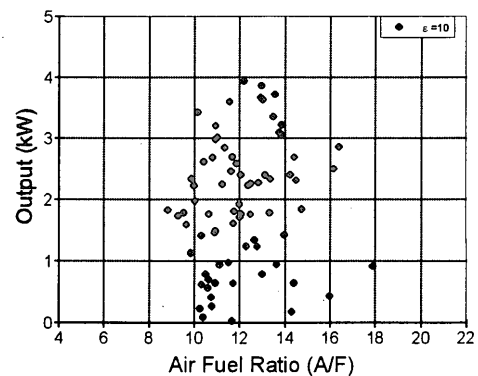
Furthermore, it notes that we could not draw the contour lines of A/F or the engine speed in Figs. 3 and 4, respectively, because the data obtained do not have systematical trends on the engine speed or A/F. It may be caused by the engine system employed which can not operate at constant engine speed, or can give optional A/F in the working area.

### 3.3 Thermal Efficiency

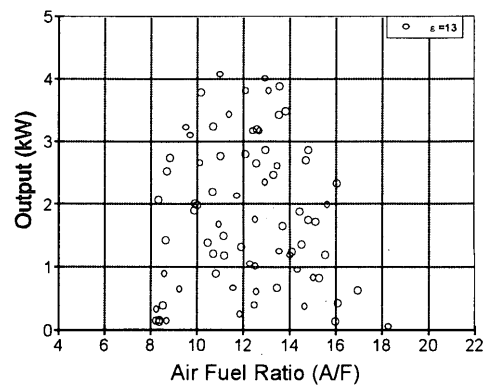
As is well known, although the thermal efficiency of Otto cycle increases as  $\epsilon$  increases, it is very difficult to operate the gasoline engine by regular gasoline at the higher compression ratio, because of the gasoline knock occurring there, usually. Therefore, there have hardly been known such as studies on the gasoline engine

operated by regular gasoline at the higher compression ratio, so far, and thus it will be significant to know how  $\epsilon$  influences the thermal efficiency of the gasoline engine employed.

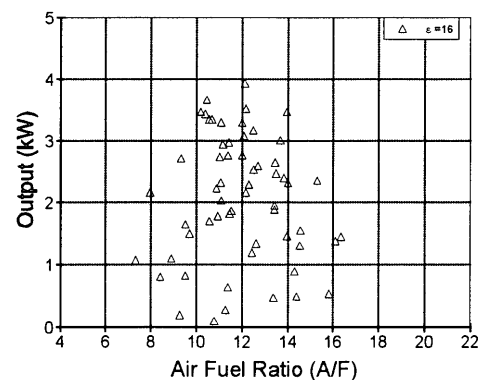
Figs. 5-(a)~(c) show the brake thermal efficiencies obtained for  $\epsilon = 10, 12$  and 16, where there are taken the brake thermal efficiency and A/F in the ordinate and the abscissa, respectively. These figures can be regarded as the working areas of the gasoline engine viewing from the brake thermal efficiency, in which it is very difficult to know the effects of  $\epsilon$  or A/F on it. Then, in order to clear them, there were attempted to show the characteristics of brake thermal efficiency by the envelopes in Fig. 6, which were drawn along the upper boundary of them displayed in Fig. 5. These envelopes



(a) the case of  $\epsilon = 10$



(b) the case of  $\epsilon = 13$



(c) the case of  $\epsilon = 16$

Fig. 3 Working area of engine employed

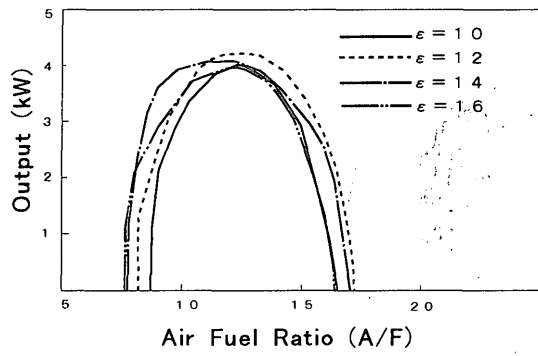
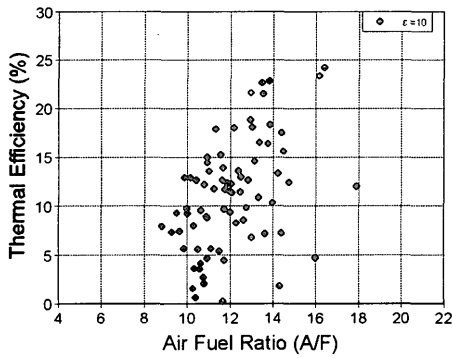
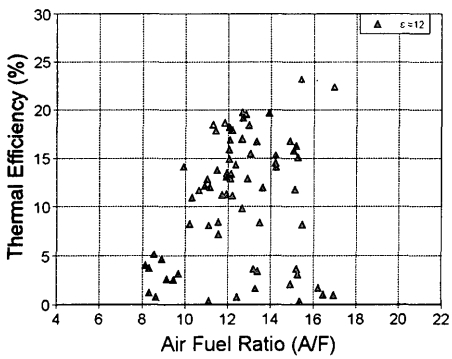


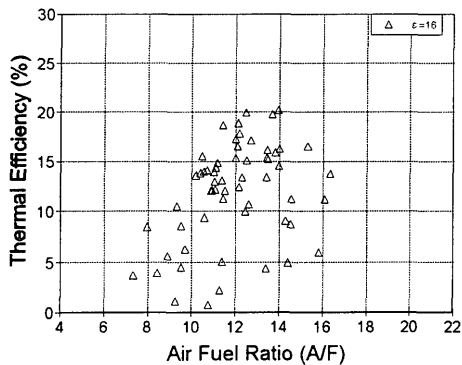
Fig. 4 Working Area



(a) the case of  $\epsilon = 10$



(b) the case of  $\epsilon = 12$



(c) the case of  $\epsilon = 16$

Fig. 5 Thermal efficiency

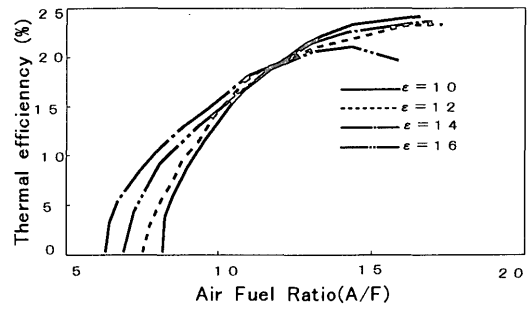


Fig. 6 Thermal efficiency

characterized the brake thermal efficiency actually depend on  $\epsilon$  and A/F, as shown in Fig. 6, which can not be described as a single curve like that of Otto cycle. Also regarding the envelopes, their tendencies are far different from that of Otto cycle. That is, in Fig. 6, the dependence on  $\epsilon$  in thermal efficiencies is reversed in front and in rear of A/F=12, namely, those become small conversely in A/F>12. Therefore, in A/F>12, the maximum thermal efficiency is obtained in A/F<12 increase except  $\epsilon = 16$ , as  $\epsilon$  increases, but  $\epsilon = 10$ , decreasing with increase of  $\epsilon$ . However, considering contribution of  $\epsilon$  to the thermal efficiency, it has larger effects on the working area of A/F rather than the thermal efficiency itself like the theory of Otto cycle.

### 3.4 Characteristics of Exhaust Gas Composition

In this study, the exhaust gas analysis was carried out for NO<sub>x</sub>, HC, CO and CO<sub>2</sub>, as mentioned above. Figs. 7 and 8 display taking up NO<sub>x</sub> and HC from 4 kinds of gases measured, respectively, where there are taken the concentrations of gases and A/F in the ordinate and the abscissa, respectively. In both the figures, the plotted results include the those of  $\epsilon = 10, 11, 12, 13, 14$  and 16 carried out, noting that the ordinate is a log scale. According to these figures, it shows that characteristics of exhaust gas are not essentially depended on  $\epsilon$ , having same tendency for every  $\epsilon$ . That is, in Fig. 7, NO<sub>x</sub> increases naturally from several to 1000 ppm, as  $\epsilon$  increases, where low NO<sub>x</sub> was given in the operations with low A/F and low output, and high NO<sub>x</sub> was given conversely in those with high A/F and high output.

While, regarding HC which has the reversed characteristics with NO<sub>x</sub>, high values of HC over 10,000 ppm occur in the operations with low A/F (A/F < 9) and

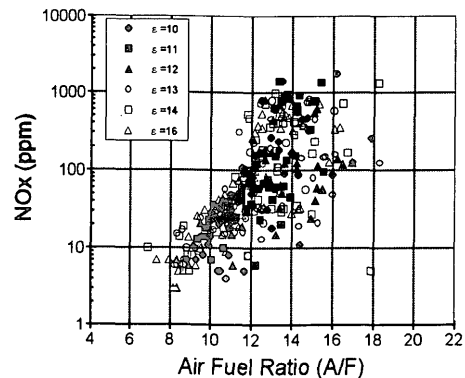


Fig. 7 Exhaust gas characteristics (NO<sub>x</sub>)

low output (under 1 kW), and then they become small with increase of A/F as shown in Fig. 8.

Furthermore, since HC relates to an amount of unburned fuel, it will be significant to know how the rate of HC influences on the engine performances. Fig. 9 shows the relations between HC and the thermal efficiency from such point of view, which are the upper boundaries for each  $\epsilon$  the same as Fig. 6. According to Fig. 9, the greater the thermal efficiency, the less HC, but there should be paid attention on two interesting features. The first is that the characteristic values of HC increase except the case of  $\epsilon = 16$ , as  $\epsilon$  increases, which become the smallest in  $\epsilon = 10$  and become the largest in  $\epsilon = 14$ , on the average. Therefore, this shows that the combustion under the high compression ratio increases an amount of unburned fuel, by which the thermal efficiency is not basically affected.

The second feature is the remarkable behavior appeared in  $\epsilon = 12-13$ , which are rapid increases of HC in the lower thermal efficiency under 7%, in addition, the very small HC in those over 10%. In this study, such the higher values of HC over 10,000 ppm cause in low A/F ( $A/F < 10$ ) with very low output and thermal efficiency, most of which occurred in  $\epsilon = 12-13$ .

Furthermore, Fig. 9 shows that although the values of HC in  $\epsilon = 14$  and 16 are very higher than those of another  $\epsilon$ , the higher HC does not give effects on thermal efficiencies. That is, this means that nevertheless the higher the value of HC, the more unburned gas increases, the thermal efficiency does not decrease. Accordingly, such the trend which the thermal efficiency increases in spite of the raise of unburned gas may be due to the effect of the compression ratio based on theory of Otto cycle.

#### 4 CONCLUSION

The performances of spark-ignition engine using the regular gasoline was investigated at the higher compression ratios, as a result, the following facts were clarified.

1. The working areas of gasoline engine employed which are seen from the relation of output and engine speed, are hardly affected by the compression ratio below  $\epsilon = 14$ .
2. The working areas for air-fuel ratio reach the maximum at  $\epsilon \approx 14$ , the range of which is between  $A/F = 8$  and 18.
3. Thermal efficiency is affected by  $\epsilon$  and A/F, the characteristics of which are reversed in front and rear of  $A/F \approx 12$ . Namely, thermal efficiency increases as  $\epsilon$  increases in  $A/F < 12$ , the trend of which is reversed in  $A/F > 12$ .
4. Characteristics of exhaust gases are not quite influenced by  $\epsilon$ . But there are found out the interesting trends between HC and thermal efficiency.

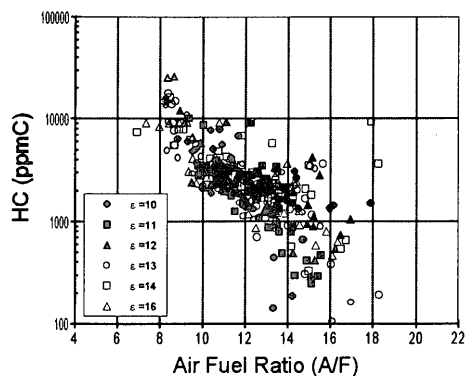


Fig. 8 Exhaust gas characteristics(HC)

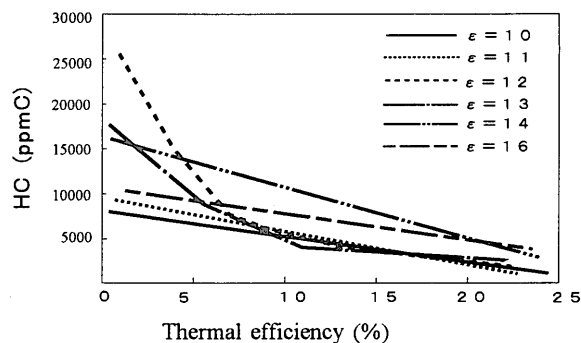


Fig. 9 HC and thermal efficiency

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