A Study on Homogeneous Charge Compression Ignition Gasoline Engines*

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A new engine concept consisting of HCCI combustion for low and midrange loads and spark ignition combustion for high loads was introduced. The timing of the intake valve closing was adjusted to alter the negative valve overlap and effective compression ratio to provide suitable HCCI conditions. The effect of mixture formation on auto-ignition was also investigated using a direct injection engine. As a result, HCCI combustion was achieved with a relatively low compression ratio when the intake air was heated by internal EGR. The resulting combustion was at a high thermal efficiency, comparable to that of modern diesel engines, and produced almost no NO_x emissions or smoke. The mixture stratification increased the local A/F concentration, resulting in higher reactivity. A wide range of combustible A/F ratios was used to control the compression ignition timing. Photographs showed that the flame filled the entire chamber during combustion, reducing both emissions and fuel consumption.

Key Words: Gasoline Engine, Ignition, Fuel Injection, Premixed Combustion, Thermal Efficiency

1. Introduction

The homogeneous charge compression ignition (HCCI) engine is more efficient and produces lower emissions than conventional internal combustion engines. HCCI engines can operate with high thermal efficiencies that are comparable to those of diesel engines and well above those of SI engines. Nitrogen oxide (NO_x) emissions from HCCI engines are the same or less than those produced by SI engines with three-way catalysts, and significantly lower than those produced by diesel engines⁽¹⁾⁽²⁾.

The main problem with HCCI is that the ignition is completely controlled by chemical kinetics, and is therefore affected by the fuel composition, equivalence ratio, and thermodynamic state of the mixture⁽³⁾. There are no external controls such as the fuel-injection or spark timing that are used on diesel

or SI engines. Another problem with HCCI is the low BMEP. The relatively high intake air temperature and very lean air to fuel (A/F) ratios reduce the output compared with SI engines.

Our concept is to employ compression ignition for low load operations, and to switch to conventional spark ignition for brief periods when high power is required. This concept satisfies the requirements of high efficiency, low emissions, and high power in the same engine. The compression ratio was chosen to avoid knocking when operating under a full load using SI, and was not high above typical ratios for SI engines.

2. Ignition Control

Thermal efficiency generally reaches a maximum for a compression ratio of 15. A ratio of 13 seems to be the practical upper limit when switching to a conventional spark ignition for high loads. Here, control of auto-ignition was studied using a compression ratio of 12. A numerical simulation code GT-Power, was used to predict the conditions for

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auto-ignition. Three special engine features were considered.

First, inlet air was pre-heated by the residual gas. The timing was set to close the exhaust valve early and to delay the inlet opening so as to retain the residual gas in the combustion chamber. The gas was then compressed and expanded again⁽⁴⁾⁽⁵⁾. A high gas temperature was maintained until it was mixed with fresh gas in the compression phase. The results of this effect were predicted by numerical simulation and are shown in Fig. 1.

Second, the effective compression ratio was controlled. The timing was changed to close the inlet valve early, thereby raising the effective compression ratio by as much as 2. Numerical predictions of this feature are shown in Fig. 2. The effect on the pressure rise is relatively small, but does exist at speeds below 2000 rpm. The valve timing of the calculated cams is shown in Table 1.

Third, a direct injection was used to make stratify mixture formation during compression ignition. The stratification of the mixture increased the local A/F concentration, resulting in higher reactivity. Stratification also reduced the heat loss caused by combustion at the chamber wall. The wide range of combustible A/F ratios for the thermo-kinetic explosion enabled the overall heat release characteristics to be controlled.

3. Results And Discussion

3.1 Experimental Setup

A schematic of the experimental setup is shown in Fig. 3, and the engine specifications are given in table 2. Direct gasoline injection was used for the fuel

supply system. A blended fuel consisting of 50% iso-octane and 50% n-heptane was also used. Compression ratios of 12.0, 14.8, and 18.0 were tested. The valve timing of the tested cams, A, C, and D, are shown in Table 1.

Fig.1 Temperature of compressed gas

Fig.2 Pressure of compressed gas

	STD	A	B	C	D
EVO(° BBDC)	50	50	50	50	5
EVC(° ATDC)	10	-30	-45	-75	-30
IVO(° BTDC)	10	-30	-45	-75	-30
IVC(° ABDC)	46	50	50	50	5

Engine Type	4-stroke Single Cylinder
Fuel Supply	Direct Injection
Bore x Stroke	φ 96.9 x 75 mm
Displacement	553cm ³
Compression Ratio	12, 14.8, 18
Fuel	Regular Gasoline 50%Iso-Octane+50%n-Heptane

Fig.3 Experimental apparatus

3.2 Combustion characteristics

The fuel consumption and exhaust emissions versus the A/F ratio are shown in Fig. 4. The best ISFC was obtained between $36 < A/F < 40$. Almost no NO_x emissions or smoke (not shown on the graph) were observed. The THC emissions were comparable to those of SI engines, unlike most HCCI engines. The

Fig.4 Emissions & fuel consumption characteristics
(Engine speed: 1000rpm, Compression ratio: 12,
Cam: D, WOT, Fuel: 50%iso-Octane+50%n-Heptane,
Injection timing: 300deg.BTDC)

Fig.5 Temporal change of heat release rate for A/F

heat-release curve obtained for these conditions is shown in Fig. 5. When the A/F ratio was richer than 36, the ignition was too far advanced, leading to engine knock. When the A/F was leaner than 42, the ignition was too retarded, causing a misfire. A basic characteristic of HCCI is that the ignition phase is varied by the A/F ratio.

Figure 6 shows possible HCCI combustion regions with a compression ratio of 12, when the fuel and cam are varied. HCCI combustion was achieved at each plotted data point. The region corresponding to gasoline had a higher IMEP than that of the blended fuel, and the region corresponding to the large negative overlap cam had a lower IMEP than that of the small negative overlap cam.

A tendency of narrower regions for higher engine speeds was also observed. Therefore, HCCI combustion occurred when gasoline was used with a compression ratio of 12. The engine load could be controlled by the negative overlap, and the HCCI

Fig.6 Region of possible HCCI combustion
(Compression ratio:12)

Fig.7 Region of possible HCCI combustion
(Compression ratio:18)

region could be expanded while maintaining controlled combustion.

Figure 7 shows regions of possible HCCI combustion with a compression ratio of 18. The fuel type and cam were fixed. The HCCI region was between 700 and 1000 rpm, and around 4000 rpm. However, auto-ignition did not occur between 2000 and 3000 rpm when the compression ratio was 18 and gasoline was used as fuel.

The following mechanisms of two-stage ignition and the negative temperature coefficients of the ignition delay time can explain this phenomenon. In regions with negative temperature coefficients, a temperature increase will cause an increase, rather than the usual decrease, of the ignition delay time⁽⁶⁾⁽⁷⁾.

ROHR curves are shown in Fig. 8 for each case given in Fig. 7. Two-stage ignition occurs at 700 rpm; single-stage ignition occurs at 4000 rpm. The range of 2000 to 3000 rpm is just inside the region of negative temperature coefficients. Auto-ignition occurs again at 4000 rpm, which is beyond the region of negative temperature coefficients.

Fig.8 Difference of R.O.H.R. curve
(Compression ratio: 18, Cam: A, WOT, Fuel: Gasoline,
Injection timing: 300deg.BTDC)

3.3 Ignition control by mixture stratification

Theoretically, it is possible to control the ignition phase using only the negative overlap. However, under actual driving conditions there are several concerns about transient conditions such as the mechanical delay of the variable cam and the effect of the previous cycle of combustion. Therefore, we concentrated on the stratification of mixture formation. Fig. 9 shows a large expansion of the compression ignition region towards lean A/F ratios. The injection timing of each data point was adjusted to obtain the best IMEP.

The control of the ignition phase is explained in Fig. 10. A homogeneous mixture with A/F = 42 exhibited a remarkable ignition delay that reduced the IMEP. Retarding the injection timing led to an advance in ignition timing. It is important that the cam and A/F ratio remain constant. If the A/F ratio of a homogeneous mixture was leaner than 45, misfiring occurred; however, the same A/F ratio in a stratified mixture produced stable combustion.

Fig. 11 shows the ISFC versus the engine load for cams C and D, with either a fixed homogeneous mixture or a stratified mixture. Even though the ISFC of the stratified mixture was slightly worse, the values were much better than those obtained from SI engines.

Fig. 12 shows the NO_x emissions for each test condition. The emissions from the stratified mixture were also slightly worse than those obtained from the homogeneous mixture; however they were better than the emissions from SI engines. This is the result of lean, high EGR combustion.

Controlling combustion using the fuel injection timing is fast and reliable. If this was applied to actual engines for rapid transient operations, using

Fig.9 Expansion of operating range by mixture stratification
(Engine Speed: 1000rpm, Cam: D, WOT
Fuel: 50%iso-Octane+50%n-Heptane)

either several-stage or continuously variable cams, high thermal efficiencies and low emissions would be achieved over a wide range of operations.

Fig.10 Control of ignition by mixture stratification
(A/F:42 Other Conditions: See Fig.9)

Fig.11 Operating range and fuel consumption
(Conditions: See Fig.8)

Fig.12 Comparison of NOx amount for each condition

4. Photographic Studies

A transparent single-cylinder engine was used for the photographic studies. An extended piston permits viewing through a sapphire window mounted in its top. The combustion chamber could be observed from underneath the extended piston. The specifications of this engine were the same as those of the previous test engine. Photographs were taken using an image converter camera (IMACON 468).

The upper portion of Fig. 13 shows a sequence of high-speed movie frames of the SI combustion process. The irregular shape of the turbulent flame front is apparent. The middle portion of Fig. 13 shows HCCI. Auto-ignition was distributed throughout the cylinder volume. As the mixture in the hottest region burned, the expanding gases caused further compression of the unburned mixture and subsequent ignition occurred without flame propagation.

The lower portion of Fig. 13 shows stratified charge compression ignition. Stratification of the mixture increased the local A/F concentration, resulting in higher reactivity; auto-ignition occurred in such regions. These auto-ignition regions ignited other regions in which mixing was not yet complete. Although over-rich mixture appeared as a bright flame, smoke was not detected.

The formation and combustion of soot occurred inside the rich zone. Large amounts of radicals, CO, and fresh air were introduced into this region; therefore, the soot generated was burned-up in a short period⁽⁸⁾.

5. Conclusions

- (1) A new engine concept was introduced, consisting of HCCI combustion for low and midrange loads, and spark ignition combustion for high loads.
- (2) A high thermal efficiency was attained, comparable to those of modern diesel engines, with almost no NOx emissions or smoke.
- (3) HCCI combustion was observed using gasoline and a compression ratio of 12; the engine load was controlled by the amount of negative overlap.
- (4) Mixture stratification from direct injection extended auto-ignition into the low-load region.
- (5) A two-stage ignition process was observed over a range of low speeds, and a single-stage ignition process was observed at a higher speed. This is a key factor for controlling auto-ignition and combustion.

Fig.13 Flame development process for each ignition type
(Engine Speed:700rpm, Compression ratio:10, Cam:A, Fuel: 100%n-heptane)

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