한국자동차공학회논문집 제 4 권 제 6 호, pp.65~77, 1996.

〈논 문〉 SAE NO. 96370120

Combustion Characteristics of a Small Diesel Engine Converted to Spark Ignition Operation and Fuelled with Natural Gas

디젤 기관을 개조한 소형 전기점화식 천연가스기관의 연소 특성 연구

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ABSTRACT

A small-sized industrial diesel engine was converted to a spark ignited engine and then adapted for fuelling with natural gas. After conversion work, general combustion characteristics of the gas engine(such as ignition delay, main and total combustion durations, and heat release characteristics) were studied as a function of major engine operating variables such as air to fuel ratio, spark timing, and spark plug type. Some other studies on cyclic variation characteristics in IMEP, Pmax and (dp/dθ)max, and also optimum combustion phasing angle were performed.

Key Words : Combustion characteristics, Ignition delay, Combustion duration, Heat release characteristics, Cyclic variation, Optimum combustion phasing angle

INTRODUCTION

Considerable numbers of gas engines have been already utilized since around at the end of the last century. In recent years, the necessities of oil products conservation, energy security, and environmental protection have been giving much stronger motivations and inevitable necessities for the development of gas engines. Moreover, natural gas has lots of advantages as a fuel for internal combustion engines; due to low cost, abundent amount of resources, excellent chemical and physical properties in the point of security and fuel itself, and one of the most hopeful fuels for improving actual air quality problems from the point of short-term basis. The conversion/adaptation of a diesel engine into a spark ignited engine is well known to one of the most effective methods for the application of natu-

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ral gas for internal combustion engines. From this point of view, we performed the systematic study concerning this subject.

In this study, a small-sized industrial diesel engine was converted to a spark ignited engine and then adapted for fuelling natural gas. After conversion/adaptation work, general combustion characteristics such as ignition delay, main and total combustion durations, heat release characteristics and cyclic variation characteristics in IMEP, Pmax and $(dP/d\theta)max$ were studied and some studies on optimum combustion phasing angle were also performed as a function of major engine operating variables including air to fuel ratio, spark timing, and spark plug type for the converted spark ignited engine.

BASELINE DIESEL

Table 1 gives the general characteristics of the direct injection type air cooled four stroke baseline diesel engine.

Table 1 General Characteristics of Baseline Diesel

Maker/Model	ALSTHOM /Dieselair 272 A
Bore/Stroke	103mm/116mm
Total Swept Volume	1,930cc
Number of Cylinders	2 Cylinders in Line
Compression Ratio	16 : 1
Engine Speed(RPM)	1,100~1,800
Maximum Power(kW)	18.5
Cylinder Phasing Angle	180° Crank Angle
Firing Order	2~1
Injection Pressure	200Bar

SPARK IGNITION CONVERSION[1),2)]

A bowl-in piston type asymetrical compact combustion chamber was adopted to assure the fast burn of air-gas mixture[3)~6)]. The compression ratio was lowered to 12:1[5),7)] and Table 2 shows the geometric characteristics of the modified S. I. engine. The valve event of baseline diesel, which is close to that of typical gas engines with small valve over lap, was adopted without any modification[8),9)]. A transistorized ignition system by Hall Generator(Bosch T2-H) was used and a typical type spark plug was installed for each cylinder in the injector mounting hole positioned near the axis of the combustion chamber[10)~13)]. The converted gas engine was fuelled through a variable restriction type carburettor(IMPCO CA 50) fixed on the intake manifold[14)].

TEST SYSTEM LAYOUT

Fig.1 shows schematic diagram of engine test bed which contains all the measuring and analyzing equipments and tools, the positions and items for measurement.

ENGINE OPERATION CONDITIONS

- Engine speed: 1,300rpm
- Engine load: WOT(Wide open throttle)
- Excess air ratio: from $\lambda=1.0$ to $\lambda=1.6$
- Spark timing: MBT(Minimum spark advance for Best Torque)

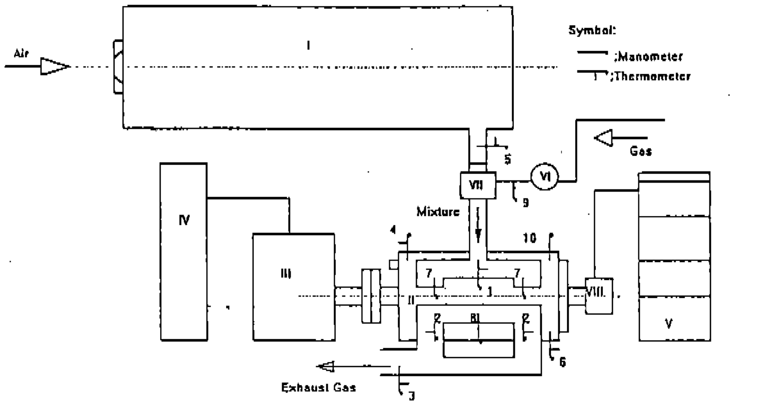
NATURAL GAS

City gas supplied for northern area in France were used for this study. The principal element of the used natural gas is Groningen gas and several different sources such as Rus-

Table 2 Modified Combustion Chamber for S. I. Gas Engine

Part	Volume (CC)	
	Baseline Diesel	Modified S. I. Gas Engine
Combustion Chamber Shape	Toroidal asymetrical chamber in pistion head	Asymetrical bowl in piston
Main Chamber	47.5	70.4
Squish Height	10.0	10.0
*Cylinder Head	7.0	7.0
Total Combustion Chamber	64.5	87.4
Cylinder Volume	965.0	965.0

* Cylinder Head Volume : Total Recess Volume for Intake & Exhaust Valves and Spark Plug



• LEGEND

1). EQUIPMENT:

I. Air Box
II. Engine Assembly
III. Eddy Current Engine Dynamometer
IV. Dynamometer Console
V. Devices for Combustion Analysis (Indimaster, Charge Amplifier, Oscilloscope)
VI. Gas Flowmeter
VII. Carburetor
VIII. Optical Encoder
VIIII. Inductive Pick-up
X. Pressure Transducer

2). MEASUREMENTS:

1. Thermocouple (Mixture temperature in intake manifold)
2. Thermocouple (Exhaust gas temperature for cylinder no. 1and no. 2)
3. Thermocouple (Exhaust gas temperature for common)
4. Thermocouple (Combustion chamber temperature for cylinder no. 1)
5. Thermocouple (Inducing air temperature)
6. Thermocouple (Lubrication oil temperature)
7. Manometer (Intake manifold pressure for cylinder no. 1 and no. 2)
8. Manometer (Exhaust manifold pressure)
9. Manometer (Gas supply pressure)
10. Manometer (Lubrication oil pressure)
11. Environmental Condition (Pressure, Temperature, Humidity)

Fig.1 Schematic diagram of engine test system

sian and Algerian gases are mixed to increase its heating value and decrease the volumetric nitrogen percentage. The used natural gas is finally composed of about 84% of methane, 10% of nitrogen and 6% of other hydro-carbons respectively on the volume basis. Table 3 shows the averaged phisycal properties of this natural gas.

GENERAL COMBUSTION CHARACTERISTICS

We used a computerized approach associated with pressure-crank angle diagrams to calculate[6),15)16)] :

The heat release rate;

Tabel 3 Major Physical Properties

Properties	Groningen Gas	Used Natural Gas
Heating Value(kJ/Nm³)		
High/Low	35,028/31,542	37,080/33,372
Wobbe Number	1,176	1,245
Specific Gravity	0.825	0.826
Specific Gravity with Respect to Air	0.635	0.639
Stoichiometric Air to Fuel Ratio	12.95	13.82
Inflammation Temperature(℃)	660	—
	129/89	—

Data Source : Gaz de France

$$\frac{dQ}{d\alpha}=(\frac{\kappa}{\kappa-1})p\frac{dV}{d\alpha}+(\frac{1}{\kappa-1})V\frac{dp}{d\alpha}$$

The net heat released during combustion;

$$Q=\int_{\alpha A}^{\alpha}(\frac{dQ}{d\alpha})\,d\alpha$$

The mass burned fraction;

$$x=\frac{1}{Q_M}\int_{\alpha A}^{\alpha}(\frac{dQ}{d\alpha})\,d\alpha$$

with:

$$Q_M=\int_{\alpha A}^{\alpha f}(\frac{dQ}{d\alpha})\,d\alpha,$$

maximum net heat release

α, crank angle

α_A, crank angle corresponding to the start of combustion

α_f, crank angle for the end of combustion

p, pressure

V, volume

κ, polytropic coefficient

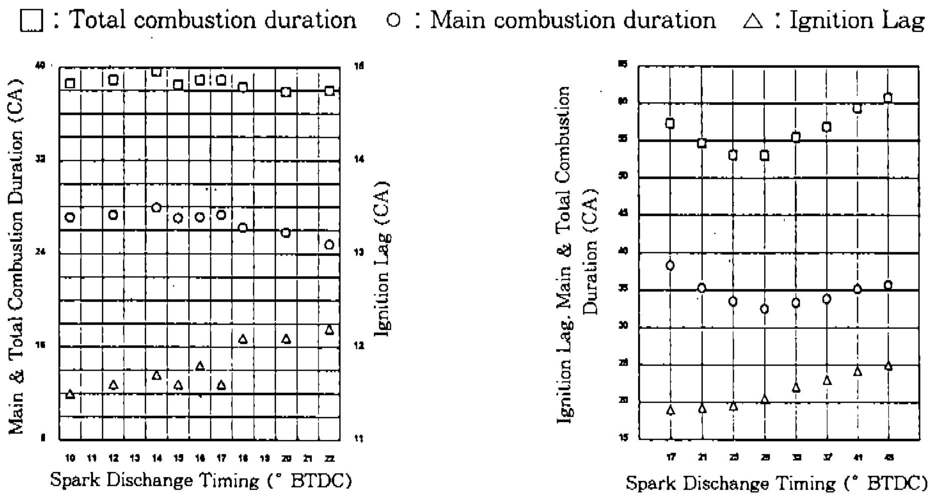
From an experimental logarithmic (p-V) diagram, we obtained k=1.27 which is the value currently admitted for spark ignition engines and corresponds to an intermediate value between burned and unburned gases. Ignition lag, which is the time between the spark occuring and the first development of a flame kernel producing a significant energy release to initiate the flame front[6),17)], was defined as 5% of mass burned fraction. Main combustion duration could be defined as the time between the flame initiation point(5% mass burned fraction) and the end of actual combustion(90% mass burned fraction)[5),18),19)]. Total combustion duration is the sum of ignition lag and main combustion duration.

Effect of Spark Plug

The present study was performed for 6 kinds of spark plugs which have three different kinds of heat ranges(cold, medium, medium hot) and gap widthes(0.6mm, 1.0mm, 1.5mm). No remarkable differences in Pmax and Pmax position between spark plugs were observed both for soichiometric and lean mixtures. Nevertheless, the coldest spark plug showed a little lower Pmax at a slightly retarded position particularly with the lean mixture $\lambda=1.5$[3),20)]. With stoichiometric mixture, there was no differences in ignition lag and main combustion durations. With lean mixture $\lambda=1.5$, a small but distinct decrease tendency in ignition lag and total combustion duration and nearly constant main combustion duration were observed by widening the gap width or ascending the heat range of spark plug. These results point out that we can use the medium heat range spark plug for the modified S. I. engine with no significant consequences for excess air ratios between $\lambda=1.0$ and $\lambda=1.5$.

Effect of Spark Timing

Spark timing was varied from 10° to 22° BTDC for stoichiometric mixture and between 17° and 45° BTDC for lean mixture. The ignition lag and combustion duration variations are presented in Fig.2 as a function of spark timing. With stoichiometric mixture, the igni



(a) with stoichiometric mixture (b) with lean mixture($\lambda=1.5$)

Fig.2 Combustion characteristics as a function of spark discharge timing

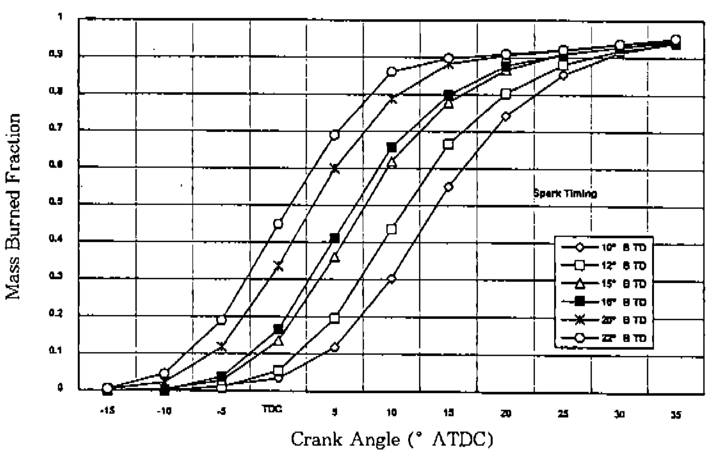


Fig.3 Mass burned fraction versus crank angle as a function of spark discharge timing with stoichiometric mixture

tion lag varies in a very narrow range of about 1° CA but it shows slight increasing tendency with spark advance[17),19)]. On the other hand, the main combustion duration has nearly the same value with over-delayed spark timings and it decreases with over-advanced spark timings on the basis of 16° BTDC(MBT). The total combustion duration shows nearly the same value(38°~39.5°CA) with all spark timings. With the lean mixture $\lambda=1.5$, some clear variation trends on combustion duration characteristics are observed[17)~19)];a distinct increase of ignition lag with spark advance (from 19° to °25CA) and a considerable increase of main and total combustion durations (maximum differences of 6° and 8°CA respectively) both with advancing and retarding spark timing on the basis of 29° BTDC (MBT). In Fig.3, we can see no difference in the slope of the mass burned fraction curves which means no combustion speed difference with spark timings and consequently no noticable variations of the combustion duration as mentioned before in case of stoichiometric mixture.

Effect of Exess Air Ratio

Fig.4 shows the variation of the combustion characteristics(ignition lag, main and combustion durations) versus excess air ratio. By leaning out mixture strength, the main combustion duration curve rises very slowly between $\lambda=1.0$ and $\lambda=1.3$(about 1°CA), but it grows rapidly with much steeper grade(about 11°CA) for the excess air ratios in the range of $\lambda=1.4$ and $\lambda=1.6$[21)]. The lgnition lag curve also shows similar variation trends to main combustion duration curve, but with greater increase grade, especially for excess air ratios between $\lambda=1.4$ and $\lambda=1.6$. On the other hand, the indicated output efficiency shows its maximum value with the mixture of $\lambda=1.3$ and the indicated mean effective pressure decreases nearly linearly with leaning out mixture strength as seen in Fig.5.

Heat Release Characteristics

With advancing spark timing, the peak cylinder pressure increases and its position is also advanced. The maximum heat release rate reaches its maximum value at MBT and

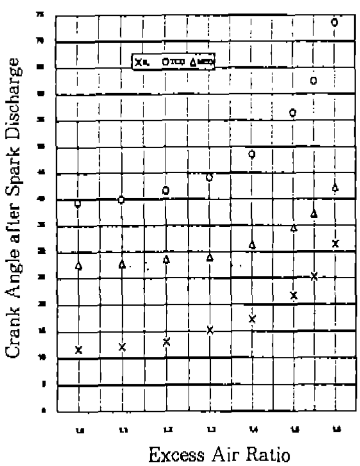


Fig.4 Combustion characteristics as a function of excess air ratio

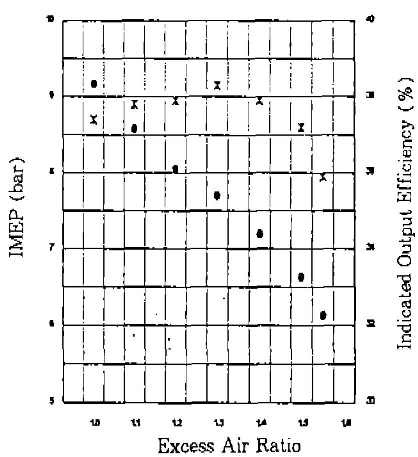
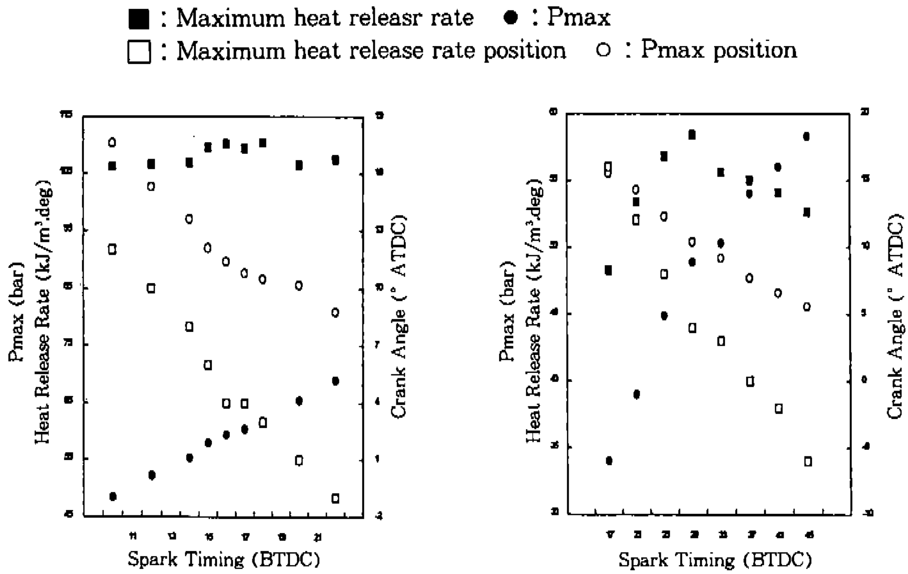


Fig.5 IMEP and efficiency as a function of excess air ratio



(a) with stoichiometric mixture (b) with lean mixture(λ=1.5)

Fig.6 Heat release rate and Pmax versus spark timing

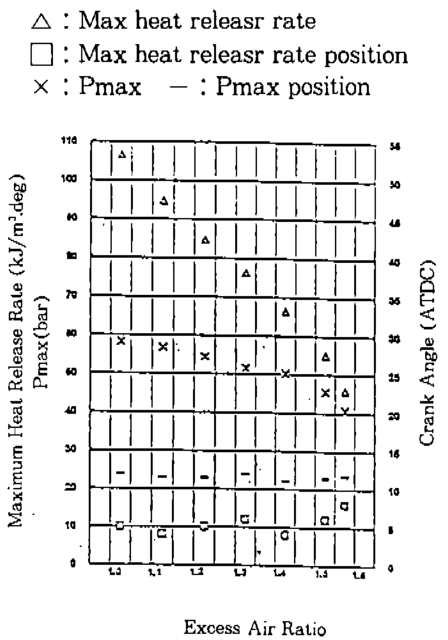


Fig.7 Variations of maximum heat release rate as a function of excess air ratio

it decreases from this point with advancing or retarding spark timing. The position of maximum heat release rate is advanced with spark advance just like the peak cylinder pressure position as seen in Fig.6[22]. All the trends mentioned above, especially that for maximum heat release rate can be observed much more clearly with the lean mixture λ=1.5. The peak cylinder pressure and maximum heat release rate decrease steadily when excess air ratio is increasing but the positions of Pmax and maximum heat release rate are maintained nearly the same value up to λ= 1.5 shown in Fig.7.

CYCLIC DISPERSION

Peak cyclic pressures for 1074 consecutive working cycles[6),12),24)] were sampled for cyclic variation analysis. "Coefficient of Variation (COV)"[12),22)] of Pmax defined as follows(standard deviation of peak cyclic pressures devided by their mean value) was adopted for comparing cyclic variation in this study.

$$COV(P_x) = \frac{1}{\overline{P}_x}\sqrt{\sum^{n}\frac{(P_{xi}-\overline{P}_x)^2}{n}}$$

where

n : number of samples

Pxi : peak cyclic pressure at arbitrary cycle i.

$\overline{P}_x$: mean peak cyclic pressure over a sample of n cycles.

The same evaluation method as the above was used for analyzing cyclic variation of IMEP and (dP/dθ)max.

Cyclic Variations in IMEP, Pmax & (dP/dθ) max

Fig.8, Fig.9, and Fig.10 show cyclic variations in IMEP, Pmax and (dP/dθ)max as a function of spark plug type, spark discharge timing, and mixture strength respectively. As a function of spark plug type, no remarkable differences in cyclic dispersion characteristics in IMEP and Pmax were observed with stoichiometric mixture. But, with lean mixture, a general tendency as follows, which is not great but distinct, was observed; as the heat range of spark plug becomes hotter or the

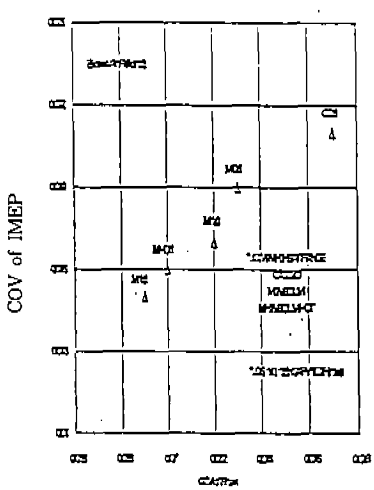
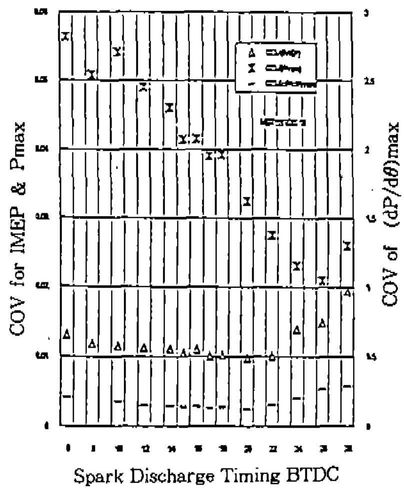
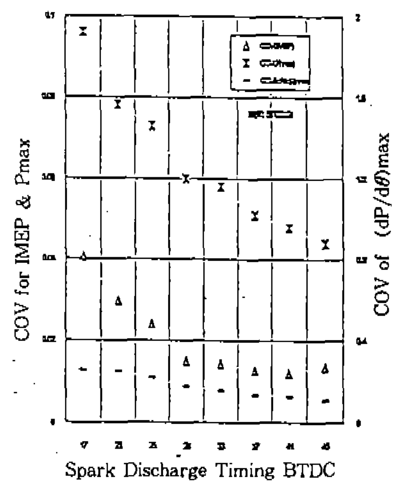


Fig.8 Cyclic variation of IMEP and Pmax as a functio of spark plug type



(a) with stoichiometric mixture



(b) with lean mixture(λ=1.53)

Fig.9 Cyclic variation of IMEP, Pmax, and (dP/dθ)max as a function of spark discharge timing

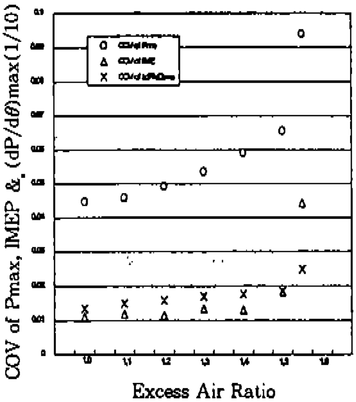


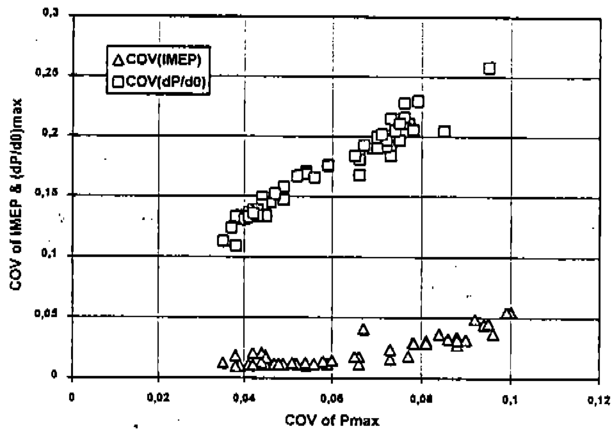
Fig.10 Cyclic variation of Pmax, IMEP & (dP/dθ)max as a functio of EAR

gap width grows bigger, cyclic variations (both of IMEP and Pmax) were decreased and relatively high correlation was observed between cyclic variation of IMEP and Pmax as seen in Fig.8. Fig.9(I), Fig.9(II) show cyclic variations in IMEP, Pmax and (dP/dθ) max with stoichiometric and lean mixture respectively as a function of spark discharge timing. As seen in Fig.9(I), with stoichiometric mixture, cyclic variation of IMEP shows the lowest value at around 20° BTDC and slight increase tendency with advancing spark discharge timing and sharp increase tendency with retarding spark discharge timing from this region[17]. Cyclic variation of Pmax gives distinct tendency of decrease with spark advance until 26°BTDC, but after this spark timing, it begins to increase[12]. Cyclic variation in (dP/dθ)max shows the lowest value at 20℃BTDC of spark discharge timing and a little moderated but nearly the same variation trends as that for IMEP. On the other hand, with lean mixture, nearly the same general tendency of cyclic variation in IMEP, Pmax was observed as seen in Fig.9 (II). It could be noted that cyclic variation in IMEP shows minimum value at around 41° BTDC and it grows both with advancing and retarding spark timing. But it shows sharper increase of that value with retarding spark timing reverse to stoichiometric mixture. Cylic variation in (dP/dθ)max is decreased steadily with advancing spark discharge timing.

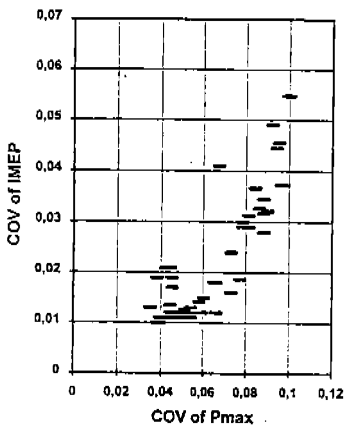
On the other hand, Fig.10 shows cyclic variation in IMEP, Pmax and (dP/dθ)max as a function of excess air ratio. With leaning mixture strength, all kinds of cyclic variations mentioned above show slight increasing tendency, but they present a little different characteristics each other;the increase of COV in Pmax was distinguishable but that for IMEP showed no remarkable increase(rather nearly constant) with all sorts of mixture strength until $\lambda(1.4)$, but around $\lambda(1.5)$, it begins to show considerable increase and between λ (1.5) to $\lambda(1.55)$, a sharp increase of these values were observed. It might suggest that λ (1.55) is situated not far from the lean misfire limit[22)23)]. and it could be easily expected that, as mixture strength is leaning over λ (1.55), the variation of Pmax and IMEP would show remarkable sharp increase until mixture strength is reached to lean misfire limit. COV of (dP/dθ)max shows very similar variation tendency to that for IMEP. It might be added that with the mixture strength of λ (1.55), the operation limit based on cyclic dispersion could be found.(COV in IMEP:around 0.045)[21)]

Relation of Cyclic Variation Characteristics in IMEP, Pmax and (dP/dθ)max

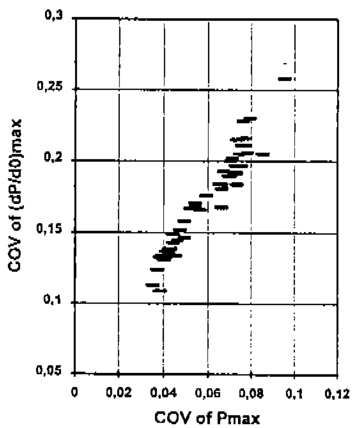
Fig.11 shows the relation of cyclic variations between IMEP, Pmax, and (dP/dθ)max for various excess air ratios, spark plug types, and spark discharge timings. Cyclic variations in Pmax and (dP/dθ)max appear to follow



(a) Relation between Pmax and IMEP & $(dP/d\theta)$max



(b) Relation between Pmax and IMEP



(c) Relation between Pmax and $(dP/d\theta)$max

Fig.11 Relation of cyclic variation characteristics on the basis of Pmax

generally the trends of the variations in IMEP values. On the other hand, COV in IMEP shows the least variation values among them, and Pmax shows about twice and $(dP/d\theta)$ max shows about six times the values of corresponding variations in IMEP respectively. Cyclic variation in $(dP/d\theta)$max might be thought to be situated in the middle position among three kinds of cyclic variations mentioned above, that is, it showed relatively a little more intimate correlation with IMEP or Pmax than that between Pmax and IMEP.

OPTIMUM COMBUSTION PHASING ANGLE

The crank angle for which 50% of cumulative heat has been released is defined as the combustion phasing angle and the combustion phasing angle at MBT spark timing is defined as the optimum combustion phasing angle. As seen in Fig.12, engine efficiency decreases both with advancing or retarding combustion phasing angle with regard to about 7°CA which is the optimum combustion phasing angle independently of mixture strength[24]. It

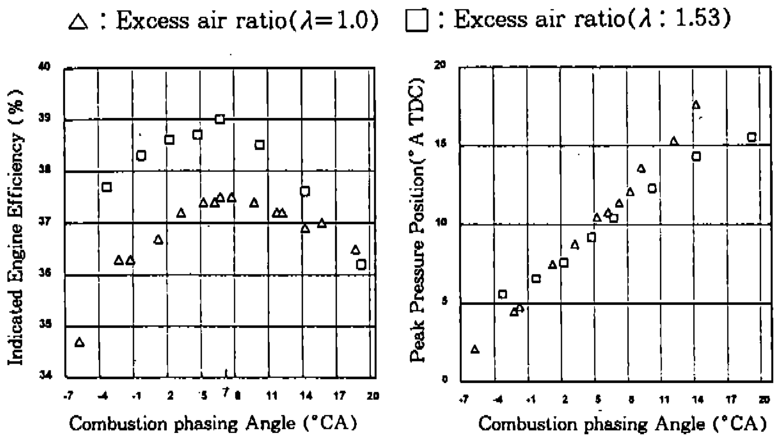


Fig.12 Indicated efficiency and peak pressure position versus combustion phasing angle

is necessary to have a sufficient spark advance for lean mixtures because of the greater influence of a retarded combustion phasing angle on engine efficiency for lean mixtures than that for stoichiometric mixture. We also see that the peak cylinder pressure position is advanced with advancing the combustion phasing angle.

CONCLUSION

Throughout the present experimental study, the following conclusion can be drawn.

1) Under normal operation conditions, the differences in combustion characteristics according to spark plug type were negligible with mixtures in the range of $\lambda=1.0$ and $\lambda=1.5$, but the coldest spark plug showed a little different characteristics especially with lean mixture($\lambda=1.5$).
2) With advancing spark discharge timing, a small but distinct increasing tendency for the ignition lag were observed with stoichiometric mixture, and with lean mixture, a clear increase tendency of ignition lag were observed and minimum values of main and total combustion durations at MBT were obtained.
3) The main combustion duration showed nearly no variance with mixtures from $\lambda=1.0$ to $\lambda=1.3$, but it begins to show a relatively rapid increase from $\lambda=1.4$ and an abrupt steep increase between $\lambda=1.5$ and $\lambda=1.6$. The ignition lag showed a similar variation trend.
4) With leaning out mixture strength between $\lambda=1.0$ and $\lambda=1.5$, the maximum heat release rate and the peak cylinder pressure showed the same variation tendency of steady decrease and the positions of them are maintained nearly the same value.
5) Some tendencies on cyclic variations as a function of spark plug type were observed especially with lean mixture;as the heat range of spark plug becomes hotter or the gap width grows bigger, cylic variation is decreased. And relatively high correlation was observed between cyclic variation of IMEP and Pmax.

6) With the variation of spark discharge timing, cyclic variations in IMEP, Pmax, and $(dP/d\theta)$max show increasing tendency both with advancing and retarding spark discharge timing from each optimum timing.
7) With leaning out mixture strength, cyclic variations in IMEP, Pmax, $(dP/d\theta)$max show slight increasing tendency, but they present a little different characteristics each other; the increase of cyclic variation in Pmax was distinguishable but that for IMEP and $(dP/d\theta)$max stayed rather nearly constant until $\lambda(1.5)$, but around $\lambda(1.55)$, a sharp increase of these calues were observed. With mixture strength $\lambda(1.55)$, limit operation condition based on cyclic variation characteristics could be found.
8) Cyclic variation in $(dP/d\theta)$max showed relatively a little more intimate correlation both with that for IMEP and Pmax than that between Pmax and IMEP.
9) The engine efficiency was decreased both with advancing and retarding the combustion phasing angle on the basis of about 7°CA which might be consequently thought as the optimum combustion phasing angle, and Pmax poistion was maintained nearly a constant value independent of mixture strength. These phenomena could be utilized as indicators for the estimation of MBT.

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