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TITLE:

Performance Tests of Reverse-Uniflow Type 2-stroke Direct Injection Gasoline Engine

Topic:

- ● FUTURE AUTOMOTIVE TECHNOLOGY
- ○ INTELLIGENT TRANSPORTATION SYSTEMS
- ○ USER FRIENDLY AUTOMOBILE
- ○ ADVANCED PRODUCTION AND LOGISTICS
- ○ VEHICLES & THE ENVIRONMENT

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National Society: ● YES ○ NO

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Abstract:

Conventional 2-stroke engines have defects such as unstable combustion, high fuel consumption rate, and high HC emissions. In order to overcome the defects, a direct fuel injection system and a new scavenging system were adopted. The authors tested a newly developed reverse-uniflow type 2-stroke direct injection gasoline engine, that was designed by numerical simulations. In comparison with the base engine, HC emission was decreased by up to 80%, and BSFC was reduced by around 40%. Power and BSFC were sperior to that of a latest 4-stroke engine. Furthermore, the effects of the start of injection timing and the fuel spray amount on the diffusion of fuel were examined by performance tests and numerical simulations, and the process was found important to improve the engine performance.

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INTRODUCTION

Conventional 2-stroke engines are commonly used for two-wheeled and marine vehicles that have small engine displacement. However problems such as unstable combustion, high fuel consumption rate, high HC emission, need to be resolved. 2-stroke engines have advantages over 4-stroke engines such as higher torque and power, being more compact and lighter. In order to solve these problems without compromising these advantages, a new 2-stroke engine was designed with numerical simulations to produce what is called a reverse-uniflow type 2-stroke direct injection gasoline engine.

In this study, performance of the reverse-uniflow engine was researched, and considerations concerning fuel consumption and exhaust gas emissions were made. These findings were then compared via simulations to consider gas flow and diffusion of fuel inside the cylinder.

EXPERIMENTAL CONDITIONS

Table 1 shows the specifications of the test engine. Conventional 2-stroke engines have shortcomings, such as under high engine speed conditions having high fuel consumption and higher HC emissions due to fuel short-circuiting to the exhaust port. Under low engine speed conditions, unsteady combustion is created due to a low scavenging efficiency. To resolve these issues, the reverse-uniflow scavenging system with direct fuel injection system was adopted.

Figure 1 shows a cross section of the test engine. The main characteristic of the test engine is having intake valves which are on the cylinder head rather than the cylinder wall, on the opposite side to the exhaust port.

Table 1 Engine specifications

Engine type	Two-stroke, single-cylinder, water cooled, direct injection, gasoline engine
Bore x Stroke	71.3 mm x 65 mm
Displacement	260 cm^3
Compression ratio	7.6 (effective)
Scavenging type	Reverse-Uniflow scavenging
Intake valve open - close	40 deg.BBDC – 80 deg.ABDC
Exhaust port open - close	70 deg.BBDC – 70 deg.ABDC

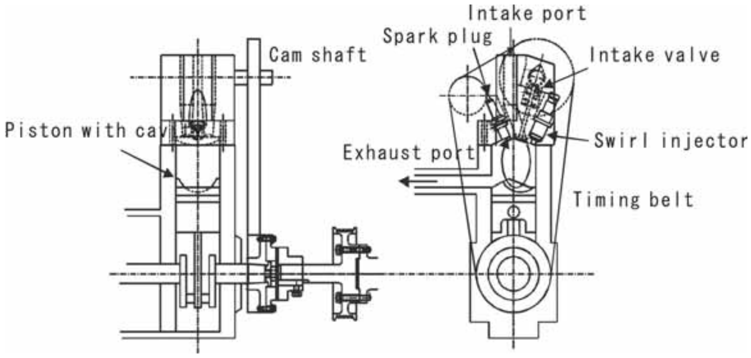

Fig.1 Schematic of reverse uniflow type engine

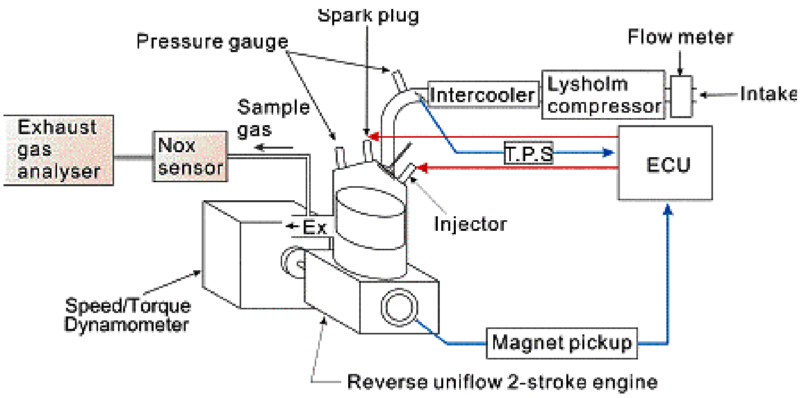

Fig.2 Experimental apparatus

As a result, fresh air flows in one single direction and a higher scavenging efficiency is obtained. Besides, that enables the best intake valve timing to be established. However, compression within the crankcase is not available, so this reverse-uniflow type engine must adopt an external compressor for scavenging.

Incorporating the cylinder gas flow like in a 4-stroke engine and a direct injection system, stratified charge combustion can be employed. Another benefit of the DI system is to inhibit the fuel short-circuiting due to injection after exhaust port closure.

This test engine is controlled with start of injection (SOI) timing, ignition timing and period of injection time. Figure 2 shows the experimental apparatus.

CHARGING EFFICIENCY

Two-stroke engines have a characteristic that is fresh air short-circuiting, so searching for the correct air fuel ratio is difficult. By comparing the gas exhaust emissions to the theoretical equation and the experimental results, the ideal air fuel ratio can be determined. Table 2 shows experimental conditions. Injection at 130 deg.BTDC is the earliest injection timing without fuel short-circuiting. In the calculations, the equation of theoretical equilibrium composition of combustion products is used. To solve the equation, it is assumed that the combustion chamber volume is constant. Comparing both results, an evaluation was made by using the “CO_2” concentration. Two approximated lines are used to represent the local maximal value. The comparison of the fuel amount and air fuel ratio at the local maximal value is used to determine the fresh air amount in cylinder.

Table 2 Experimental conditions

Engine speed [rpm]	1500, 2000
Boost pressure [kPa]	12.5, 25, 40
SOI timing [deg.BTDC]	130
Ignition timing [deg.BTDC]	MBT
Throttle opening ratio	WOT

Figures 3 and 4 are examples of result of exhaust gas emissions by theoretical equation in chemical equilibrium and experiment, respectively.

Relations between the delivery ratio or charging efficiency and boost pressure in different engine speeds are respectively reported in Fig. 5 and 6. The delivery ratio of the base engine is about 0.7, it was found that delivery ratio of test engine is much better than the base engine. Also, the charging efficiency of the reverse-uniflow type engine is sufficient for this low engine speed condition.

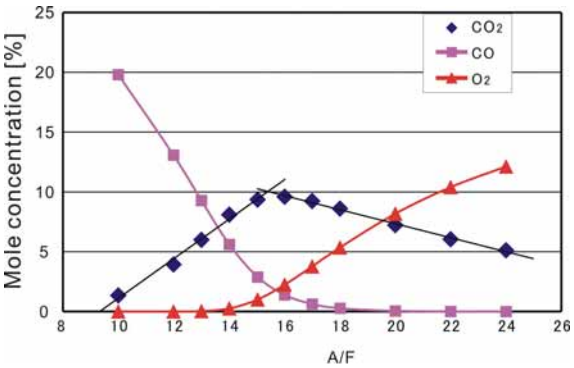

Fig.3 Exhaust gas emissions by theoretical calculation in equilibrium

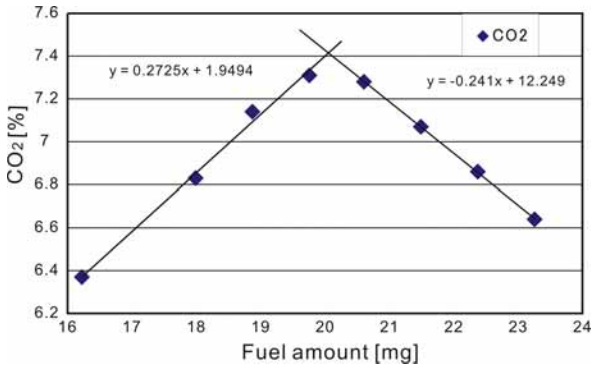

Fig.4 Exhaust gas emissions by engine test

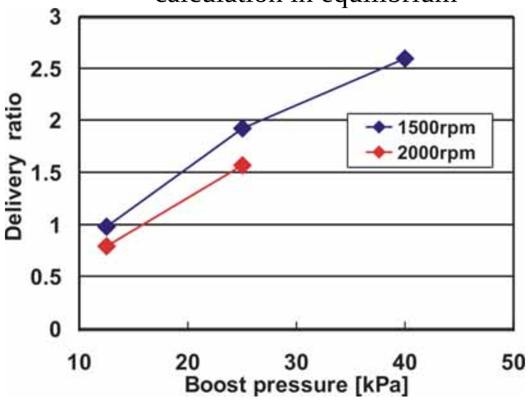

Fig.5 Delivery ratio vs. boost pressure

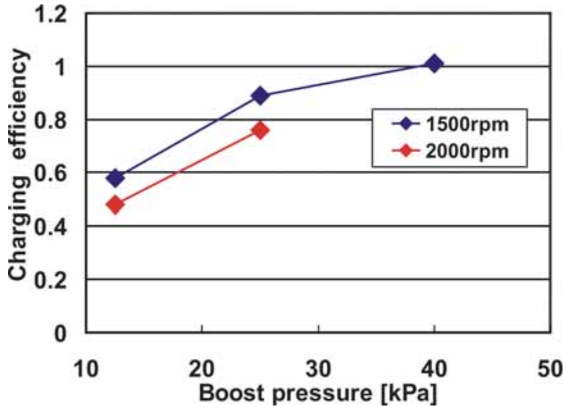

Fig.6 Charging efficiency vs. boost pressure

STRATIFIED CHARGE COMBUSTION

Prior to the stratified charge combustion, optimisation of injection timing and ignition timing is necessary in order to make a region of rich mixture in the vicinity of spark plug. In order to investigate the effects of injection timing and ignition timing, the injection timing was kept constant whilst the ignition timing was varied, and vice versa. These two conditions were investigated using different air fuel ratios. Table 3 shows the conditions for the stratified charge combustion.

Table 3 Stratified charge conditions for comparison

	Condition A	Condition B
Engine speed [rpm]	1500	1500
Boost pressure [kPa]	25	25
A/F	20	25
Throttle opening ratio	WOT	WOT

EXHAUST GAS EMISSIONS

1) Constant SOI timing

Figure 7 is an example of emissions in Condition A. Concentrations of CO and CO_2 were not affected by ignition timing. It was found that there was an increase in HC with delayed ignition timing. At the same time, NO_x increased with earlier ignition timing. Fig.8 is an example of emissions in Condition B. The aptitude of CO, HC and NO_x was similar to that of Condition A. However, CO_2 concentration is affected by ignition timing.
Moreover, Fig.9 shows the concentration of CO_2 at each SOI timing in Condition B. The curve showed a general increase when moving from left to right. Especially, when SOI

timing is 70, 80 deg.BTDC, the concentration of CO_2 decreased as ignition timing was delayed. This is considered to come from the difference in the diffusion of fuel.
Utilizing the simulation computed by Moriyoshi et al. [1] when test engine was designed, the fuel tend to diffuse quicker with earlier SOI timing . This phenomenon is caused by the effect of tumble flow in the cylinder on the diffusion of fuel. Additionally, diffusion of fuel is delayed with lower injected fuel amount. From these results, the region of rich mixture became narrower in Condition B, and optimization of ignition timing is complicated. Therefore ignition timing has a profound influence.
In another SOI timing, CO_2 concentration was kept at a constant value. It was found that injection at 70, 80 deg.BTDC is too late, i.e. fuel tended to blow over when fuel was injected. As a result, 60~70deg.CA was found necessary between injection timing and ignition timing due to use stratified charge combustion.

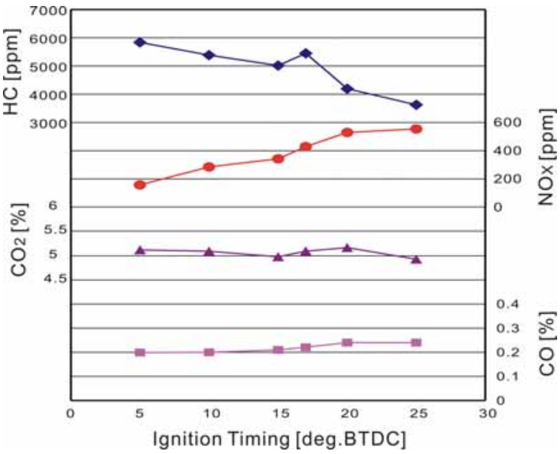

Fig.7 Exhaust gas emissions
vs. ignition timing
(Condition A, SOI at 80deg.BTDC)

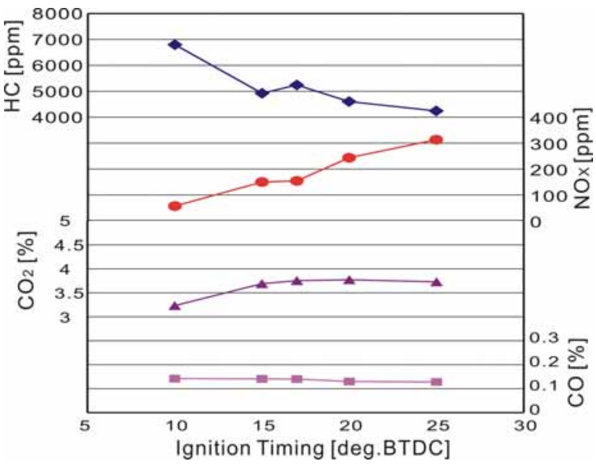

Fig.8 Exhaust gas emissions
vs. ignition timing
(Condition B, SOI at 80.BTDC)

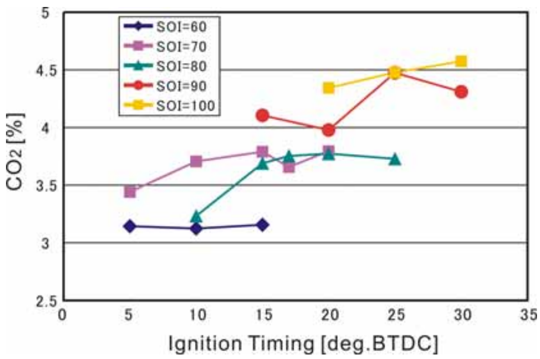

Fig.9 Exhaust gas emissions
vs. ignition timing with varying SOI timing
(Condition B)

2) Constant ignition timing

In Condition A, tendency of emissions in each ignition timing tended to be similar, so an example of emissions is showed in Fig.10. The ignition timing is 20 deg.BTDC. The concentrations of CO, CO_2 and NO_x increased with earlier SOI timing. At the same time, HC concentration decreased. In Condition B, consistency in emissions was not found.
However, the local maximal value was determined when ignition timing was set at 10 deg.BTDC. Figure 11 shows the emissions with ignition timing at 10 deg.BTDC. It is

considered that ignition timing has a profound influence due to diffusion of fuel being restrained by a decreased fuel amount. In addition, since a delay in ignition timing produces a delay in injection timing, fuel diffusion was controlled to produce a suitable injection timing. Consequently, apparent the local maximal value was found.

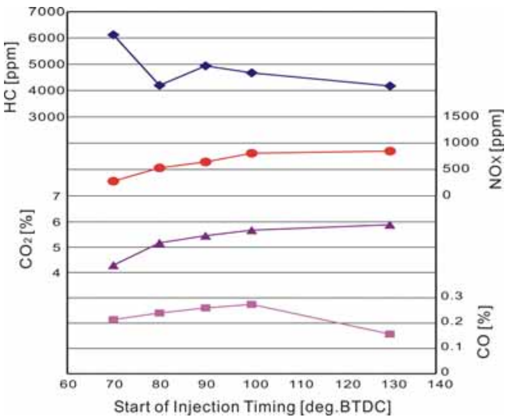

Fig.10 Exhaust gas emissions vs. SOI timing (Condition A, Ignition timing at 20deg.BTDC)

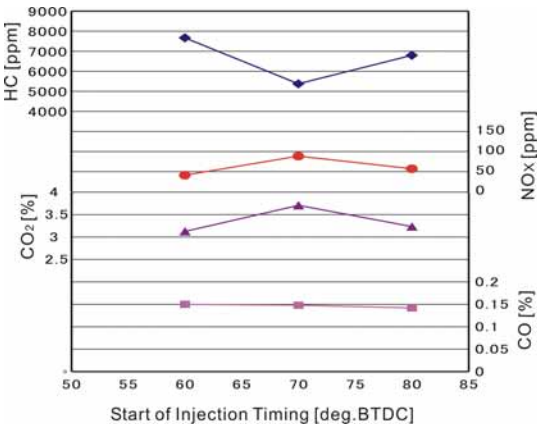

Fig.11 Exhaust gas emissions vs. SOI timing (Condition B, Ignition timing at 10deg.BTDC)

THERMAL EFFICIENCY

In the same way as the above, the effects of injection timing and ignition timing on thermal efficiency were investigated. Figure 12 shows the results for Condition A, where the SOI timing was kept constant. It was found that higher thermal efficiency was obtained with earlier SOI timing at each ignition timing. It shows that enhanced combustion was attained with appropriate fuel diffusion without leaning out.
The results when ignition timing was kept constant in Condition A are showed in Fig.13. The curve at each ignition timing was nearly consistent, also the thermal efficiency was not much affected by ignition timing because fuel diffused sufficiently. However, thermal efficiency of the earliest and latest ignition timing at each injection timing interval was depreciated. Furthermore, fluctuations in thermal efficiency increased with delayed ignition timing. This was consistent with the computed simulation.

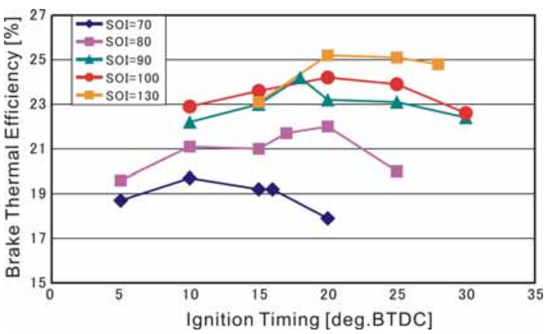

Fig.12 Effect of ignition timing on brake thermal efficiency with varying SOI timing (Condition A)

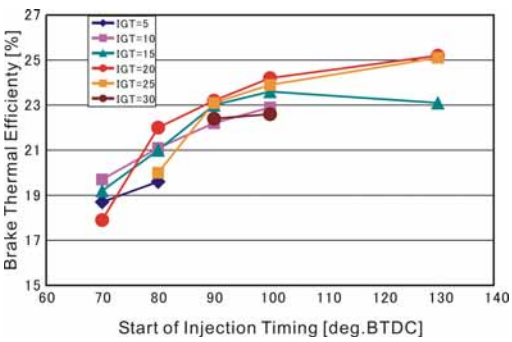

Fig.13 Effect of SOI timing on brake thermal efficiency with varying SOI timing (Condition A)

In Condition B, Fig.14 shows the result with SOI timing was kept at a constant. It is found that thermal efficiency increased with earlier ignition timing, as shown in Condition A. However, whilst it slightly decreased, the ignition timing varied from 90 deg.BTDC to 100 deg.BTDC.

It is considered that the fuel is diffused with earlier SOI timing, and its concentration distribution is evenly configured. Thus, the thermal efficiency decreased due to nearly a homogeneous charge combustion which cause slow and unstable combustion at air fuel ratio of 25.

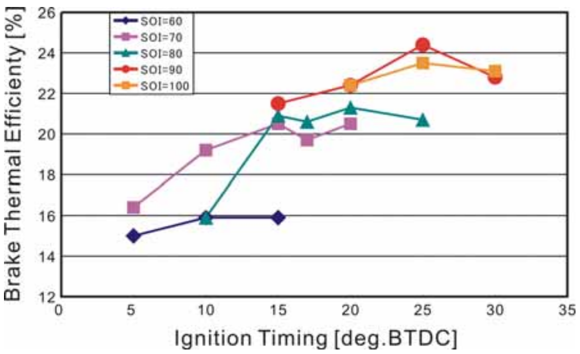

Fig.14 Effect of ignition timing on brake thermal efficiency with varying SOI timing (Condition B)

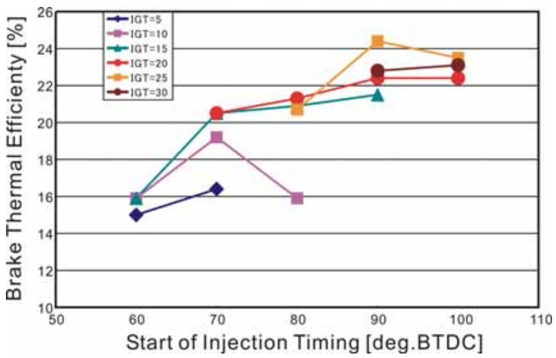

Fig.15 Effect of SOI Timing on brake thermal efficiency with varying SOI timing (Condition B)

The results are shown in Fig.15 where the ignition timing was kept constant in Condition B. The tendency is different from that in Condition A. The effect of ignition timing on the thermal efficiency can be found in each SOI timing. This came from the decreased diffusion of fuel with lower fuel amount. In comparison to Fig.13, the thermal efficiency increase is little when ignition timing was varied from 70 deg.BTDC to 100 deg.BTDC due to slow and unstable combustion.

PERFORMANCE TESTS

Performance tests of the reverse-uniflow type DI engine were conducted by varying three parameters; air fuel ratio, boost pressure and engine speed. Engine speed was limited to 1000~2000 rpm due to the confines of this experiment.

COMPARISON OF HOMOGENEOUS CHARGE COMBUSTION AND STRATIFIED CHARGE COMBUSTION

Figure 16 shows the comparison of homogeneous charge combustion with stratified charge lean burn of each engine speed. Boost pressure was kept at 25 kPa.

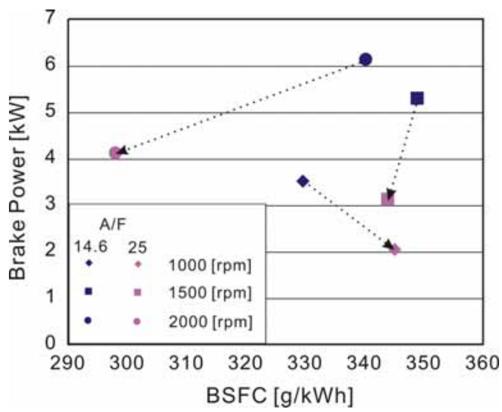

Fig.16 Effect of combustion mode on brake power and BSFC in different engine speeds

It is found that the BSFC improved except for the conditions with engine speed at 1000 rpm. That is to say there is a compromise to be made between increased power or BSFC by switching the combustion mode. It is considered that gas flow inside the cylinder was not sufficient with low engine speed, i.e.1000 rpm.

OPTIMIZASION OF AIR FUEL RATIO

Figure 17 shows the relation between power and BSFC when the engine speed was kept constant whilst the air fuel ratio was varied. It was found that the BSFC had the local minimal value with varied air fuel ratio and engine speed. The minimal value is attained at different air fuel ratio in each engine speed.

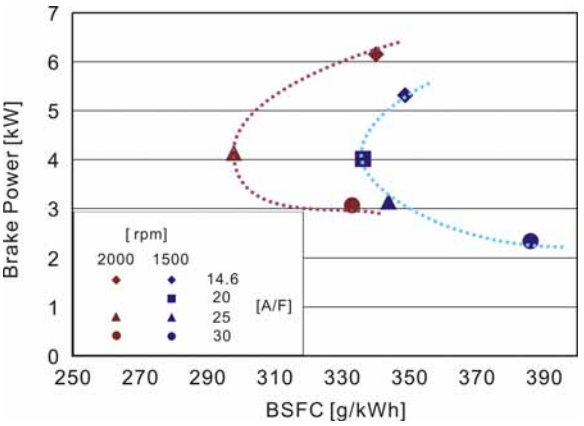

Fig.17 Relation between brake power and BSFC in different engine speed and A/F

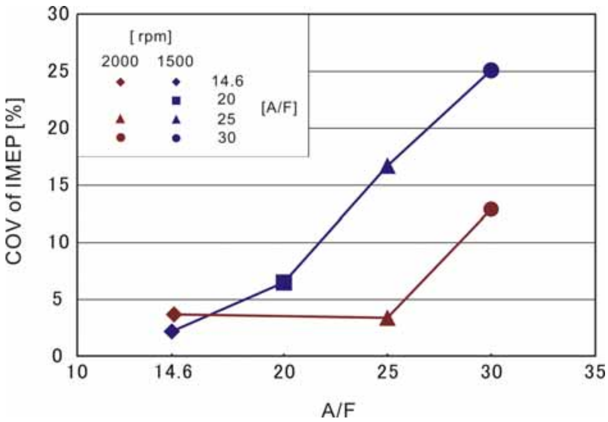

Fig.18 Relation between COV of IMEP and A/F in different engine speed

OUTPUT PERFORMANCE, EXHAUST EMISSION AND FUEL CONSUMPTION

Figure 19 shows the output performance of a test engine. The composition of a base engine is presented in the graph as a dotted line, and the latest 4-stroke engine is detailed in the graph as a solid line. The brake power of the test engine when the boost pressure is 12.5 kPa has been found to be better than the base engine, and it is equivalent to that of the latest 4-stroke engine. With boost pressure increased to 25 kPa, it exceeds the latest 4-stroke engine.

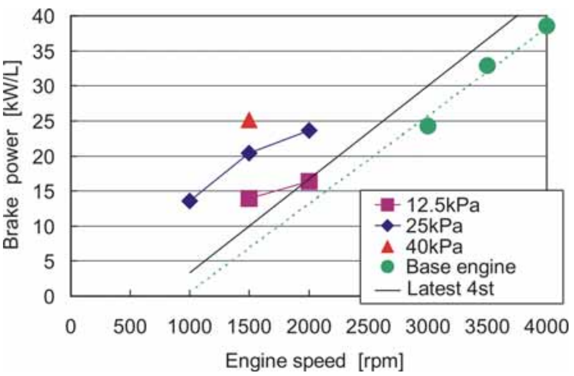

Fig.19 Characteristic of Brake Power

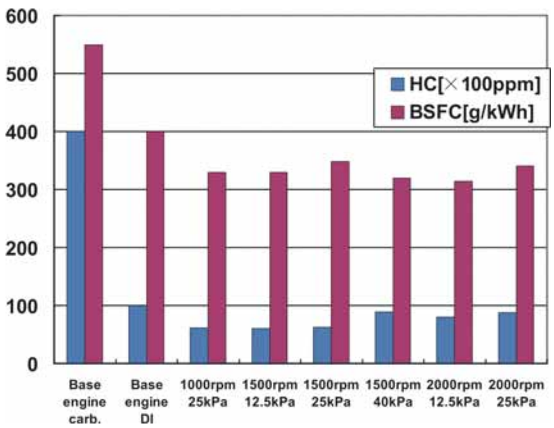

Fig.20 THC and BSFC

Figure 20 shows the THC concentration and BSFC performance. This is compared with the base engine (base engine carb) and base engine converted to a direct fuel injection (base engine DI) in the full load condition. In each test engine condition, concentration of THC was decreased by up to 80%, and fuel consumption was reduced by around 40%. It was found that the BFSC of test engine exceeded that of the DI base engine. Also, the BSFC was fond equivalent to the latest 4-stroke engine. In lower load conditions, the test engine shows superior BSFC to the latest 4-stroke engine.

CONCLUSIONS

1. By combining a new scavenging method and a direct injection system, a higher scavenging efficiency was obtained, and concentration of THC was vastly decreased.

2. Unstable combustion was improved with the stratified charge combustion, and then NO_x concentration decreased because gas temperature inside cylinder fell due to lower air fuel ratio.

3. By comparing with numerical simulations, it was found that diffusion of fuel affected the stability of stratified charge combustion.

4. Output power, exhaust gas emissions and fuel consumption rate of the reverse-uniflow type DI 2-stroke engine exceeded those of the base engine and a latest 4-stroke engine.

References

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(2) Koji Morikawa and Hideharu Takimoto, Taiichiroh Ogi, "A Study of Direct Fuel Injection Two-Stroke Engine for High Specific Power Output and High Engine Speed", SAE Paper, 1999-01-3288