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ABSTRACT

A new V-6 3.5-liter gasoline engine (2GR-FSE) uses a newly developed stoichiometric direct injection system with two fuel injectors in each cylinder (D-4S: Direct injection 4-stroke gasoline engine system Superior version). One is a direct injection injector generating a dual-fan-shaped spray with wide dispersion, while the other is a port injector. With this system, the engine achieves a power level among the highest for production engines of this displacement and a fuel economy rating of 24mpg on the EPA cycle. Emissions are among the lowest level for this class of sedans, meeting Ultra Low Emission Vehicle standards (ULEV-II).

The dual-fan-shaped spray was adopted to improve full-load performance. The new spray promotes a homogeneous mixture without any devices to generate intense in-cylinder air-motion at lower engine speeds. For this reason the engine has improved volumetric efficiency compared to engines having these charge motion devices, resulting in improved full-load performance throughout the engine speed range. Together with Dual VVT-i (Variable intake and exhaust Valve Timing intelligence), the engine achieves specific power near the top of all naturally aspirated production gasoline engines in the world: 66kW/L, 228kW at 6400r/min.

Fuel economy is improved compared to a conventional DISI engine with both injectors optimized to improve combustion. As for improvement of the exhaust emissions, simultaneous injection by the two injectors is effective in reduction of HC emissions during cold start.

1. INTRODUCTION

The requirements of reduced emissions and enhanced driving comfort are critical for modern automobiles. Several automobile manufacturers have introduced direct-injection spark-ignition (DISI) engines to the worldwide markets. Initially, DISI engines utilized stratified operations to improve fuel consumption. As the emission regulations have become more stringent, DISI engines have changed to utilize stoichiometric operations to meet emission regulations. The reason is that stoichiometric DISI engines have several advantages over port fuel injection (PFI) engines. These advantages of DISI engines are higher full-load performance, reduced emissions such as THC, NOx and CO emissions and better fuel economy i.e. CO_2 emissions [1-3].

Concerning full-load performance, it was reported that the fuel latent heat can be utilized to reduce the intake charge temperature. DISI engines exhibit improved full-load performance by injected fuel evaporation that uses thermal energy inside the cylinder; this results in DISI engines demonstrating higher volumetric efficiency (ηv) and lower knocking tendencies [4-8]. Several studies investigating the benefit of volumetric efficiency that a DISI engine gains through fuel latent heat have been conducted. They found that volumetric efficiency of a DISI engine is higher by approximately 2-3% compared to an equivalent port fuel injection (PFI) engine [4-8]. They report that through charge cooling by fuel evaporation a DISI engine can obtain a higher compression ratio by approximately 1-2 points [8]. Higher volumetric efficiency and lower knocking tendencies bring a DISI engine higher torque and power output. It was reported that torque can increase across a wide speed range by approximately 5-10% [7].

As for emissions performance, a DISI engine can utilize stratified charge combustion and an intense turbulence generated by the spray injected at the end of the compression stroke during the catalyst warm-up phase; this enables stable combustion at highly retarded ignition timing, which significantly improves the warm-up speed of the catalyst. It was observed that the increase of the catalyst temperature for a DISI engine compared to an equivalent PFI engine is approximately 500 °C [1], that results in a decrease in THC emissions.

For a stoichiometric DISI engine, suppressing knock improves fuel economy. The DISI engine achieves better fuel consumption than an equivalent PFI engine due to reduced knocking tendency that enables higher compression ratios of approximately a 1-2 point increase [8].

A DISI engine, however, has a disadvantage in forming a homogeneous air-fuel mixture in the cylinder because of a lack of time to form a homogeneous mixture from the time fuel is injected until ignition occurs. This causes mixture stratification in the cylinder. The combustion efficiency and the combustion variability of a DISI engine, consequently, are worse at higher loads and at lower engine speeds due to weak in-cylinder air-motion. In order to improve these combustion deteriorations, a DISI engine requires some devices to generate in-cylinder charge-motion to promote a homogeneous mixture formation. These devices include a tumble intake-port, a helical intake-port, a swirl control valve (SCV) and so on. These devices, however, decrease the intake-port flow efficiency compared to that of a PFI engine. The intake-port design requires higher volumetric efficiency and also more intense air-motion, but these are conflicting requirements. Figure 1 shows the trade-off between flow coefficient and tumble intensity. It was reported too in other studies that the creation of stronger in-cylinder flows typically reduces volumetric efficiency [7]. DISI engines, consequently, have a disadvantage with respect to full-load performance, because they require intense in-cylinder air-motion to form a homogeneous mixture resulting in reduced intake-port efficiency and torque and power loses at higher engine speeds.

The main purpose of this study is to improve full-load performance with a high flow efficiency intake-port and adopting a new dual-fan-shaped spray. This spray helps to promote a homogeneous mixture at lower engine speeds without any devices to generate intense air-motion to improve combustion stability. With respect to fuel consumption and torque fluctuations at part loads, it will be described that the use of a PFI injector together with a DISI injector produces a more homogeneous mixture improving combustion efficiency, reducing fuel consumption and torque fluctuations compared to a PFI engine. Additionally, simultaneous injection is effective in reduction of HC emissions during cold starts before the start catalyst is activated. Lastly the specifications for the 2GR-FSE engine adopting this new direct injection system with dual-fan-shaped spray and installation of port fuel injection will be reported.

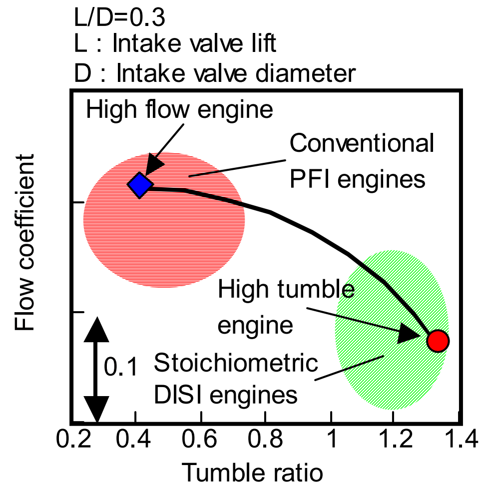


Fig.1 Tumble intensity vs.flow coefficient

2. THE MAIN ISSUES OF COMBUSTION OF THE HIGH FLOW EFFICIENCY ENGINE

2.1 ENGINE DETAILS OF THIS STUDY

The test engine for this study is a V-6 3.5-liter engine having the major specifications as given in Table 1. This new engine is a stoichiometric DISI gasoline engine with intake and exhaust variable valve timing system (Dual VVT-i), and is a larger displacement version of the 3GR-FSE engine: a V-6 3.0-liter engine introduced to the North American markets in 2005. The combustion chamber of this engine is a pentroof type with a tapered squish. The direct injection spray is a single-fan-shaped spray pattern with wide dispersion and moderate penetration, the same as the 3GR-FSE.

In order to compare effects on full-load performance between a high flow efficiency and a high tumble intake-port, the swirl control valve (SCV) was removed and a high flow efficiency intake-port, improving air-flow by approximately 20% from the high tumble engine was adopted. Consequently tumble ratio of the high flow test engine was reduced by one-third at the valve-lift divided by the valve-diameter (L/D) of 0.3 (shown in Fig. 1).

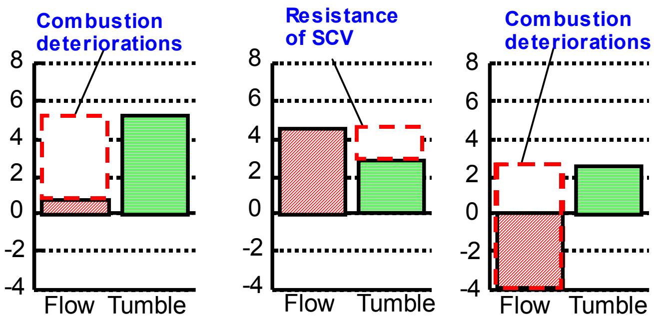
Table 1 Test Engine Specifications

Type	6 Cylinder 60° V Angle
Bore	94.0mm
Stroke	83.0mm
Displacement	3456cc
Compression Ratio	11.8 : 1
Valve Train	Intake & Exhaust VVT-i

2.2 STUDY OF FULL-LOAD PERFORMANCE WITH THE HIGH FLOW EFFICIENCY INTAKE-PORT

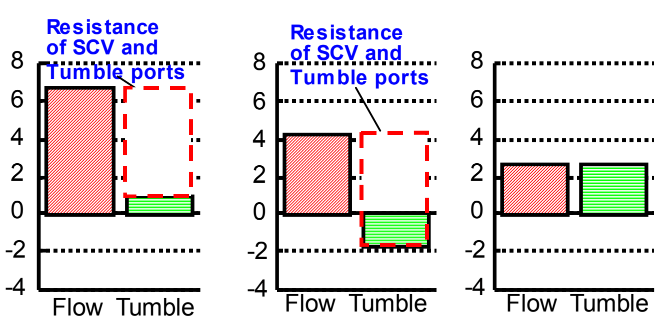
Comparison of full-load performance for the high flow engine with that of the high tumble engine was conducted to quantify effects on combustion by weaker air-motion for the high flow engine and effects of reduced air-flow for the high tumble engine. Figure 2 and 3 show comparisons of full-load torque between the high flow engine and the high tumble engine. The test condition of Figure 2 is full-load at 2800r/min, and the condition of Figure 3 is full-load at 6400r/min.

The full-load torque of the high flow engine drops by over 4% from the high tumble engine at 2800r/min although the volumetric efficiency (ηv) of the high flow engine is a little higher than that of the high tumble engine because of flow losses due to the SCV. The reason why the high flow engine drops torque is because the combustion efficiency (Pme/ηv, Pme: brake mean effective pressure) deteriorates due to the heterogeneous mixture formation in the cylinder. In this study, Pme/ηv can represent combustion efficiency because friction losses of the high flow engine are equal to those of the high tumble engine. On the other hand,



(a) Improvement of brake torque from PFI (%) (b) Improvement of hv from PFI (%) (c) Improvement of Pme/hv from PFI (%)

Fig. 2 Improvement of brake torque from a PFI engine at 2800r/min



(a) Improvement of brake torque from PFI (%) (b) Improvement of hv from PFI (%) (c) Improvement of Pme/hv from PFI (%)

Fig. 3 Improvement of brake torque from a PFI engine at 6400r/min

the output of the high flow engine at 6400r/min improves from that of the high tumble engine because of higher air flow as shown by the volumetric efficiency. Combustion efficiency of the high flow engine is the same as that of the high tumble engine because the in-cylinder air-motion becomes sufficiently intense to make a homogeneous air-fuel mixture as the engine speeds increases. To improve engine output, removing the tumble intake-ports and SCV and improving the combustion efficiency at lower speeds without any devices to generate intense in-cylinder air-motion must be undertaken.

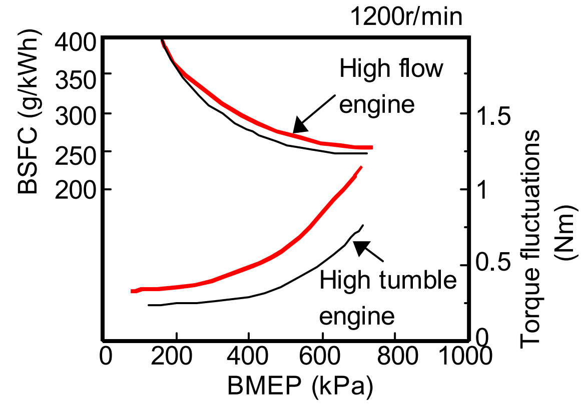


Fig. 4 Comparison of fuel consumption and torque fluctuations between the high flow engine and the high tumble engine for part loads at 1200r/min

2.3 STUDY OF PART-LOAD COMBUSTION WITH THE HIGH FLOW EFFICIENCY INTAKE-PORT

The study of torque fluctuations and fuel consumption for the high flow engine was also conducted at lower engine speeds (1200r/min). Figure 4 shows a comparison of torque fluctuations and fuel consumption between the high flow engine and the high tumble engine. The high flow engine cannot satisfy the drivability limit for torque fluctuations, and fuel consumption of the high flow engine deteriorates compared to the high tumble engine. The mixture for the high flow engine is thought to be heterogeneous due to the weak in-cylinder air-motion causing the combustion efficiency to deteriorate. Combustion must be improved and the torque fluctuations must be reduced under the drivability limit in order to adopt the high efficiency intake-ports to increase the full-load torque at higher engine speeds.

2.4 SOLVING COMBUSTION ISSUES FOR THE HIGH FLOW EFFICIENCY INTAKE-PORT

The issues at full loads and part loads must be solved in order to adopt the high flow efficiency intake-ports and improve the full-load performance at higher engine speeds. The issue at full loads is combustion deterioration at lower engine speeds. The deterioration is confirmed to be due to a heterogeneous mixture formation in the cylinder at lower engine speeds. This condition is due to removal of devices to make intense in-cylinder air-motion in exchange for improved volumetric efficiency. Efforts to improve in-cylinder air-motion typically reduce volumetric efficiency resulting in a loss of full-load performance. Besides full-load performance can be improved by the DI injection due to the injected fuel evaporation. For these reasons, development of the DI spray pattern should be undertaken at full loads in order to improve the full-load combustion efficiency.

Heterogeneous mixture at lower engine speeds under part-load conditions is the same as the issue at full loads. Solutions to address combustion variability to improve homogeneous mixture without the use of intense air-motion devices are limited by trade-offs. Because the amount of fuel and mixture may vary in size, development of only the DI spray pattern cannot sufficiently form a homogeneous mixture in the cylinder under all engine conditions. The only solution that can be chosen is to premix the mixture before induction into the cylinder. To put it other words, a PFI injector is installed with a DI injector in each intake-port and simultaneous injection of a PFI injector and a DI injector is utilized.

2.5 DETAILS OF THE NEWLY DEVELOPED FUEL INJECTION SYSTEM : D-4S SYSTEM

To solve the combustion issues, the new fuel injection system was adopted that has both a PFI injector and a DI injector with a newly developed spray and a high flow efficiency intake-port. Figure 5 shows the detail of D-4S system. High flow efficiency intake-ports were adopted to increase air-flow at high engine speeds. And a PFI injector was installed in the high flow efficiency intake-ports. The DI spray is newly developed to improve combustion during full-load performance at lower engine speeds. The DI injector has a dual-fan-shaped spray injected vertically into the cylinder with wide dispersion, shown in Table 2. Table 2 shows the specifications of both DI injectors and PFI injectors. Figure 6 shows the newly developed spray. The spray is injected perpendicularly to the piston with wide dispersion in the cylinder. Figure 7 schematically shows the spray pattern in the cylinder. This figure demonstrates that this spray is injected vertically in the cylinder and spreads toward the exhaust side of the combusiton chamber. For this reason, this spray has a wide dispersion ability in the cylinder.

This paper will describe that the newly developed fuel injection system improves combustion at full loads without devices to generate intense in-cylinder air-motion. Additionally, D-4S system is effective in promoting a more homogeneous mixture than that of a PFI engine without intense in-cylinder air-motion at part loads.

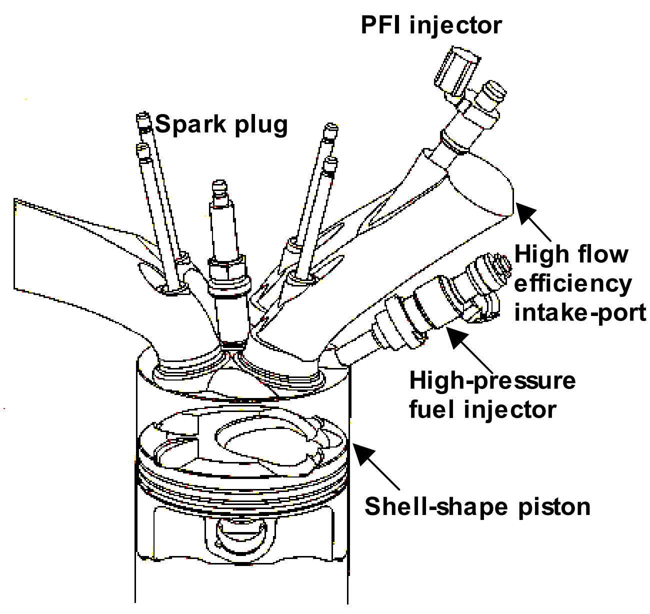


Fig. 5 A perspective view of the combustion chamber geometry of D-4S system

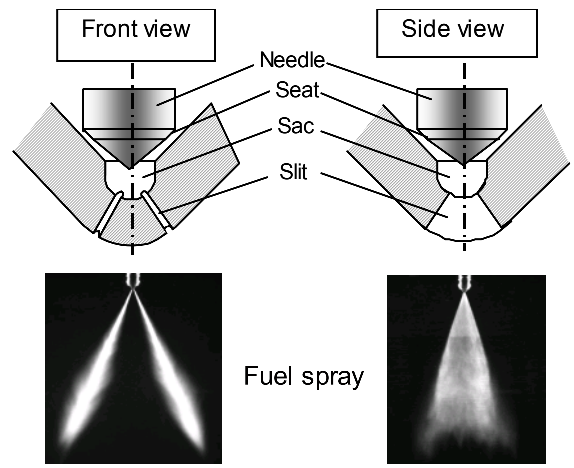


Fig. 6 Dual-fan-shaped spray shape

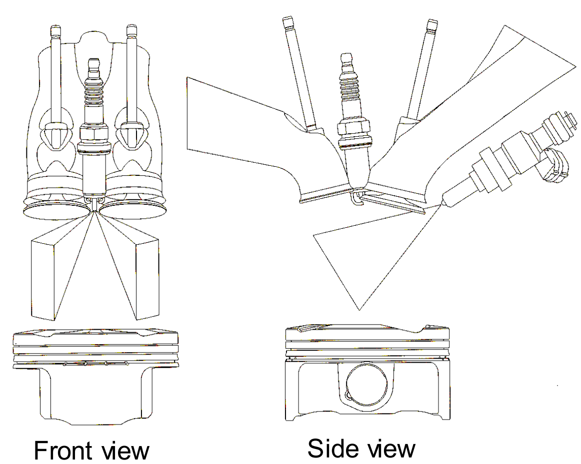


Fig. 7 Schematic view of in-cylinder pattern of the newly developed spray

Table 2 Specifications for the injectors and fuel system

Specification of injectors for DI	
Spray shape	Dual-fan-shaped spray
Hole shape	Two slit (W:0.52mm,H:0.13mm)
Pressure	4-13MPa
Injection rate	948cc/min @12MPa
Specification of injectors for PFI	
Spray shape	12 Multiholes
Hole shape	Multiholes (φ0.19*12)
Pressure	400kPa
Injection rate	295cc/min

3. IMPROVEMENT OF FULL-LOAD PERFORMANCE WITH THE DEVELOPED SPRAY

3.1 EFFECT OF THE NEWLY DEVELOPED SPRAY UNDER THE FULL-LOAD CONDITION

CFD analysis of the mixture distribution in the cylinder was conducted in order to reveal the effect of the newly developed spray. Star-CD was used as a solver and mesh generator for CFD analysis. Approximately 470,000 cells were made for these analyses. The calculated model is shown in Fig. 8, and the calculated conditions are shown in Table 3.

Table 3 CFD conditions of this study

Solver	Star-CD v3.150A
Differencing scheme	MARS
Turbulence model	Standard k-ε model
Spray model	DDM (Discrete Droplet Model)
Spray impinging model	Senda model ('99 COMODIA)
Number of cells	approximately 470,000

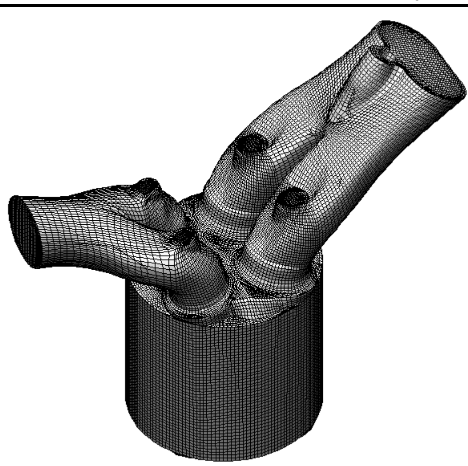


Fig. 8 CFD model of test engine

The analysis was conducted at full-load 2800r/min. Figure 9 shows the spray pattern injected into the cylinder. The dual-fan-shaped spray is injected perpendicularly to the piston and with wide dispersion in the cylinder. For this reason the spray is injected without being trapped by the piston cavity and the spray can be injected toward the exhaust side of the combustion chamber. On the other hand the conventional spray is trapped by the piston cavity and cannot spread toward the exhaust side of the combustion chamber. As a result the newly developed spray can make a homogeneous mixture in the cylinder as shown in Figure 10. This spray can spread in the cylinder without intense in-cylinder air-motion. The conventional spray, alternatively, makes a less homogeneous mixture especially on the exhaust side of the combustion chamber. This lean region is because the spray itself does not have the ability to spread inside the cylinder.

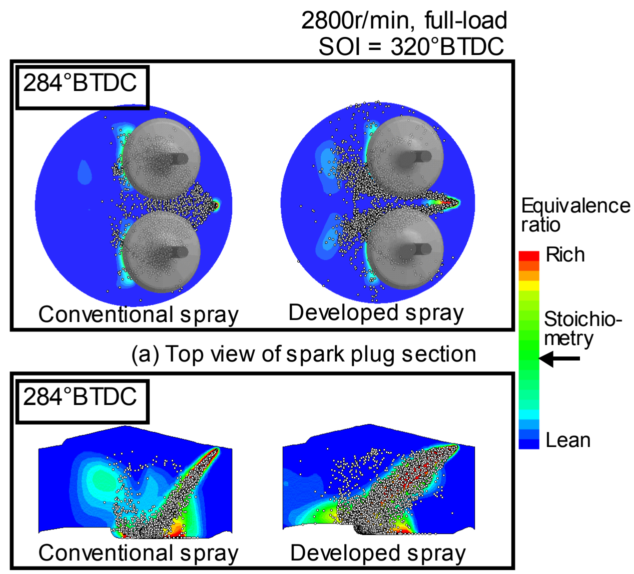


Fig. 9 Comparison of the spray pattern calculated by CFD

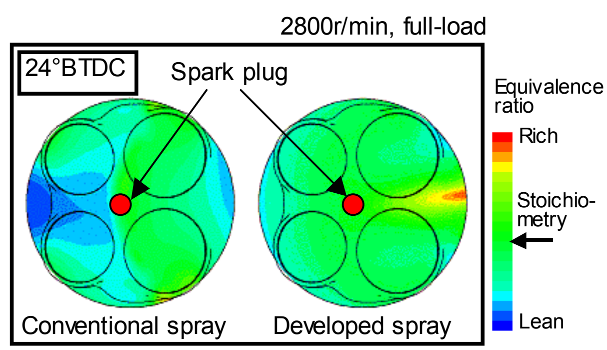


Fig. 10 Comparison of the mixture formation of the developed spray at the cross section of the spark plug

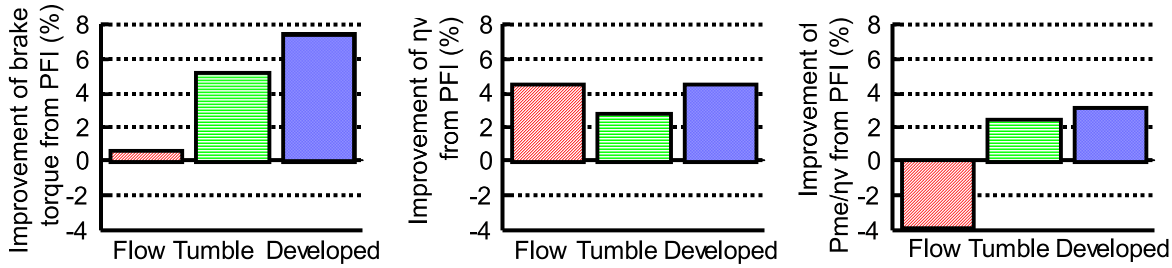


Fig. 11 Improvement of brake torque from a PFI engine at 2800r/min

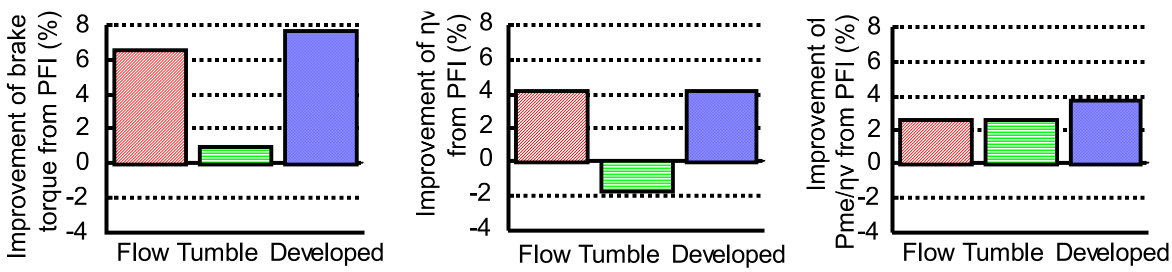


Fig. 12 Improvement of brake torque from a PFI engine at 6400r/min

3.2 COMBUSTION IMPROVEMENT AND FULL-LOAD PERFORMANCE OF THE DEVELOPED SPRAY

Full-load performance results for the newly developed spray are shown in Figure 11 and 12. Torque at lower engine speeds is improved due to improved combustion efficiency, allowing the developed spray engine to take advantage of its superior volumetric efficiency. Figure 13 shows the improvement of the conventional DI engine and the developed DI spray engine compared to a PFI engine. The improvement from a PFI engine is 6-8% through the engine speeds range. A conventional DISI engine, a high tumble engine, can improve torque at lower engine speeds, but cannot sustain this advantage at higher engine speeds due to the resistance from the air-motion device. With the developed spray, this test engine can improve at all engine speeds even though tumble is weaker without a SCV.

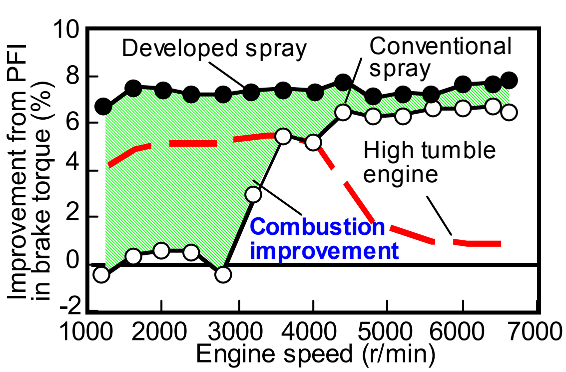


Fig. 13 Improvement in torque from a PFI engine

4. IMPROVEMENT OF FUEL CONSUMPTION AND COMBUSTION VARIABILITIES WITH D-4S SYSTEM

4.1 IMPROVEMENT STRATEGY FOR MIXTURE FORMATION AT PART LOADS

Mixture formation in the cylinder needs to approach the homogeneity of the PFI engine in order to improve fuel consumption and reduce torque fluctuations over the PFI engine. Two techniques can be utilized to produce a homogeneous mixture. One method is to promote in-cylinder mixing with intense air-motion, and the other is induction of a premixed homogeneous mixture into the cylinder. The former, of course, is typical of most DISI engines, and the latter is description of PFI. As mentioned above, the former technique decreases full-load torque due to air-flow restriction. This method should be abandoned to achieve the best possible full-load performance. For this reason use of a PFI injection system was adopted to produce a homogeneous mixture and still deliver the desired level of full-load performance. Based on these findings a unique fuel injection system was devised incorporating both a PFI injector and a DI injector. The strategy for this system allows each injector to operate simultaneously or independently based on the speed/load conditions of the engine. The PFI injector is located to inject fuel without increasing port wetting. Additionally, an intake-port was designed to accommodate both the PFI and DI injectors without compromising air-flow.

4.2 COMBUSTION EFFECTS AT PART LOADS OF SIMULTANEOUS INJECTION

Fuel consumption and torque fluctuations at 1200r/min, BMEP 650kPa were studied by varying the amount of fuel injected by the DI injector as a percentage of total injection fuel at part loads. The results show in Figure 14 that simultaneous injection by-both injectors has better fuel consumption and reduces torque fluctuations compared to using a DI injector or PFI exclusively. Fuel consumption is improved by approximately 7% at a 30% DI injection ratio. Torque fluctuations at 30% DI ratio are lower than PFI only. Moreover the burn rate at 30% DI injection ratio is faster than PFI only. Although the PFI injection has a homogeneous mixture, the PFI injection has a slower burn rate compared to simultaneous injection. The reason why the burn rate of simultaneous injection is faster than that of PFI injection is because simultaneous injection has two injectors, providing improved fuel dispersion. For this reason, simultaneous injection produces a more homogeneous mixture than PFI injection, resulting in simultaneous injection having faster combustion speed.

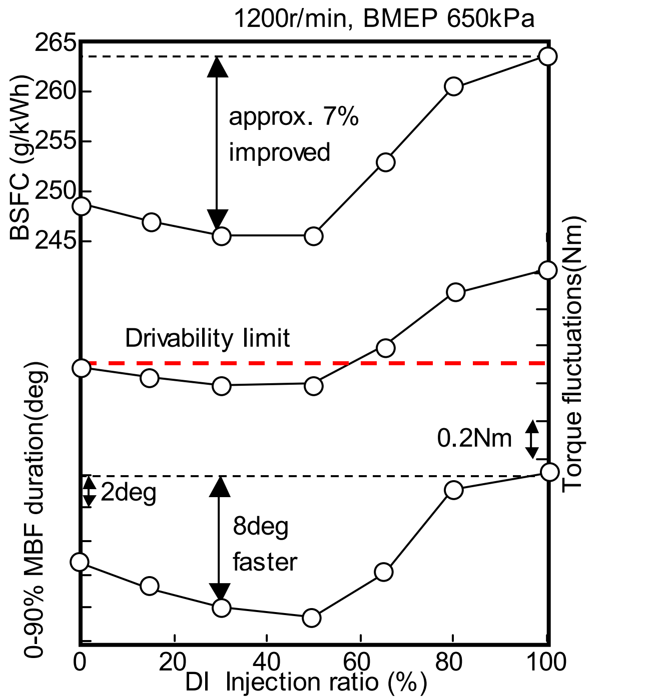


Fig.14 Improvement of combustion with simultaneous injection

4.3 IMPROVEMENT OF FUEL CONSUMPTION AND TORQUE FLUCTUATIONS WITH SIMULTANEOUS INJECTION

A study of fuel consumption and torque fluctuations at part loads was completed to confirm the effects of simultaneous injection. For this evaluation, DI injection ratio was adjusted for best torque at each mapping point. Figure 15 shows BSFC, torque fluctuations and DI ratio results for this testing. Simultaneous injection demonstrated superior BSFC and lower toque fluctuations than PFI or DI injection. The data also shows that the DI ratio decreases as the load becomes higher to achieve the highest torque. This suggests the amount of fuel injected by the DI injector is reduced as the total amount of fuel becomes larger, supporting the former theory that dividing the fuel delivery between two injectors is favorable to promote a homogeneous mixture.

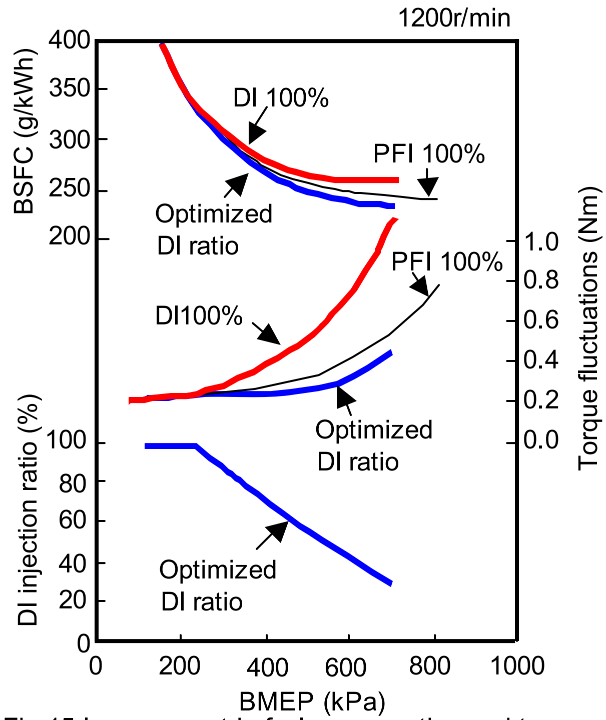


Fig.15 Improvement in fuel consumption and torque fluctuations of D-4S system

4.4 FINAL INJECTION STRATEGY ADOPTED FOR PRODUCTION VEHICLES

The final strategy of D-4S system for introduction to production now will be discussed. Simultaneous injection is utilized to optimize combustion at lower engine speeds. The mixture formation using 100% DI at higher engine speeds is sufficiently homogeneous and combustion efficiency is at a level comparable to simultaneous injection due to the piston speeds. For this reason simultaneous injection is not required at higher engine speeds. Another consideration when simultaneous injection is utilized at higher engine speeds, the injector-tip temperature of the DI injector becomes too high promoting injector deposits and degrading the durability of the injector. According to the research [9], a DI injector has issues when the injector-tip temperature reaches approximately 150°C. For this reason utilization of simultaneous injection is limited. Figure 16 shows the DI ratio of the utilization area for simultaneous injection.

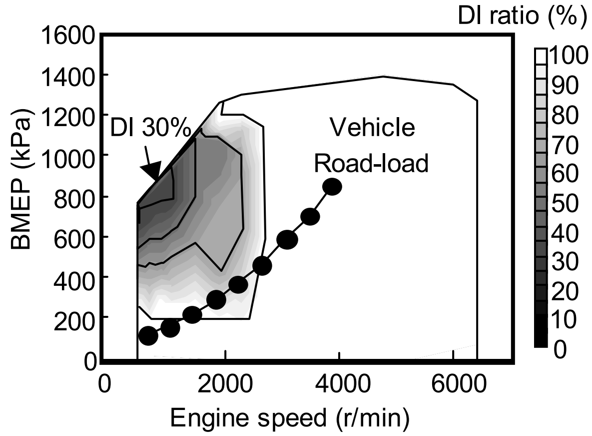


Fig.16 Final injection strategy of DI injection ratio

5. IMPROVEMENT OF EXHAUST EMISSIONS WITH SIMULTANEOUS INJECTION

5.1 ADVANTAGE FOR EXHAUST EMISIONS OF A DISI ENGINE

It is critical that exhaust emissions be reduced during cold start conditions to achieve low emissions levels. Because catalyst efficiency typically exceeds 90% after reaching operating temperature, it is crucial to activate the catalyst rapidly after the engine starts. Figure 17 shows the transition of HC emission for the FTP mode for a LEV engine. More than 80% of the total HC emissions are produced before the catalyst is activated. For this reason quick activation of the catalyst and HC emissions reduction before catalyst warm-up are significant to achieve ULEV and low emission levels.

A DISI engine has advantages for exhaust emissions. One is stratified charge combustion, and other is intense turbulence generated by the DI spray injected at the end of the compression stroke during catalyst warm-up. These two factors permit a mixture to ignite at highly retarded ignition timing. This significantly improves the warm-up speed of the catalyst [1], and it can quickly activate the catalyst in cold conditions.

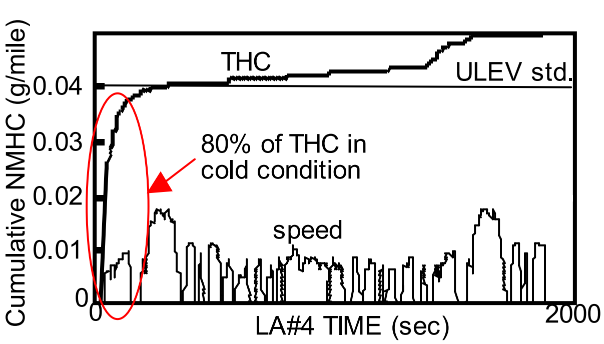


Fig. 17 Transition of HC emissions for FTP mode

5.2 REDUCTION OF HC EMISSIONS DURING CATALYST WARM-UP

A DISI engine has higher engine-out HC emissions than a PFI engine during the catalyst warm-up. Figure 18 shows the direction and strategy for reduction of the exhaust emissions. The horizontal axis is the exhaust gas heat energy supplied to the catalyst, calculated by exhaust gas amount and temperature during 20 seconds of fast idling from the time the engine starts. The vertical axis is the total engine-out HC emissions produced during the same time. The target line is the empirical formula that is derived from the results of vehicles meeting ULEV standards. While a conventional DISI engine that has been introduced in US markets has higher exhaust energy than an equivalent PFI engine, this DISI engine has more engine-out HC emissions during the first 20 seconds of fast idling after the engine starts. Based on this, a DISI engine requires reduced engine-out emissions during catalyst warm-up to meet SULEV standards.

According to the latest research [10,11], dual stage injection of fuel with a DI injector during the intake and the compression strokes is an effective technique to reduce HC emissions during fast idling. The goal of this strategy is stabilization of the mixture to promote ignition at highly retarded ignition timing compared to a conventional DISI engine. The first injection during the intake stroke is critical to forming a stable mixture in the cylinder, while the second injection during the compression stroke aids in forming a stratified mixture for stable ignition at highly retarded ignition. Using this dual stage injection concept reduces emissions in a conventional DISI engine. With the D-4S system, PFI injection can be used to make a stable mixture in the cylinder similar to the intake injection of a conventional DISI engine.

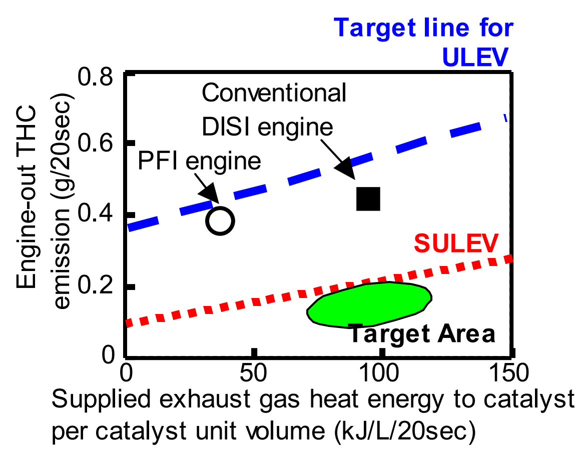


Fig. 18 Direction and strategy for reduction of the exhaust emissions

5.3 REDUCTION OF HC EMISSIONS AT FAST IDLING BY SIMULTANEOUS INJECTION

A study of the effects of simultaneous injection at fast idling was completed. Figure 19 shows the result of torque fluctuations and engine-out HC emissions for various DI injection ratios. Torque fluctuations must be suppressed under the drivability limit to avoid vehicle vibration. Torque fluctuations, obviously, becomes worse as the DI injection ratio is reduced. On the other hand engine-out HC emissions becomes lower with reduced DI injection ratio. Selecting a DI ratio of 65% suppresses torque fluctuations under the drivability limit, while HC emissions can be reduced by approximately 20%. This shows simultaneous injection can reduce HC emissions while maintaining the drivability limit.

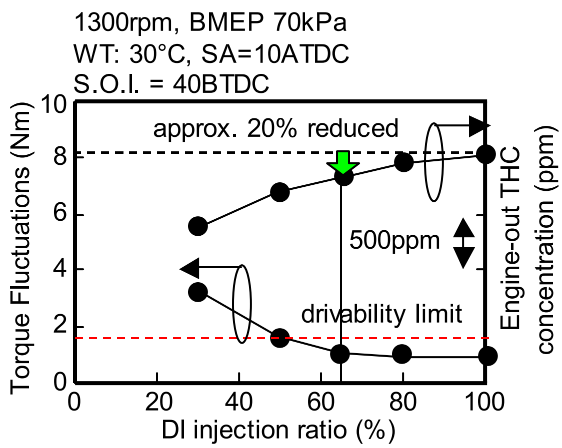


Fig. 19 Effect of simultaneous injection at fast idling

5.4 STUDY OF EFFECT ON REDUCTION OF EMISSION WITH SIMULTANEOUS INJECTION

The effect on emissions reduction with simultaneous injection was studied. Figure 20 shows the result of simultaneous injection from engine cranking to fast idling. PFI injection is utilized during engine cranking to reduce the amount of fuel injected before the engine starts; this reduces engine-out HC emissions by around 20% compared to a conventional DISI engine. Furthermore simultaneous injection is utilized during fast idling after the engine starts. Simultaneous injection can reduce engine-out HC emissions by approximately 20%. Simultaneous injection can suppress torque fluctuations so that the ignition timing can be retarded the same as a DI injection ratio of 100% although the amount of fuel injected by the DI injector during the compression stroke is reduced. As a result, this engine can reduce HC emissions to the target level during the first 20 seconds of fast idling after the engine starts. For this reason this engine can acquire an potential to meet SULEV standards. Besides the hybrid GS450H that carries this engine with a hybrid system, can meet SULEV standards by utilizing PFI injection during engine cranking and simultaneous injection at fast idling.

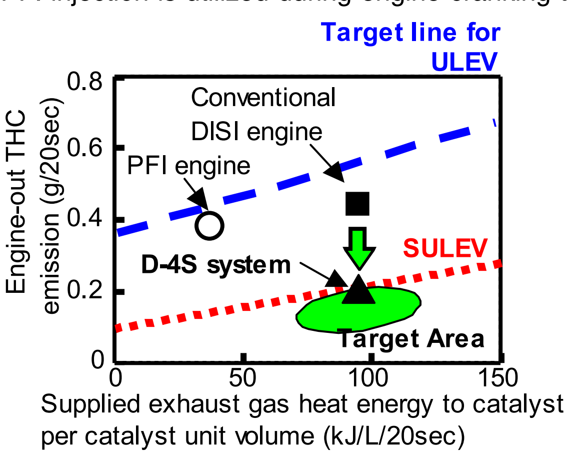
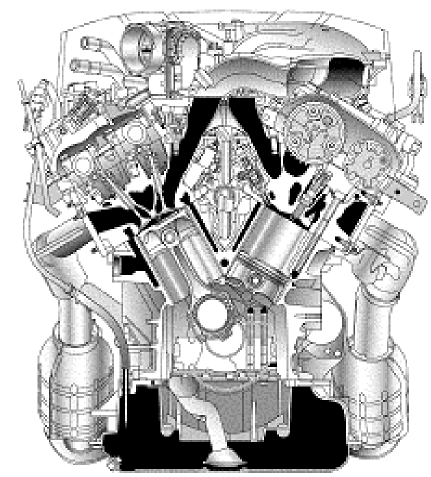


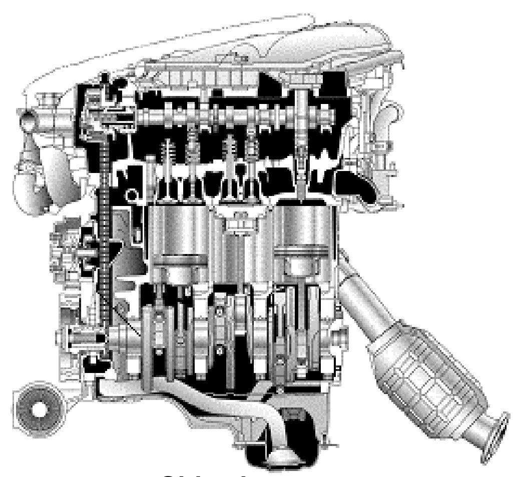
Fig. 20 Result of reduction of THC at fast idling with the simultaneous injection

6. APPLICATION OF THE NEW INJECTION CONCEPT TO A NEW DISI ENGINE

The specifications of the new V-6 3.5-liter DISI engine with D-4S system, introduced to the Japanese and the



Front view



Side view

Fig. 21 Section view of the 2GR-FSE

US markets in 2005 in the LEXUS IS350, will be described. Figure 21 shows section views of a newly developed V-6 3.5-liter engine with D-4S system, 2GR-FSE engine. With D-4S system, this engine achieves high full-load performance and excellent fuel economy. Figure 22 shows the full-load performance for the 2GR-FSE engine compared to an equivalent PFI engine. This engine improves maximum torque by approximately 7% compared to a PFI engine due to the new DISI system, reaching 1.36MPa in brake mean efficient pressure (BMEP). Full-load performance characteristics of this engine are shown in Figure 23. This engine achieves power levels among the highest for production engines of this displacement, 228kW at 6400r/min. Furthermore thanks to the benefits of this direct injection system, this engine generates 90% of the maximum torque from 2000r/min and achieves board torque range of 4500r/min. As for fuel consumption improvement, this engine achieves a minimum brake specific fuel consumption (BSFC) of less than 230g/kWh not only because of friction reduction and increased expansion ratio and reduced pumping loss with the Dual VVT-i, but also combustion improvement by simultaneous injection with D-4S system.

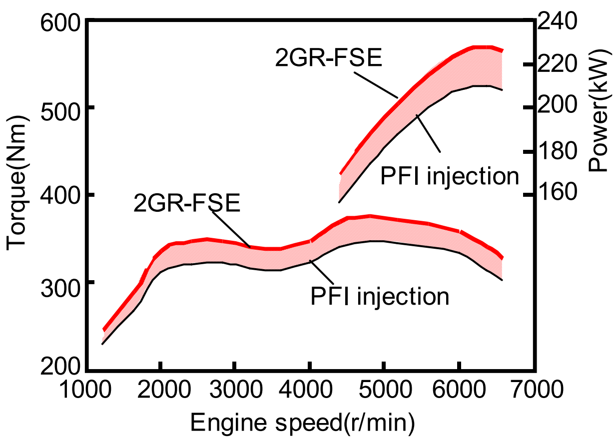


Fig.22 Full-load performance of the 2GR-FSE

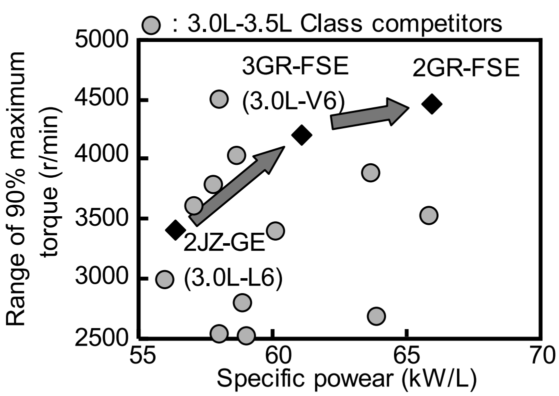


Fig.23 Full-load performance characteristic of the 2GR-FSE

CONCLUSION

This paper described improvement of full-load performance for the D-4S system that has high flow efficiency intake-ports, dual-fan-shaped sprays for the DI injectors and simultaneous injection using PFI injectors and DI injectors. CFD analyses confirmed the dual-fan-shaped sprays satisfy the requirements for full-load performance. The specification of the new 2GR-FSE V-6 3.5-liter engine and the vehicle performance with this engine were described.

1. With the aim of improvement of full-load performance, a high efficiency intake-port is adopted. But combustion efficiency at lower engine speeds deteriorates due to less homogeneity of the mixture formation in the cylinder. The newly developed dual-fan-shaped spray DI injector promotes a homogeneous mixture without any devices to generate intense air-motion to improve full-load performance.

2. It was revealed that adoption of the new spray and higher efficiency intake-port cannot sufficiently suppress torque fluctuations at part loads due to a heterogeneous mixture. To improve the mixture formation, a PFI injection is installed. Simultaneous injection of two injectors can improve combustion over a PFI only system due to a more homogeneous mixture.

3. Simultaneous injection is effective in reduction of HC emissions during catalyst warm-up under cold conditions. Furthernore utilizing PFI injection during engine cranking can reduce HC emissions: these can bring this engine a potential for SULEV standards.

4. The 2GR-FSE engine has been developed with D-4S system, and this engine contributes significantly to improved vehicle performance. With this engine, the LEXUS IS350 delivers class leading full-load performance, and excellent fuel economy while meeting low emission standards.

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