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# Development of a New V-6 High Performance Stoichiometric Gasoline Direct Injection Engine

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## ABSTRACT

A new V-6 stoichiometric gasoline direct injection engine was developed for high performance FR (Front Engine Rear Drive) vehicles. High power performance, low fuel consumption and low exhaust emissions were achieved by employing a stoichiometric direct injection system that uses Toyota's unique slit nozzle injector that generates a fan-shaped fuel spray and variable intake and exhaust valve timing systems.

Focusing on the power performance, maximum power of 183kW (61kW/L) is achieved at 6200rpm and maximum torque is 312Nm at 3600rpm. This power performance is among the top production 3.0 L gasoline engines in the world.

This paper outlines the features of this engine and some special technologies contributing to the achievement of the above-mentioned high performance.

Optimizing the intake-port design was done to improve power performance. In a typical conventional intake-port design, flow coefficients are decreased to generate in-cylinder charge-motion that is required for homogeneous mixture formation. The advantage of the direct injection system could not be utilized sufficiently in this type of engine. In the development of the stoichiometric gasoline direct injection engine, an intake-port that obtains both a high flow coefficient and high tumble ratio, was designed by employing various analyses techniques. Improvement of power performance, therefore, could be obtained efficiently as an advantage of the direct injection system.

To improve fuel consumption, Dual VVT-i (variable intake and exhaust valve timing) systems were employed to decrease pumping loss and increase the expansion ratio. As a result of combustion pressure analyses, it was found that the initial combustion speed could be improved with internal exhaust gas recirculation (EGR) and the engine could be operated with larger amounts of EGR.

One of the advantages of the direct injection system was

utilized to improve exhaust emissions under cold start and warm-up conditions. The amount of fuel injected during cold start conditions could be significantly reduced by adopting stratified charge combustion with sufficiently atomized fuel spray by high fuel pressure.

During warm-up, stratified charge combustion, enables retarded spark ignition timing to significantly improve the warm-up speed of the catalyst, while homogeneous combustion promotes optimum performance once the engine reaches operating temperature.

## 1. INTRODUCTION

Recently, many car makers have introduced gasoline direct injection spark ignition (DISI) engines to the Japanese and European markets. Combustion concepts and objectives of these engines are mainly grouped into two types. One is the stratified charge combustion concept which can improve fuel economy by lean burn operation while the second type is the homogeneous charge stoichiometric combustion concept which can obtain high WOT performance by utilizing latent heat of injected fuel.

Toyota launched the stratified DISI engine that was called D-4 in 1996 [1], and introduced a new stratified combustion process as the 2nd generation D-4 in 1999

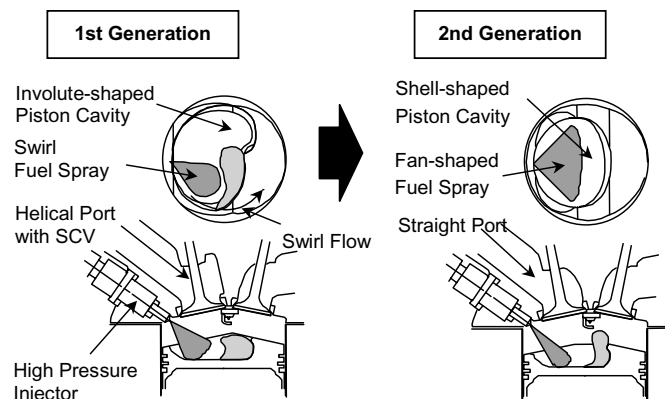


Figure 1. Combustion System Configuration

[2]-[3]. These combustion concepts achieved excellent fuel economy under the 10-15 mode Japanese emissions cycle. Figure 1 shows the combustion process of the 1st and the 2nd generation D-4 engines. The 1st generation D-4 engine employs swirl nozzle injectors with the stratified mixture guided to the vicinity of the spark plug by in-cylinder swirl flow. The 2nd generation D-4 engine has slit nozzle injectors with the stratified mixture guided by the piston's oval-shaped wall cavity. This new combustion system achieves a wide range of stratified combustion. The slit nozzle injector generates a fan-shaped spray with wide dispersion and

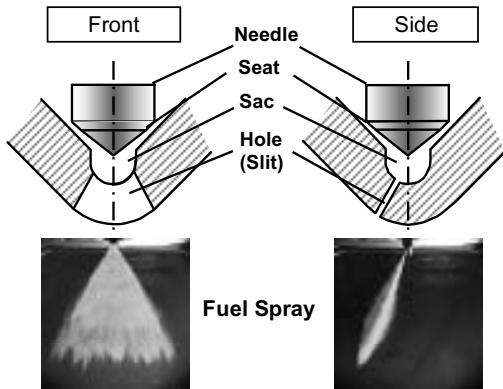


Figure 2. Slit Nozzle and Fan-shaped Spray

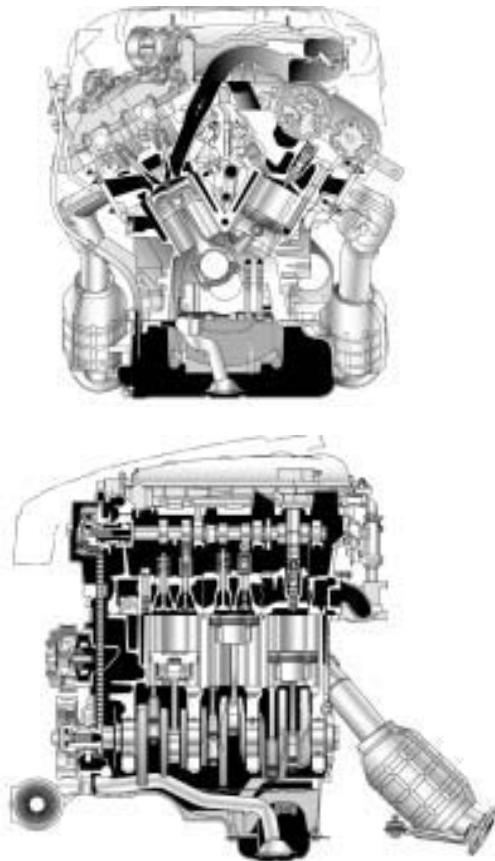


Figure 3. Section View of 3GR-FSE

Type	6 Cylinder 60° V Angle
Bore × Stroke	87.5 × 83.0 mm
Displacement	2994 cc
Compression Ratio	11.5
Valve Train	Intake & Exhaust VVT-i
Max Torque	312 Nm
Max Power	183 kW ( 61 kW/L )
Range of 90% Torque	4200 rpm
Fuel Economy	over 29.0 mile/gallon
Exhaust Gas Emission	LEV1/ULEV (California)

Table 1. Engine Specification (for North America)

moderate penetration. A section view of the slit nozzle and photograph of fan-shaped spray are shown in Figure 2.

A new V-6 3.0L engine (3GR-FSE) was introduced to the Japanese market at the end of 2003, and to the North American European markets, for the new Lexus GS300 at the beginning of 2005. This new V-6, gasoline direct injection engine (Figure 3) employs the above-mentioned 2nd generation stoichiometric D-4 system mainly to obtain high WOT performance and high thermal efficiency. An outline of this new V-6 engine is shown in Table1. Employing not only stoichiometric direct injection, this engine also uses intake and exhaust variable valve timing (Dual VVT-i) systems, and reduced engine friction, to achieve a maximum power of 183kW at 6200rpm, (61kW/L) and maximum torque is 312Nm at 3600rpm. This WOT performance is near the top of 3.0L naturally aspirated gasoline engines in the world with respect to engine performance. Additionally ULEV exhaust emission standards were easily met by using fast warm-up technology with stratified combustion. Some technologies that achieved the above-mentioned high performance in the development of this new V-6 stoichiometric DISI engine will now be described.

## 2. IMPROVEMENT OF WOT PERFORMANCE

### 2.1 OPTIMIZATION OF INTAKE-PORT DESIGN

To obtain both high volumetric efficiency and high combustion speed, intake-port design optimization was done. High tumble ratio is required to form a homogeneous mixture to obtain excellent and stable combustion. Figure 4 shows the effect of tumble ratio on thermal efficiency and temperature of the start catalyst at 4000rpm WOT condition. Both parameters are improved by high tumble ratio. In conventional intake-port designs, higher tumble ratios usually generate lower coefficients of flow. To obtain both high tumble ratio and high coefficient of flow, the intake-port design was optimized by CFD analyses and flow visualization. Calculated cylinder model and an outline of the calculation method are

shown in Figure 5. When the mixture formation is calculated, DDM spray model results were verified by rig testing of the slit-nozzle injector. Comparison of the calculated intake-port and in-cylinder charge motion between the conventional intake-port and newly developed one is shown in Figure 6. In the case of the develop-

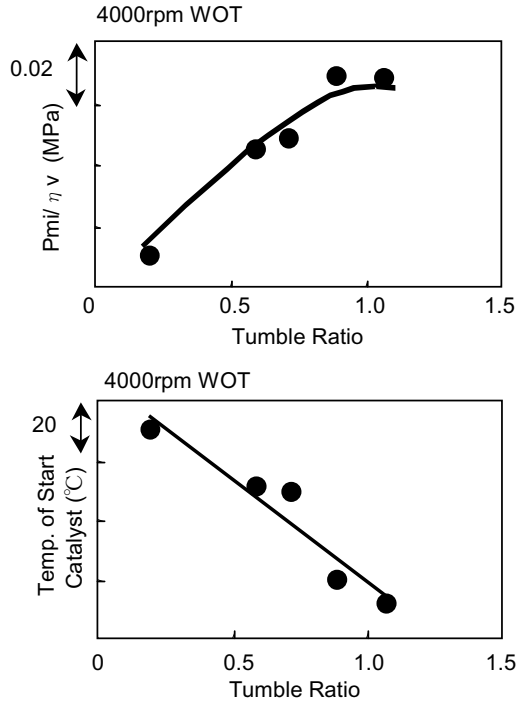


Figure 4. Effect of Tumble Ratio on WOT Combustion

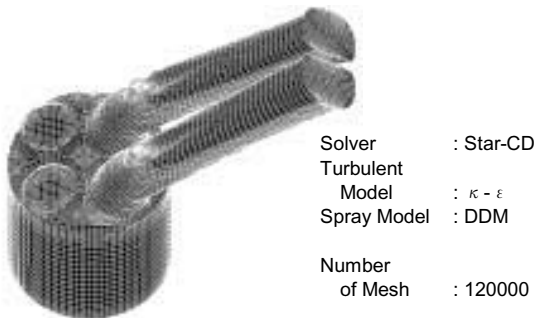


Figure 5. Model for CFD Analysis

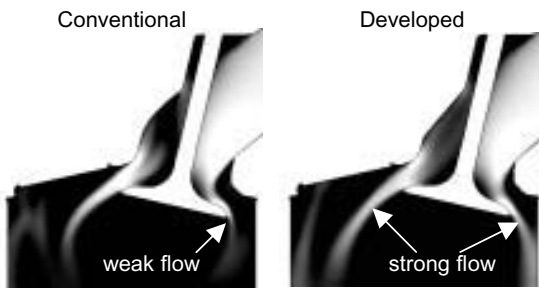


Figure 6. Comparison of CFD Results

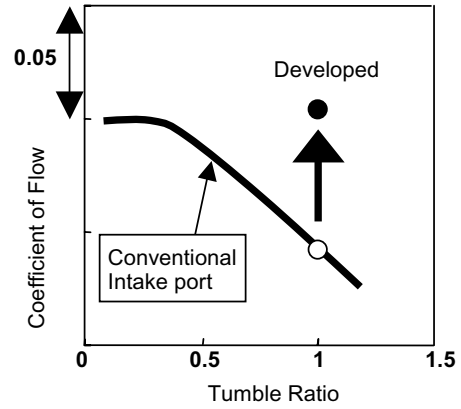


Figure 7. Effect of Developed Intake-port

ed intake-port, strong airflow can be generated to both the intake side and exhaust side, and a high tumble ratio and high coefficient of flow are obtained as shown in Figure 7.

## 2.2 OPTIMIZATION OF PISTON CAVITY

Homogeneous combustion does not require a deep piston cavity. However, a shallow piston cavity is required for the cold start emissions to guide the stratified mixture to the vicinity of the ignition plug. Cold start emissions can be significantly improved by adopting stratified combustion. Figure 8 shows the effect of the depth of the piston cavity on WOT performance. The cavity depth was normalized by dividing the cavity depth ( $h$ ) by the piston diameter ( $D$ ). In the case of the deep cavity, the amount of fuel wetted on the piston surface is increased and the over-rich zone of the mixture tends to be formed in the cavity. To avoid such affects a shallow cavity was selected for this engine. The function of guiding the stratified mixture under cold start conditions was also confirmed for this design.

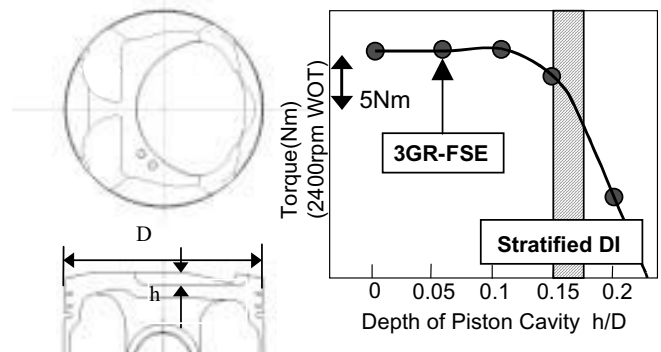


Figure 8. Effect of Piston Cavity

### 2.3 IMPROVEMENT OF WOT COMBUSTION

Combustion speed was significantly improved by the above-mentioned intake-port and piston designs. Figure 9 shows the comparison of in-cylinder pressure between the newly developed engine (3GR-FSE) and a conventional PFI (Port Fuel Injected) engine. The compression

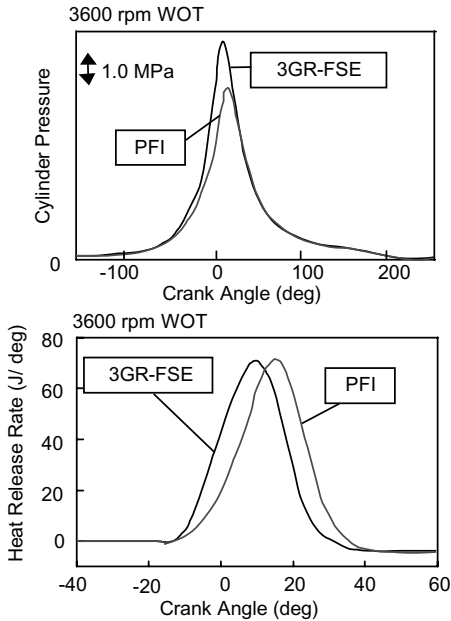


Figure 9. Improvement of Combustion Speed

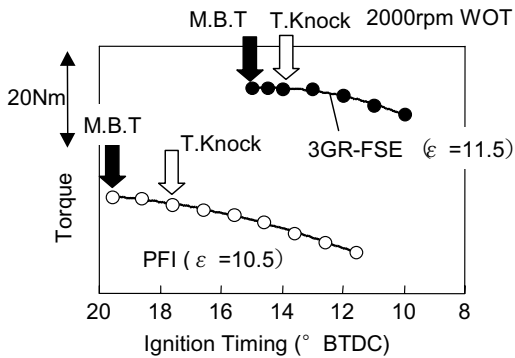


Figure 10. Improvement of Compression Ratio

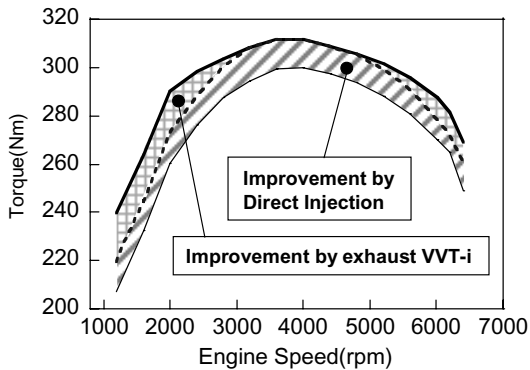


Figure 11. WOT Performance of 3GR-FSE

ratio of the PFI engine for this comparison is 10.5:1. By increasing the combustion speed, MBT is considerably retarded and trace knock ignition timing can be relatively advanced for the new 3GR-FSE despite its higher compression ratio as shown in Figure 10. WOT performance obtained is shown in Figure 11. In this figure, WOT performance of 3GR-FSE was significantly improved by DISI and the Dual VVT-i system.

### 3. IMPROVEMENT OF FUEL ECONOMY

The Dual VVT-i system was employed not only for WOT performance but also for fuel economy. Figure 12 shows a comparison of the valve overlap effect on fuel consump-

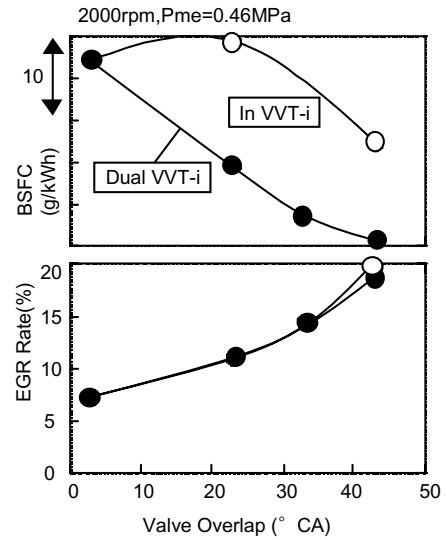


Figure 12. Advantage of Dual VVT-i on BSFC

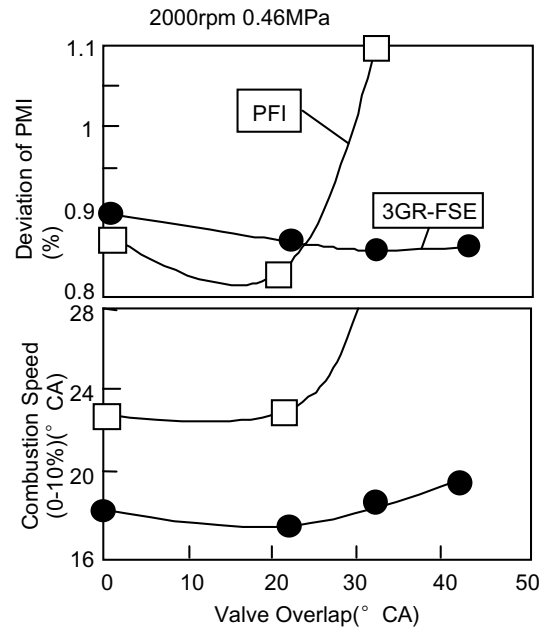


Figure 13. Improvement of EGR Limit

ption and the amount of internal EGR under partial load condition between intake VVT-i and the Dual VVT-i. Fuel consumption is improved when the valve overlap is increased because of a reduction in pumping losses. In the case of Dual VVT-i, additional improvement of fuel consumption was obtained when the exhaust valve closing timing was retarded as shown in this figure.

This is caused by an increase in the expansion ratio. Internal EGR causes a decrease in combustion speed, so valve overlap is limited by engine torque fluctuation. Figure 13 shows the comparison of valve overlap (EGR) sensitivity for combustion between the conventional PFI engine and 3GR-FSE. In the case of 3GR-FSE, deterioration of combustion speed and increased torque fluctuation caused by increasing the amount of internal EGR gas is relatively small. The comparatively high tumble ratio was confirmed to be the main reason for this increased internal EGR tolerance.

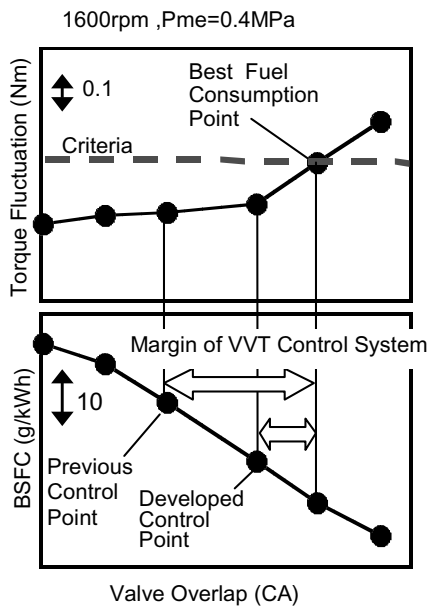


Figure 14. Effect of Valve Overlap Control System

Combustion fluctuation is typically the criteria used to determine valve overlap for VVT-i control systems. However, as the system has a tolerance, the valve overlap is shortened to provide a margin. In this new V-6 Engine, Dual VVT-i (exhaust VVT-i was added to intake VVT-i) was adopted and the tolerance was doubled. In this new VVT-i control system, valve overlap can be more tightly controlled, which influences combustion directly compared to an ordinary single VVT-i. By using this system, the margin was reduced, resulting in improved fuel economy (Figure 14).

#### 4. EXHAUST EMISSION CONTROL UTILIZING STRATIFIED CHARGE COMBUSTION

In the development of conventional PFI engines, WOT performance tends to be sacrificed to improve exhaust emissions. For example, reducing the thermal inertia of the exhaust manifolds and increasing the cell density of the catalytic converter is required. This engine was desi-

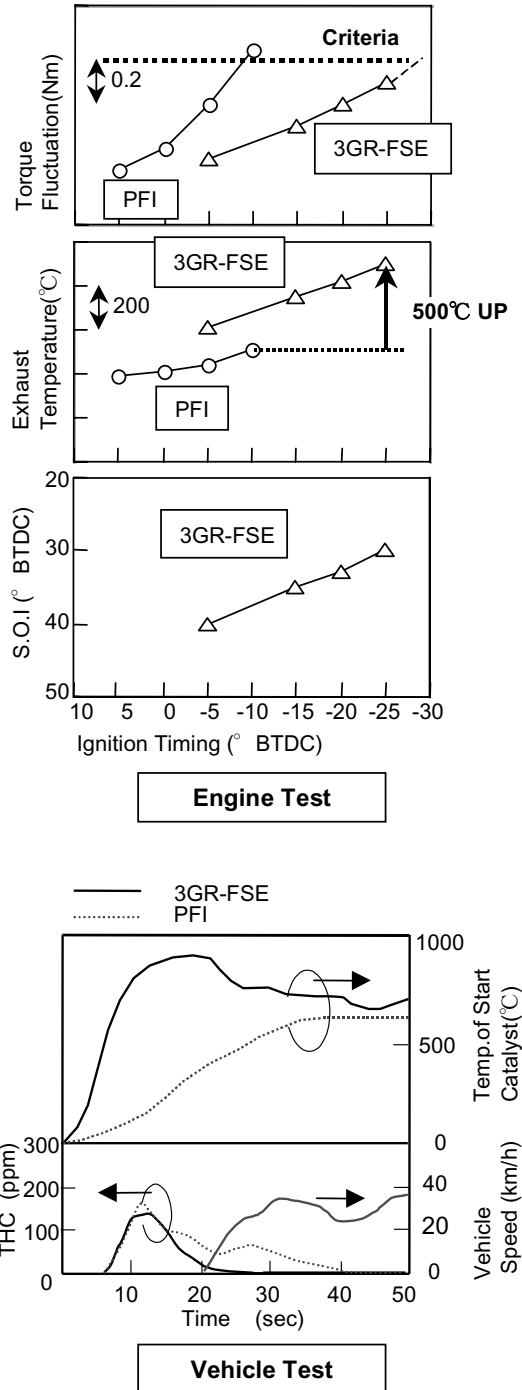
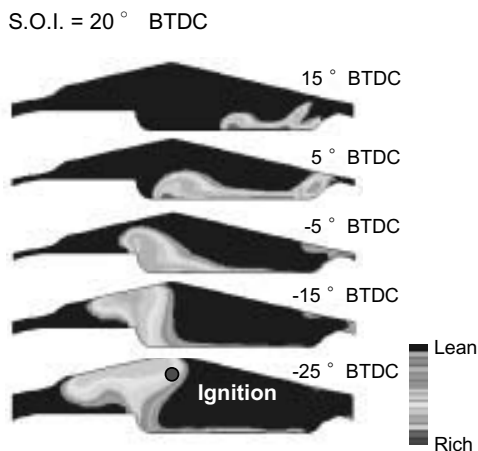


Figure 15. Emission Control Utilizing Stratified Charge Combustion

ned to meet ULEV exhaust emission standards without sacrificing WOT performance by applying stratified combustion during warm-up. Figure 15 shows the engine and vehicle test results of late injection during warm-up. As shown in the engine test results, exhaust gas temperature can be significantly increased when fuel is injected during the compression phase and ignition timing can be retarded. This effect is limited by engine torque fluctuations to avoid vehicle surge. Five hundred degrees centigrade of exhaust temperature rise was realized compared with the conventional PFI engine at the same torque fluctuation level. Applying this methodology, THC emissions during vehicle start up were significantly reduced as shown in Figure 15.

Figure 16 shows the stratified mixture formation during fast idle calculated by the CFD analysis described previously. Fuel is injected during the compression phase and guided near the ignition plug by the piston cavity wall. It is found that the stratified mixture enables the ignition timing to be retarded during fast idle conditions.



**Figure 16. CFD Analysis on Stratified Mixture Formation (Fast Idle)**

## 5. CONCLUSION

1. A new V6 3.0L stoichiometric direct injection spark ignition engine was developed for the new Toyota Crown in the Japanese market and the new Lexus GS-300 for the North American and European markets. This engine uses stratified charge combustion during cold start and homogeneous combustion for optimum performance once the engine has reached operating temperature.
2. Advantages of DISI and Dual VVT-i systems for WOT performance were obtained by the newly developed intake-port which has both high tumble ratio and a high coefficient of flow.

3. The high compression ratio achieved by DISI and low pumping loss and high expansion ratio realized by the Dual VVT-i system significantly contributed to the excellent fuel economy of these vehicles.
4. A cold emission strategy for the DISI engine was developed, with stratified charge combustion resulting in reduced HC emissions during warm-up. This advantage enabled Toyota to meet ULEV emission standards employing simple exhaust systems optimized for WOT performance.

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