

New Combustion Concept for Turbocharged Gasoline Direct-Injection Engines

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ABSTRACT

The advantages of gasoline direct-injection are intake air cooling due to fuel vaporization which reduces knocking, additional degrees of freedom in designing a stratified injection mixture, and capability for retarded ignition timing which shortens catalyst light-off time. Stratified mixture combustion designs often require complicated piston shapes which disturb the fluid flow in the cylinder, leading to power reduction, especially in turbocharged gasoline direct-injection engines. Our research replaced the conventional shell-type shallow cavity piston with a dog dish-type curved piston that includes a small lip to facilitate stratification and minimize flow disturbance. As a result, stable stratified combustion and increased power were both achieved.

CITATION: Mitani, S., Hashimoto, S., Nomura, H., Shimizu, R. et al., "New Combustion Concept for Turbocharged Gasoline Direct-Injection Engines," *SAE Int. J. Engines* 7(2):2014, doi:10.4271/2014-01-1210.

1. INTRODUCTION

Direct-injection gasoline engines are becoming more widespread in recent years due to their improved fuel efficiency and power by increased volumetric efficiency, and reduced knocking due to intake air cooling. Since these engines have a larger knock reduction effect under high-load conditions which are conventionally more susceptible to knocking, direct injection is considered to be highly compatible with turbocharging. In addition, reducing the size of the engine by using a turbocharger to boost torque increases the engine load while driving and enables synergistic improvements in fuel efficiency. For these reasons, technology for direct-injection turbocharged engine downsizing is being actively developed [1], [2], [3].

Another advantage of direct injection is stratified mixture formation. Direct injection allows a greater degree of injection freedom to form a stratified mixture immediately after engine start. Utilizing the high combustion stability of direct-injection, ignition can be greatly retarded to achieve earlier catalyst lightoff. It is therefore an extremely effective technology for addressing exhaust-emissions related issues [4]. In contrast, direct-injection often requires a complicated piston shape for stratified mixture formation, which may reduce in-cylinder flow and the intensity of turbulence. This has the effect of worsening homogenous combustion and full-load characteristics. These trends are particularly prominent in turbocharged engines that use large intake air and fuel injection volumes.

Therefore, this research aimed to improve the trade-off between power and stratified mixture formation in direct-injection gasoline engines. The conventional shell-type shallow cavity piston was replaced with a dog dish-type curved piston that includes a small lip to facilitate stratification. In addition, the injection angle and intake ports were optimized to minimize flow disturbance and to intensify in-cylinder flow.

As a result, the developed combustion concept is capable of increasing power while maintaining stratified combustion stability immediately after engine start. This paper describes the results of verification tests for this combustion concept.

2. BASIC CONCEPT

The fan spray shown in [Fig. 1](#) induces intense turbulence inside the mixture due to spray shear. Research has also found that optimizing (i.e., synchronizing) the intake flow and the injection direction increases this effect and improves both the intensity of in-cylinder turbulence and mixture homogeneity [5]. Therefore, this concept adopted a fan spray.

This combustion concept replaced the conventional shell-type shallow cavity piston ([Fig. 2](#)), which is advantageous for mixture stratification, with a dog dish-type curved piston ([Fig. 3](#)) to minimize disturbance of tumble. This new piston shape also includes a small wall guide, or lip, which enables the stratified operation in the catalyst warm-up mode.

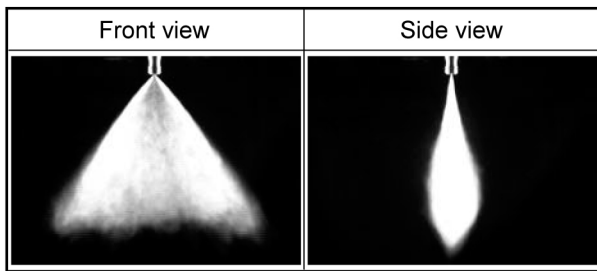


Figure 1. Fan spray.

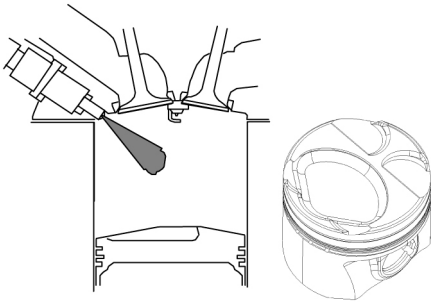


Figure 2. Conventional type: shell-type shallow cavity piston.

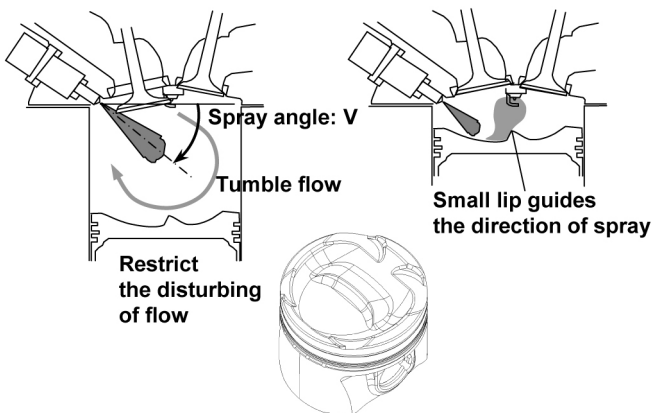


Figure 3. New combustion concept.

These measures increased power by intensifying in-cylinder flow while maintaining stratified combustion stability in the catalyst warm-up mode.

3. VERIFICATION OF PRINCIPLES OF NEW CONCEPT: MEASUREMENT OF INTENSIFICATION OF IN-CYLINDER FLOW TURBULENCE BY NUMERICAL CALCULATION

The potential flow intensification effect of the new combustion concept was verified in advance by numerical calculation (see Section 4.3 for the verification method). The test was performed under full-load conditions at an engine speed of 4,000 rpm. Figure 4 shows the unsteady tumble ratio and Fig. 5 shows the velocity vectors at the center vertical cross section of the two cylinder types. The velocity vector diagram indicates separation of the flow at section A at the conventional piston

crown surface. In contrast, the new concept reduces this separation phenomenon and achieves excellent rotation of the main flow, which increases the tumble ratio (Fig. 4).

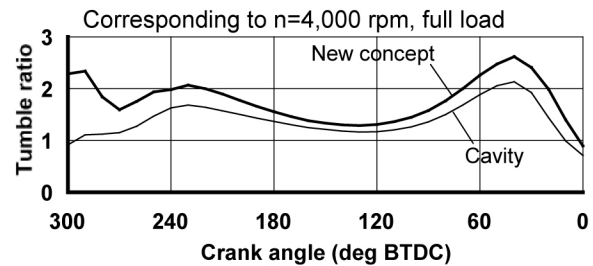


Figure 4. Comparison of calculated unsteady tumble ratio.

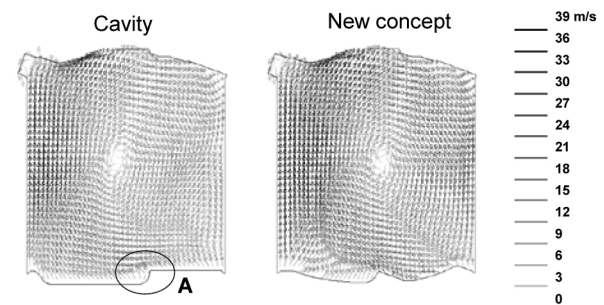


Figure 5. Comparison of calculated velocity vector at cylinder center cross section.

Furthermore, this concept intensifies the turbulence in the latter part of the compression stroke by causing compression and shear collapse of the tumble flow due to the higher tumble ratio (Fig. 6).

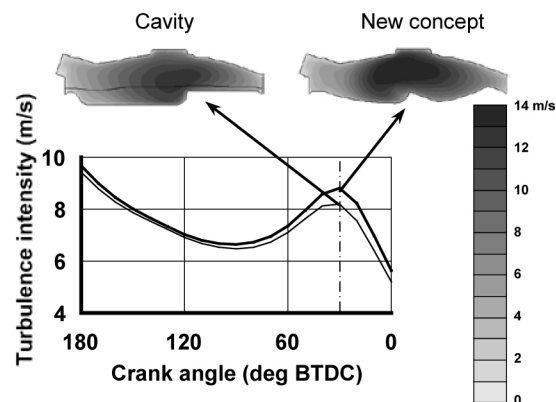


Figure 6. Comparison of calculated turbulence intensity.

Since these calculations demonstrated the potential effectiveness of the new concept, performance was then verified through further numerical calculations and tests.

4. TEST AND CALCULATION RESULTS

4.1. Test Engine Specifications

The tests used a prototype turbocharged water-cooled 4-cylinder direct-injection engine with the main specifications shown in Table 1.



Table 1. Specifications of engine.

Engine type	Turbocharged gasoline direct-injection engine Inline 4-cylinder DOHC 16-valve prototype
Bore × stroke	71.5×74.5 mm
Displacement	1,197 cm ³
Compression ratio	10.0
Fuel type	Japanese regular gasoline (RON 90)

The compression ratio was set to 10.0 and regular gasoline with an octane number (RON) of 90 was used as the fuel.

4.2. Test and Measurement Methods

4.2.1. Maximum Power

The maximum power test conditions were as follows: engine speed = 5,200 rpm, wide-open throttle (WOT), coolant outlet temperature = 88°C, oil temperature = 110°C, injection pressure = 20 MPa, injection timing = 320 deg BTDC, equivalence ratio $\phi = 1.33$ (constant), and maximum catalyst temperature = restricted to 920°C.

4.2.2. Combustion in Catalyst Warm-Up Mode

Assuming a vehicle immediately after engine start, the catalyst warm-up mode test conditions were as follows: engine speed = 1,400 rpm, mean effective pressure P_{me} = 126 kPa, coolant outlet temperature = 40°C, oil temperature = 40°C, equivalence ratio $\phi = 0.94$ (constant), ignition timing = 20 deg ATDC, injection pressure = 13 MPa, injection = two injections in the intake and compression stroke at a ratio of 40:60.

For both the maximum power and catalyst warm-up combustion tests, the total hydrocarbon (THC) concentration in the exhaust was measured by flame ionization detection (FID), CO and CO₂ were measured by non-dispersive infrared (NDIR) analysis, and NO_x was measured by chemiluminescence (CLD) analysis. The smoke concentration was measured using a reflection type smoke meter.

The equivalence ratio and intake air volume were calculated based on the THC, CO, and CO₂ concentrations and fuel flow. The cylinder pressure was obtained over 200 cycles using a piezo pressure transducer.

4.3. Method of Numerical Calculation

The commercially available fluid analysis code STAR-CD was used to calculate the in-cylinder flow and deepen understanding of the flow phenomena. Table 2 and Fig. 7 show an outline of the calculation method.

Table 2. Setup conditions of simulation.

CFD code	STAR-CD
Differencing scheme	MARS
Turbulence model	RNG k- ϵ
Spray model	Discrete droplet model (DDM)
Spray impinging model	Bai model
Fuel	n-heptane
Number of cells	Approximately 930,000
Time step	100 μ s

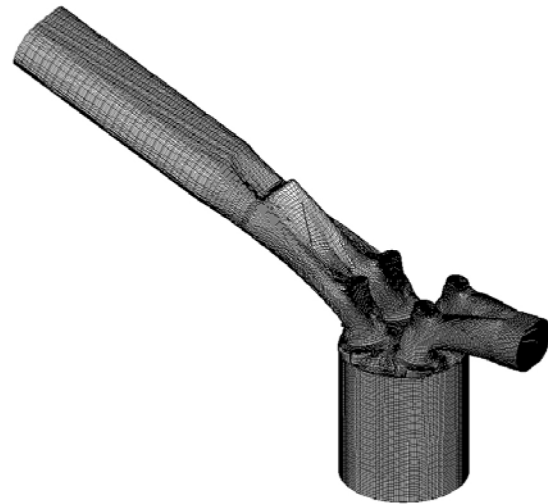


Figure 7. Calculation model of test engine.

4.4. Basic Characteristics

4.4.1. Preliminary Study: Optimization of Spray V Angle

In the case of the fine spray created using this concept, the spray V angle shown in Fig. 3 affects the power of the engine. This angle was optimized in advance.

Figure 8 shows the effect of the spray V angle on power and combustion characteristics. The η_{vb} value in the figure is calculated by multiplying the volumetric efficiency with atmospheric pressure and dividing by the intake manifold pressure. It corresponds to the unboosted volumetric efficiency at WOT.

The V angle has conflicting effects on η_{vb} . If the V angle is widened, interference between the injection spray and the intake valve is reduced, increasing the degree of intake air cooling. This causes η_{vb} to rise. In contrast, the injection and intake air synchronization effect decreases [5], in-cylinder turbulence is reduced, and the combustion duration becomes longer (top line in Fig. 8). Maximum power can be achieved at $V=41^\circ$, which improves η_{vb} while minimizing flow synchronization loss. Consequently, the V angle was fixed to 41° and the new combustion chamber shape concept was introduced. The effect of the new combustion chamber shape was then investigated.