

# A Study of a Multistage Injection Mechanism for Improving the Combustion of Direct-Injection Gasoline Engines

Yoshihiro Imaoka, Kiyotaka Shouji, Takao Inoue, and Toru Noda  
Nissan Motor Co., Ltd.

## ABSTRACT

Technologies for improving the fuel economy of gasoline engines have been vigorously developed in recent years for the purpose of reducing CO<sub>2</sub> emissions. Increasing the compression ratio for improving thermal efficiency and downsizing the engine based on fuel-efficient operating conditions are good examples of technologies for enhancing gasoline engine fuel economy. A direct-injection system is adopted for most of these engines. Direct injection can prevent knocking by lowering the in-cylinder temperature through fuel evaporation in the cylinder. Therefore, direct injection is highly compatible with downsized engines that frequently operate under severe supercharging conditions for improving fuel economy as well as with high compression ratio engines for which susceptibility to knocking is a disadvantage. On the other hand, direct-injection engines have certain issues such as the need to reduce particulate matter (PM) emissions, and technical measures must be developed for that purpose. Multistage injection is one method of improving direct-injection engines and has both advantages and disadvantages. One benefit of multistage injection is lower PM emissions caused by liquid fuel impinging on the cylinder wall. That is because multistage injections reduce the momentum of each fuel injection, and spray tip penetration is shorter than for a single injection. A second benefit is that the homogeneous quality of the fuel-air mixture is improved by the dispersion of the fuel spray. This report presents a detailed analysis of the following two new effects. One is increased air charging in the cylinder by the pressure oscillations generated by multiple fuel injections. The other is lower fuel consumption by improving the trade-off between mixture homogeneity and the cooling effect. By using computational fluid dynamics, in-cylinder visualization and tests of an actual engine, the mechanism involved in these four effects was analyzed, including consideration of the fuel injection state, fuel-air mixture formation and the effect of the in-cylinder temperature.

**CITATION:** Imaoka, Y., Shouji, K., Inoue, T., and Noda, T., "A Study of a Multistage Injection Mechanism for Improving the Combustion of Direct-Injection Gasoline Engines," *SAE Int. J. Engines* 8(3):2015, doi:10.4271/2015-01-0883.

## INTRODUCTION

Technologies for improving the fuel economy of gasoline engines such as increasing the compression ratio and downsizing have been vigorously developed in recent years for the purpose of reducing CO<sub>2</sub> emissions. A direct-injection (DI) system can prevent knocking by lowering the temperature through fuel evaporation in the cylinder. Therefore, a DI system has been adopted for most of these engines. Meanwhile, exhaust emission regulations applied to vehicles and engines have become much more stringent in recent years for the sake of conserving the natural environment. The regulation on particulate matter (PM) emissions from DI engines will be tightened further. That will require the adoption of technologies for complying with the tighter emission regulation while making the most of the advantages of DI engines, including their high levels of fuel economy, torque and power. Combustion system designs are being researched for reducing the amount of fuel impinging on the cylinder wall as a measure against PM generation [1, 2]. Reducing spray tip penetration results in less fuel impinging on the cylinder wall, which mitigates PM emissions.

This study examined a multistage fuel injection mechanism capable of improving the performance of DI gasoline engines both by extending their advantages and addressing their disadvantages. In-cylinder phenomena were identified by using an in-cylinder visualization system and computational fluid dynamics (CFD) calculations, and then methods of applying a multistage injection pattern were investigated. The results revealed the four following effects of multistage injections, and the results of investigations into each effect will be described here.

First, an investigation was made of the effect of multistage injections on reducing spray tip penetration. Next, the effect of spray dispersion achieved through multistage injections on improving the air-fuel mixture was investigated. Improvement of the air-fuel mixture is necessary for DI engines.

One generally known advantage of DI engines is that the cooling effect due to latent heat of vaporization allows more air to be inducted in the next cycle. In this study, a DI engine was used to investigate the mechanism whereby intake air pulsations produced by the vaporization of the fuel spray in the cylinder can be used to improve charging efficiency.

The fourth effect examined was the improvement of in-cylinder cooling. A relationship between the injection timing and the cooling effect was observed. Therefore, a method of applying a multistage injection pattern for improving the cooling effect was found.

## TEST APPARATUS AND METHOD

In order to examine the four benefits of multistage injections mentioned above and ways of utilizing this injection method, an investigation was made of the feasibility of applying a multistage injection pattern to an actual engine. Table 1 lists the engine specifications used in the investigation. An inline 4-cylinder 1.6L turbocharged engine was used in conducting experiments to evaluate the effect on engine performance of multistage injections.

Figure 1 shows the configuration of the laser-induced fluorescence (LIF) measurement system used for in-cylinder visualization. This visualization system was used to study the homogeneity of the air-fuel mixture when multistage injection was applied. In-cylinder visualizations were facilitated by equipping the engine with a synthetic quartz glass piston liner and a glass piston to make it optically accessible. With the LIF measurement system, a cross-section of the cylinder 1 mm below the cylinder head surface was irradiated with a 1.5-mm-thick laser sheet light and LIF images were taken with a shuttered camera through the glass piston. The measurement timing was set at 330 deg. after top dead center (ATDC).

Table 1. Engine specifications

|                          |                         |
|--------------------------|-------------------------|
| Engine Type              | Inline 4                |
| Displacement cc          | 1618                    |
| Bore x Stroke            | 79.7 x 81.1             |
| Compression Ratio        | 10.5:1                  |
| Fuel Supply System       | Direct Injection (side) |
| Injector Type            | Multihole               |
| Number of Holes          | 6                       |
| Max. Fuel pressure (MPa) | 20                      |
| RON                      | 95                      |
| Aspiration               | Turbochager             |
| Max. Boost (kPa)         | 95                      |
| Valve Train              | Int. & Exh. CVTC        |

The computational grid used in the CFD calculations is shown in figure 2. The computational grid used in the 3D calculations had a 0.5 to 3 mm mesh. These calculations were performed to investigate the spray condition, air-fuel mixture and the mechanism whereby multistage injection increases the intake air charge. Calculations were performed with STARCD, a general-purpose CFD analysis code. The renormalization group (RNG) k- $\epsilon$  model [8] was used as the turbulence model. In the discretization procedure, the implicit scheme was used as the time propagation method, the Monotone Advection and Reconstruction Scheme (MARS) was used to discretize the convection term, and the Pressure-Implicit with Splitting of Operators (PISO) algorithm was used as the solution for velocity and pressure. The physical models used are listed in Table 2. The Kelvin-Helmholtz/Rayleigh-Taylor (KH-RT) hybrid model [12] was used as the spray model. This model combines the KH model of liquid jet

breakup with the RT model of the breakup of liquid drops beyond the breakup length. Fuel viscosity that affects the breakup is adjusted by the temperature.

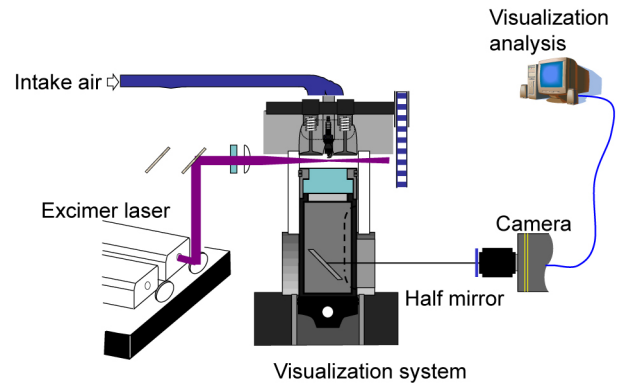


Fig. 1. LIF measurement system

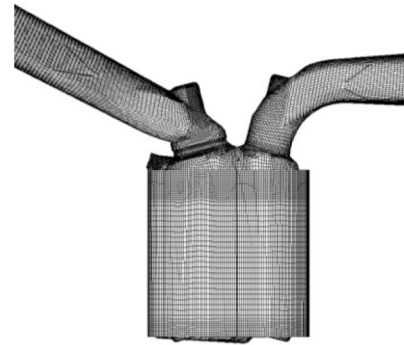


Fig. 2. CFD computational grid

Table 2. Configuration of simulation model

|                          |                               |
|--------------------------|-------------------------------|
| Turbulence model         | RNG k- $\epsilon$ model [8]   |
| Wall heat transfer       | Angelberger wall function [9] |
| Initial droplet size     | 0D cavitation model [10, 11]  |
| Spray breakup            | KH-RT model [12]              |
| Droplet evaporation      | Ranz-Marshall model [13]      |
| Droplet-wall interaction | Senda model [14]              |
| Fuel film evaporation    | Spalding                      |
| Fuel film stripping      | IFP model [15]                |

## RESULTS AND DISCUSSION

### Spray Tip Penetration

The results of previous studies have shown that multistage fuel injections reduce spray tip penetration [5, 6]. The effect of this reduction was investigated by spray visualization in a constant-volume vessel. figure 3 presents visualized spray images at 2 ms after the start of injection for two injection durations ( $T_{id}$ ). It is seen that spray penetration is reduced with the shorter  $T_{id}$ . figure 4 presents time histories of the spray tip penetration from the onset of injection for different  $T_{id}$  in a constant-volume vessel. Spray tip penetration is increasingly reduced under a shorter injection duration  $T_{id}$ . figure 5 shows the spray tip penetration at 3 ms after the start of injection as a

function of the injection duration. It is observed that  $T_{id}$  of 2 ms or less has a large effect on reducing spray tip penetration. Setting  $T_{id}$  of each multiple injection at less than 2 ms is effective in achieving shorter penetration.

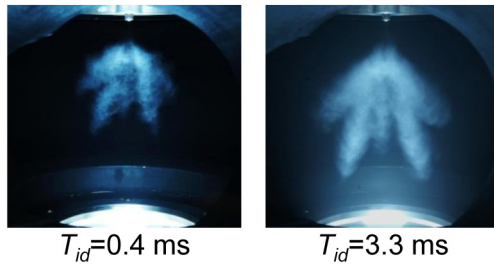


Fig. 3. Spray visualization at 2 ms after start of injection

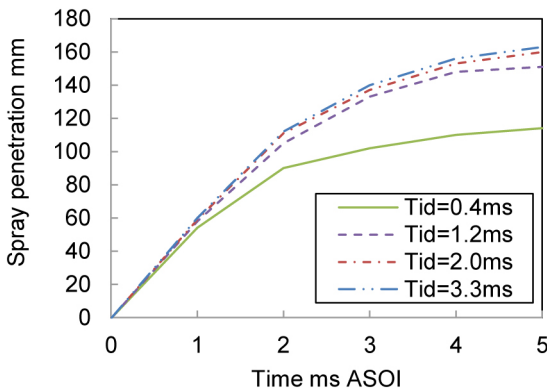


Fig. 4. Comparison of spray penetration

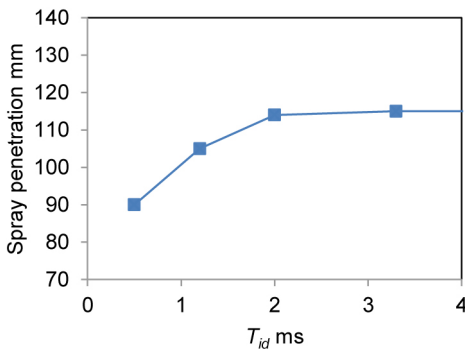


Fig. 5. Spray penetration under each injection duration at 2 ms after start of injection ( $P_{fuel}=12$  MPa)

A comparison was then made of the fuel film mass on the cylinder wall for multistage injections. figure 6 shows contours of the fuel film mass on the cylinder wall based on CFD spray calculations made for two injection patterns. The injection duration  $T_{id}$  of a single injection was 2.1 ms and that of triple injections was identical at 0.7 ms. The fuel injection amounts and fuel injection pressures under each injection pattern were the same. The results indicate that the effect of the multistage injection pattern on shortening spray tip penetration reduces the fuel film mass adhering to the bore wall and the piston crown. figure 7 compares the mass of liquid fuel film on the cylinder wall for a single injection and triple injections. The triple injection pattern reduces the wall film mass to around 40% of the single injection.

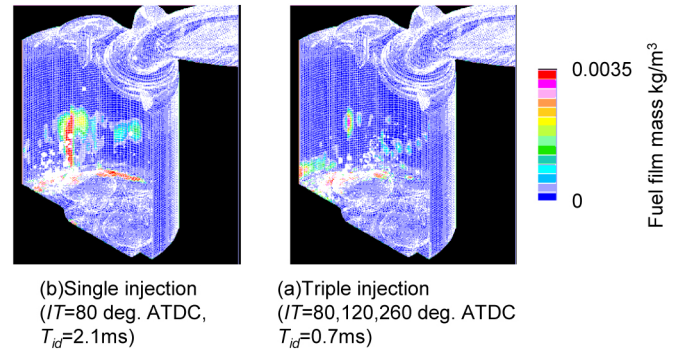


Fig. 6. Comparison of fuel wall film mass under different injection patterns ( $N_e=1200$  rpm, IMEP=9 bar,  $P_{fuel}=15$  MPa)

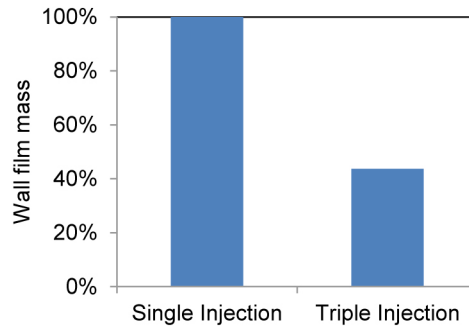


Fig. 7. Reduction of fuel film mass on cylinder wall

### Homogeneity of Air-Fuel Mixture

The improvement of air-fuel mixture formation by spray dispersion with a multistage injection pattern was investigated on the basis of LIF visualizations obtained with the optically accessible engine. figure 8 compares the air-fuel mixture distribution in the cylinder measured by LIF for single and multistage injection patterns. The air-fuel mixture shows better homogeneity for multistage injections than for the single injection. figure 9 shows the mixture homogeneity measured at the center timing of all injection durations for each injection pattern. The center timing of all injection durations was defined as shown in the equation below.

$$IT_c = \sum_{k=1}^n \left\{ \left( IT_k + \frac{T_{idk}}{2} \right) \frac{T_{idk}}{T_{id\_total}} \right\} \quad (1)$$

where  $n$  is the number of injections,  $IT_n$  is the injection timing of the  $n$ -th injection,  $T_{idn}$  is the injection duration of the  $n$ -th injection, and  $T_{id\_total}$  is the total duration of all injections.

The homogeneity index ( $HI$ ) was defined as shown in the equation below

$$HI = 1 - \frac{\sigma_\phi}{\phi} \quad (2)$$