

Research on Combustion Switching Control for Direct Injection Gasoline Engine

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ABSTRACT

Honda has developed its first-ever i-VTEC I direct injection gasoline engine employing a center injection system in which a high-pressure fuel injector is located at the center of the combustion chamber. Direct injection enables stratified charge combustion to be employed, and the new center injection engine demonstrates significantly improved fuel economy and emissions performance in the range of vehicle operation in which this type of combustion is used. The authors studied methods of switching between direct-injection stratified charge (DISC) and pre-mixed combustion in order to utilize DISC. The two forms of combustion employed in the new direct-injection engine differ in terms of combustion limits in relation to recirculated exhaust gas and air-fuel ratio. This causes the torque difference which is a specific issue in direct injection gasoline engines. The authors resolved this issue through the application of two technologies. Fuel injection during the intake stroke and the compression stroke has improved the combustion limit. When combustion is stable, the appropriate quantity of fuel to be supplied is determined from the required torque.

1. Introduction

Due to the increasing concern with the issue of global warming in recent years, it has been seen that the automotive industry has strived to develop high fuel economy and low emissions power plants. In particular, significant research and development efforts have been made to investigate various forms of direct injection gasoline engine as a means of improving the fuel economy of spark ignition internal combustion engines⁽¹⁾⁻⁽³⁾. Direct injection gasoline engines display low pumping loss and heat loss during DISC combustion. These characteristics result in high thermal efficiency, low intake air temperature during homogeneous charge combustion, superior resistance to knocking and a high charging efficiency. While direct injection gasoline engines possess a number of advantages, however, the

difficulty of controlling the mixture makes the achievement of stratified charge combustion across the entire operating range a challenge in these engines.

For this reason, direct injection engines employ both stratified charge combustion and homogeneous charge combustion depending on operating conditions⁽⁴⁾. At the same torque, the engine control parameters (spark timing, etc.) necessary for optimum fuel economy differ for these two forms of combustion. This makes a method of preventing torque fluctuations essential when switching between stratified charge and homogeneous charge combustion.

The research discussed in this paper attempted to develop a method of misfire prevention and a method of calculating the appropriate quantity of fuel to be injected to prevent the driver from experiencing any fluctuation in torque when a direct injection gasoline engine switches between combustion states.

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Table 1 Engine specifications

Engine code	K20B	
Cylinder configuration	In-line 4-cylinder	
Bore × Stroke (mm)	86 × 86	
Displacement (cm ³)	1 998	
Compression ratio	10:1	
Valve train	DOHC VTEC	
Number of valves	4 per cylinder	
Valve diameter (mm)	In.	34
	Ex.	28
Fuel supply system	Programmed fuel injection	
Fuel supply pressure (MPa)	8 - 10.5	
Fuel	Regular (91RON)	
Max. power (kW/rpm)	115 / 6 300	
Max. torque (N·m/rpm)	188 / 4 600	

2. Engine Specifications and System Configuration

2.1. Overview of Engine and Main Specifications

Table 1 shows the specifications of the test engine and Fig. 1 shows the configuration of the combustion chamber. The formation and maintenance of the mixture by the spray flow produced by the centrally positioned high-penetration injector and the deep bowl-type piston cavity, and the achievement of rapid combustion by the swirl induced by closing one valve by the VTEC mechanism, enable ultra-lean stratified charge combustion⁽⁵⁾.

2.2. System Configuration and Outline of Control Methods

Figure 2 shows the configuration of the test engine control system. The characteristics of this test system are as follows:

- (1) The system employs an air flow meter to enable accurate calculation of air intake during stratified charge combustion operation, etc., even when EGR is high; the required amount of fuel is calculated on the basis of the results for the amount of intake air.
- (2) An electronically controlled throttle valve able to converge on the target opening angle in a short period is used to enable employment of both stratified charge and homogeneous charge combustion to meet the demand for different amount of intake air.

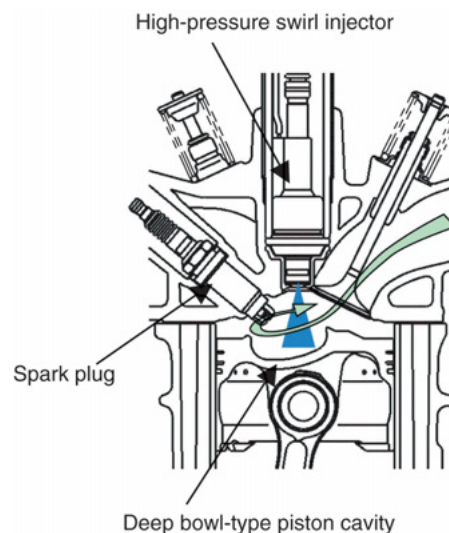


Fig. 1 Combustion chamber configuration

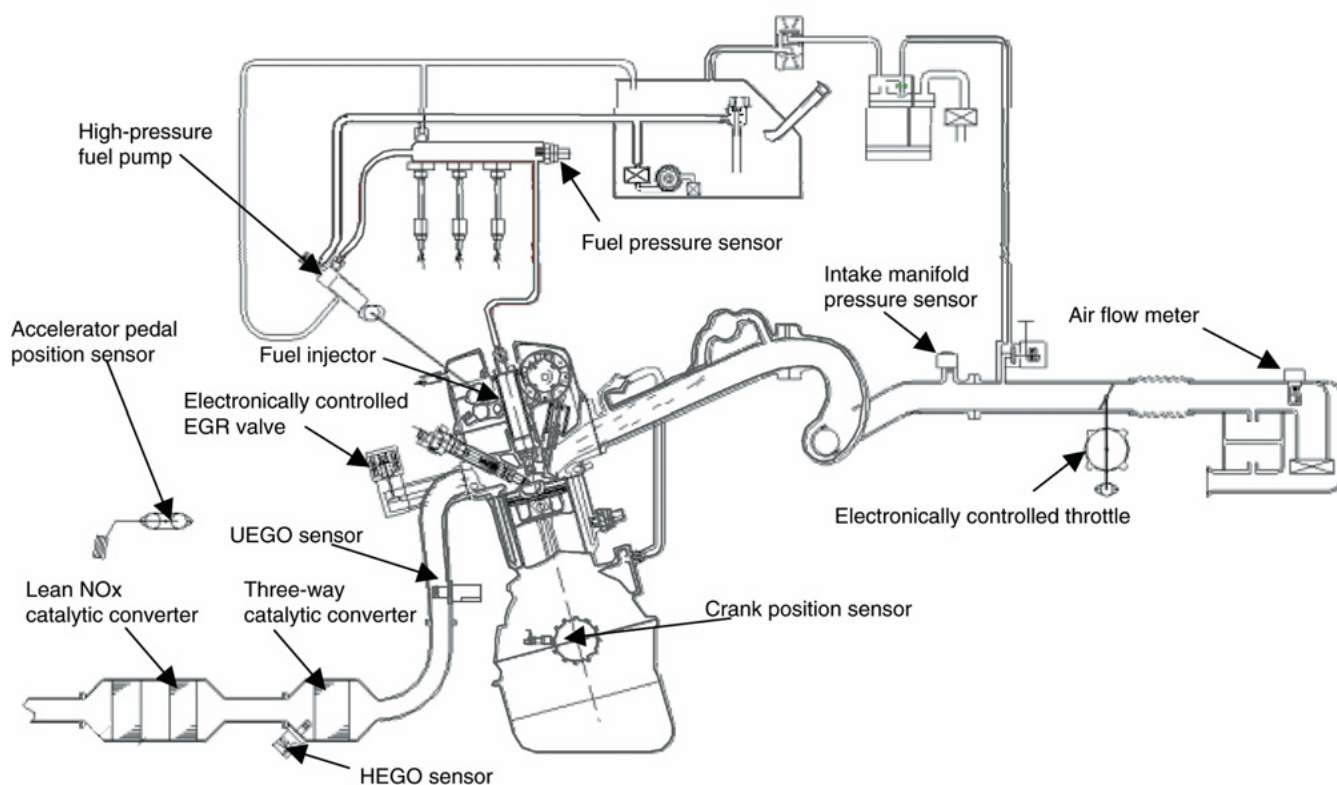


Fig. 2 System configuration

- (3) In a direct injection gasoline engine, the EGR ratio makes a significant contribution to drivability, exhaust emissions and fuel economy⁽⁶⁾. In this system, the widely used step motor EGR valve has therefore been replaced by a linear solenoid EGR valve which is five times more responsive (Fig. 3) in order to accelerate convergence on the required EGR ratio. Figure 4 compares the response times of the linear solenoid and step motor EGR valves.
- (4) The exhaust system is composed of –A universal exhaust gas oxygen (UEGO) sensor to enable air-fuel ratio feedback to be conducted during stratified charge operation in addition to homogeneous charge operation in order to maintain optimum drivability, fuel consumption and engine-out exhaust emissions; –A three-way catalyst for purification during operation at stoichiometric air-fuel ratios and HC purification during operation at lean air-fuel ratios; –A heated gas oxygen (HEGO) sensor to improve purification performance during stoichiometric operation; –A NO_x absorption reduction type three-way catalyst for NO_x purification during lean operation.
- (5) Fuel pressure control is provided by a spill valve which adjusts the amount of fuel supplied by the single-plunger high-pressure pump driven by the exhaust cam. The basic amount of control to be applied by the spill valve is calculated on a spill valve operating map from engine speed and the amount of fuel supplied by the injector, and modified on the basis of feedback from the output of a fuel pressure sensor positioned on the fuel delivery pipe⁽⁷⁾.
- (6) Because the required engine parameters differ in stratified charge combustion, it is impossible to decide engine load using amount of air and intake pipe pressure, which are widely used for this purpose in port injection systems.

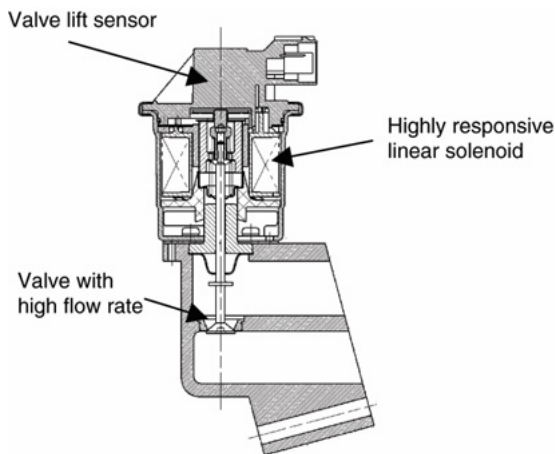


Fig. 3 Newly developed EGR valve

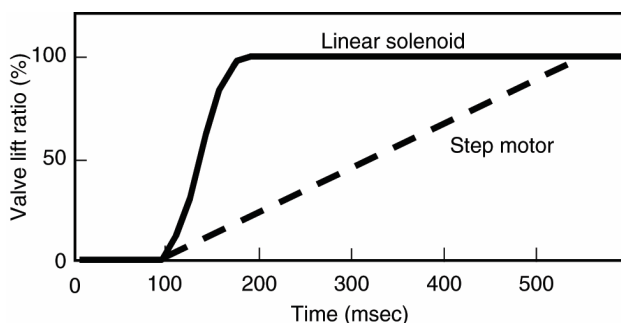


Fig. 4 Comparison of response times of linear solenoid EGR valve and step motor EGR valve

The new system employs an accelerator pedal sensor to enable the amount of torque demanded by the driver to be calculated, and control parameters (type of combustion, amount of EGR lift, etc.) are decided on the basis of the demanded torque and engine speed.

3. Combustion Switching Control

3.1. Why Combustion Switching Is Necessary

Figure 5 compares plots of brake specific fuel consumption (BSFC) against load in the test engine for DISC and homogeneous charge combustion at a stoichiometric air-fuel ratio. Significant fuel consumption improvements are offered by DISC in the low load area, but these benefits decline with increasing load, and above a certain load DISC does not offer any improvement over homogeneous charge combustion at a stoichiometric air-fuel ratio. This occurs because as load increases, the effect of improved pumping loss diminishes. It is therefore necessary to switch between DISC and homogeneous charge combustion according to engine load in order to optimize fuel consumption.

3.2. Combustion Switching Control: Targets and Issues

Fuel injection during both the intake and compression strokes is known as a method of forming weakly stratified mixture, preventing over-rich air-fuel ratios while ensuring stable combustion with lean mixture until a load has been reached at which lean combustion is possible for homogeneous charge⁽⁶⁾. This method has the benefit of ensuring stable combustion, but also makes it possible that operation at a lean air-fuel ratio will continue until a specified load is reached, creating concerns about an increase in exhaust emissions. The aim of this research project was to modify the operating parameters at combustion switching to ensure that there was no increase in emissions and no fluctuation in torque.

Figure 6 shows the behavior of control parameters at combustion switching. The figure indicates that high values are required for throttle opening and EGR lift during stratified charge combustion, while low values are required during homogeneous charge combustion. With the command for combustion switching, the command values for these parameters change, but there is a period of dead time during

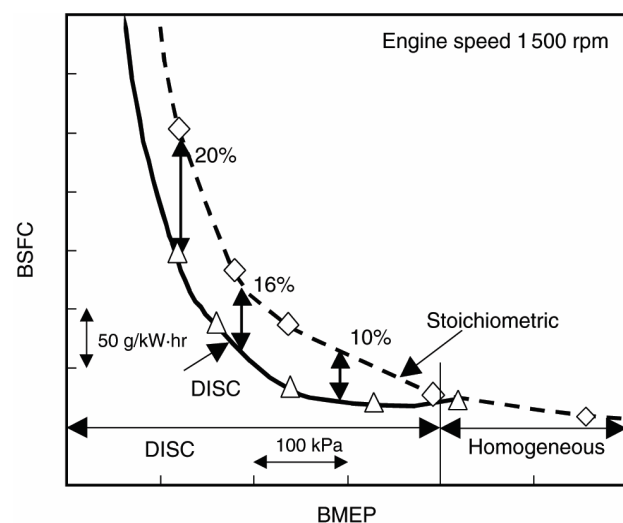


Fig. 5 Comparison of BSFC

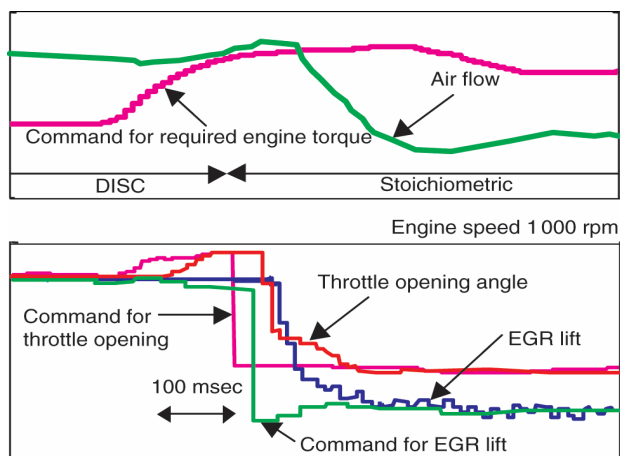


Fig. 6 Characteristics of engine control parameters during combustion switching

which actual throttle opening angle and actual EGR lift do not correspond to command values. The parameters show a first-order delay characteristic in reaching command values. This indicated that there were two challenges for combustion switching control in the period between the application of the command and the parameters converging on command values, when both EGR ratio and the amount of intake air are high: to prevent misfires during homogeneous charge combustion due to excess EGR, and to prevent a rapid generation of torque through the excess supply of fuel due to the supply of too much air.

3.3. Prevention of Misfire

To achieve equivalent torque to that supplied by stratified charge combustion during homogeneous charge combustion directly after switching, when the throttle valve opening and EGR lift retain the values they have for stratified charge combustion, the air-fuel ratio must be increased. However, if a high amount of EGR is introduced during homogeneous charge combustion at a lean air-fuel ratio, misfires occur.

A system of double injection, during the intake stroke and the compression stroke, is therefore employed to ensure that a combustible mixture is present around the spark plug at sparking even when a high EGR ratio is used. As Fig. 7

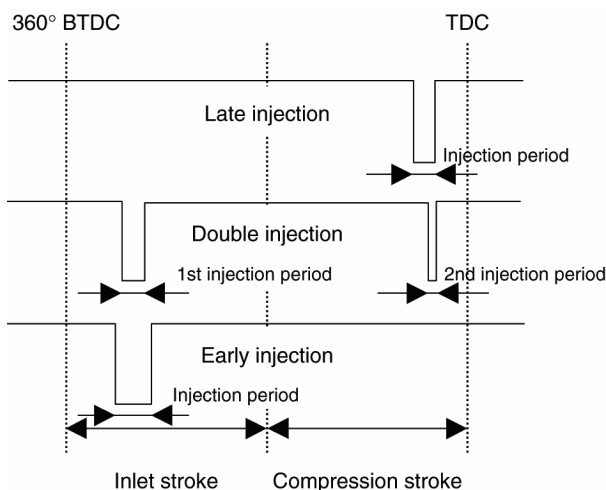


Fig. 7 Injection timing control

shows, injection timing control has three phases: compression stroke injection (late injection in the figure), in which fuel is injected late in the compression stroke for DISC combustion at low loads, intake stroke injection (early injection in the figure), in which fuel is injected early in the intake stroke for homogeneous mixture combustion at high loads, and double injection, in which fuel is injected both late in the compression stroke and early in the intake stroke to prevent misfires at combustion switching.

Figure 8 shows the comparison of combustion characteristic against EGR ratio at the same engine speed, load and air-fuel ratio for early injection and double injection. The employment of two injection timings has ensured stable homogeneous charge combustion with a low rate of fluctuation even at EGR ratios which cause misfires, and combustion is stable over a wider range of operating conditions. This indicates that late injection in the compression stroke ensures a combustible air-fuel ratio around the spark plug. Double injection is conducted until the response time for the EGR valve has elapsed and the equivalent ratio calculated from the amount of fuel required to achieve a stoichiometric air-fuel ratio (as calculated from the air intake) and the amount of fuel actually injected has become higher than a specified figure. This has ensured accurate misfire prevention.

3.4. Method to Calculate Fuel Amount for Injection

In this control system, the amount of fuel for injection is determined on the basis of air intake as measured by an air flow meter. This causes certain issues to arise at combustion switching. For example, when switching from stratified charge combustion to homogeneous charge combustion, the high amount of air which has been required for stratified charge combustion leads to the injection of an excess amount of fuel for the required torque until the throttle opening angle reaches the command angle. This causes torque to increase too rapidly. To prevent this, a limit is applied to the amount of fuel for injection as calculated on the basis of the amount of intake air. The limit value for the current amount of fuel required is calculated from the required torque and the amount of fuel for injection directly before combustion switching and the current required torque.

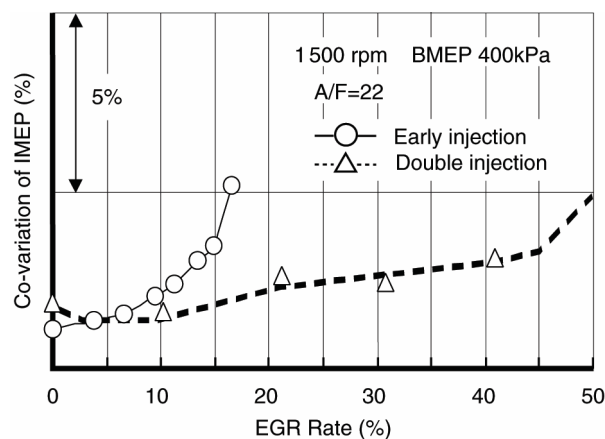


Fig. 8 Comparison of EGR characteristic for early injection and double injection

The application of this type of control requires an understanding of the relationship between torque and the amount of fuel injected. The measurement results, as shown in Fig. 9, indicated that the relationship between brake mean effective pressure (BMEP) and injection period could be expressed as a linear equation. The limit on the amount of required fuel is calculated using the equation below. The limit ceases to be applied when the limit value and the amount of fuel for injection calculated on the basis of the amount of air intake become equivalent, preventing sudden changes in the amount of fuel injected. The equation for calculation of the fuel limit value is as follows:

$$T_{lim} = \{T_{in} + (\Delta T_{req} / \Delta P_m) \times (P_m - P_{min})\} \times (1 \pm K_{cmb})$$

- T_{lim} : Limit value for amount of fuel for injection
 $\Delta T_{req} / \Delta P_m$: Amount of change in injection period against BMEP
 T_{in} : Required injection period at moment of switching
 P_m : Current torque demand from driver
 P_{min} : Torque demand from driver at moment of switching
 K_{cmb} : Correction for combustion efficiency

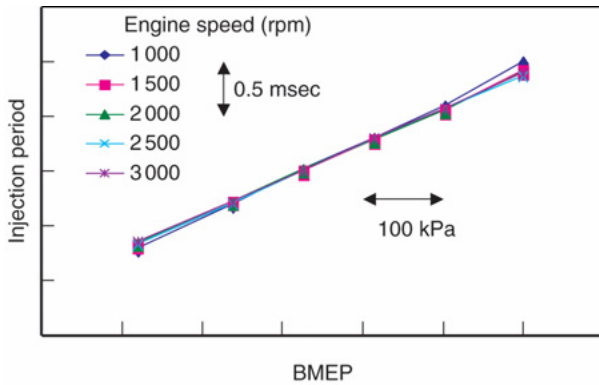


Fig. 9 Relationship of BMEP and injection period

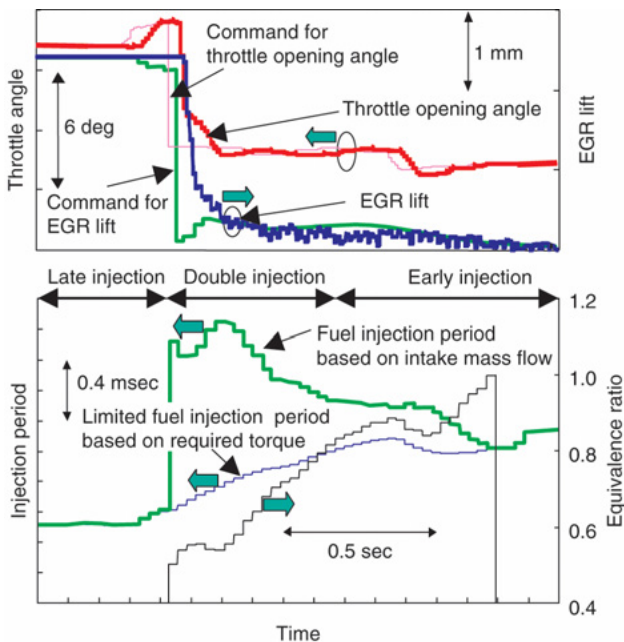


Fig. 10 Schematic diagram of transient condition

4. Engine Test Results

Figure 10 shows the behavior of engine control parameters when switching control is applied.

Double injection is no longer applied when a specified equivalent ratio is reached after EGR lift converges on the command value.

The results in the figure also indicate that the limit applied to the amount of fuel for injection has prevented any sudden increases in fuel supply, and has ensured that fuel supply changes smoothly, and in accordance with requirements based on the amount of air intake.

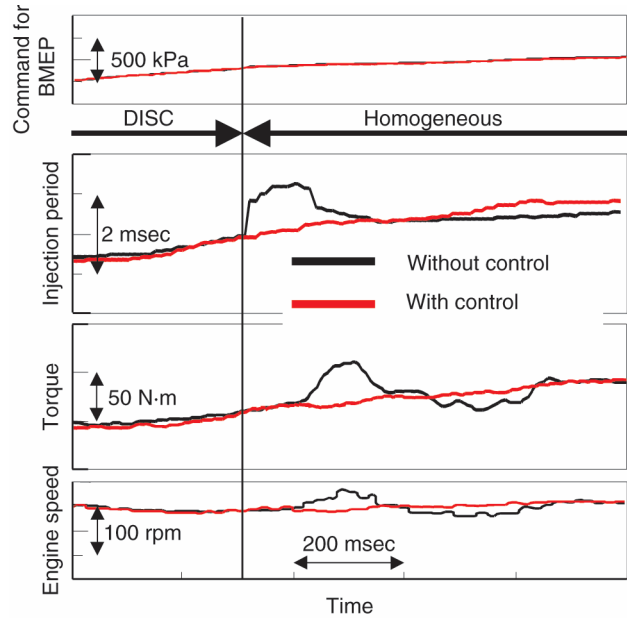


Fig. 11 Effect of combustion switching control on transient torque behavior

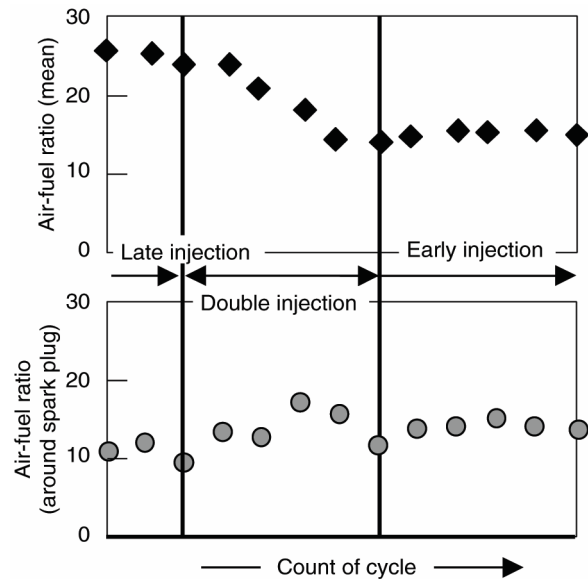


Fig. 12 Characteristics of air-fuel ratio around spark plug

Figure 11 demonstrates the effect of combustion switching control by means of changes in torque behavior. The use of combustion switching control has eliminated sudden changes in torque and in engine speed caused by the change of the combustion mode, and has ensured that combustion switching is carried out without shock and smoothly follows changes in torque demand.

Figure 12 shows the measurement results of air-fuel ratio around the plug and mean air-fuel ratio per cycle, using infrared absorption before and after switching. It is clear that the air-fuel ratio around the plug is controlled to a combustible level when combustion mode is switched, and that double injection in the intake and compression strokes has been effective in preventing misfiring. In addition, a combustible mixture is maintained around the spark plug during stratified charge combustion even when the mean air-fuel ratio is lean, and during homogeneous charge combustion the mean air-fuel ratio and the air-fuel ratio around the spark plug are equivalent.

5. Conclusions

Various control methods have been explored in this work to ensure smooth transition from stratified charge combustion to homogeneous charge combustion. The major conclusions can be summarized as follows:

- (1) The employment of double injection in the intake and compression strokes prevents misfires during combustion switching when the EGR ratio is still high.
- (2) The calculation of a limit value for the amount of fuel supplied based on the required torque and the amount of fuel supplied directly before combustion switching and the current required torque, and the employment of this value as a control parameter, prevents sudden changes in the amount of fuel supplied during combustion switching.
- (3) A control system composed of early, late and double injection timing control and fuel supply limit control enables smooth combustion switching in a direct injection gasoline engine with no negative effects on emissions performance.

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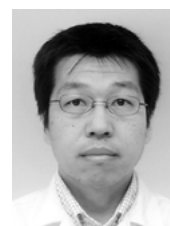
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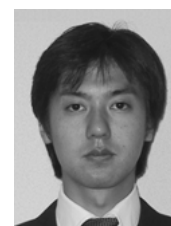
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