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Effects of operating parameters on mode transition between low-temperature combustion and conventional combustion in a light-duty diesel engine

Sangwook Han, Choongsik Bae and Seibum B Choi

Abstract

An experimental study on the mode transition between low-temperature combustion and conventional combustion was carried out in a light-duty diesel engine. The characteristics of combustion mode transition with various operating parameters, including rate of exhaust gas recirculation change, residual gas, exhaust gas recirculation path length, fuel injection pressure and engine speed, were analysed based on the in-cylinder pressure and hydrocarbon emission of each cycle. In the case of mode transition from low-temperature combustion to conventional combustion, rapid decreases in indicated mean effective pressure and hydrocarbon emission occurred due to the improper injection timing and the decrease of the exhaust gas recirculation rate. On the other hand, indicated mean effective pressure and hydrocarbon emission changed slowly during mode transition from conventional combustion to low-temperature combustion owing to the thermal effect of hot residual gas from conventional combustion. Faster mode transition could be achieved by the use of a shorter exhaust gas recirculation path. Although the trends of mode transition in terms of indicated mean effective pressure were similar, the noise levels, as represented by the maximum pressure rise rate, and hydrocarbon emissions were significantly affected by residual gas, fuel injection pressure and engine speed. In addition, smooth combustion mode transition could be achieved by cycle-by-cycle injection modulation without rapid changes of indicated mean effective pressure and maximum pressure rise rate.

Keywords

Diesel engine, exhaust gas recirculation, low-temperature combustion, conventional combustion, mode transition

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Introduction

For the past decades, diesel engines have been widely used for transportation and power-generation applications because of their high efficiency. However, diesel engines cause environmental pollution owing to their high nitrogen oxide (NOx) and soot emissions. Considerable effort has been devoted toward reducing pollutant emissions, especially NOx and soot, as these have adverse effects on the environment and on human health. Revolutionary in-cylinder combustion strategies and exhaust emission aftertreatment devices are required to meet stringent emission regulations. Emission aftertreatment devices, however, have some problems in terms of cost and durability. In-cylinder technology has therefore been the focus of intense investigation.

Diesel combustion can be controlled by the amount of exhaust gas recirculation (EGR), as inert gases with increased heat capacity can dilute the intake charge.[1–3] EGR induces low flame temperature and affects combustion and emissions. Traditionally, low combustion temperatures are thought to reduce NOx emissions but increase particulate emissions due to a reduction in particulate oxidation rates. However, as EGR rates increase to very high levels (approximately 60% or

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greater), NOx–soot trade-off is broken by achieving local temperatures below 1500 K, at which the formation of NOx and soot are suppressed.[4–7] This combustion strategy, which uses a high rate of EGR, has been called low-temperature combustion (LTC). LTC has demonstrated strong potential to reduce emissions of both NOx and soot to acceptable levels for future emission regulations. However, LTC results in increased fuel consumption and emissions of carbon monoxide (CO) and hydrocarbons (HCs) because of the low combustion temperature. CO and HC emissions are products of incomplete combustion originating from the lack of oxidation reaction during the expansion stroke. A low combustion temperature also limits the operating range at low to medium load. Many studies have been conducted to overcome these problems with injection strategies and intake charge boosting. Multiple injection strategies under LTC conditions have shown the potential to reduce CO and HC emissions, which can be attributed to the reduction of an over-lean mixture near the nozzle tip after the end of injection and fuel entering the squish region.[8,9] The increase in intake pressure provided by systems like turbochargers and superchargers reduces CO and HC emissions as well as the specific fuel consumption without any deterioration of NOx and soot emissions in LTC. In addition to these benefits, the operating range of LTC could also be extended by intake charge boosting.[10,11] However, the extended operating range of LTC is still too narrow to cover the entire operating range of diesel engines. This feature is an obstacle for its practical application for vehicles. Therefore, research on the mode transition between LTC and conventional combustion is required. It is based on the concepts to derive benefits from LTC at low to medium load and switch to conventional combustion at higher load. Most research on combustion mode transition has focused on gasoline engines. Mode transition between spark ignition (SI) and controlled auto ignition (CAI) is mostly realized by controlling the amount of hot residual gas using variable valve actuation (VVA).[12–14] However, research related to the combustion mode transition in diesel engines, which requires the control of EGR and fuel injection, is insufficient to understand the overall characteristics of combustion mode transition. One study concerning the diesel engine investigated engine-out emissions during mode transition between early premixed charge compression ignition (PCCI) and conventional combustion.[15] This study reported that differences in the time responses of the fuelling and air handling systems caused spikes in power output and exhaust emissions during mode transition. Another study associated with the diesel engine focused on the spikes in NOx and smoke emissions generated in the transition between premixed compression ignition (PCI) and conventional combustion and tried to minimize them via an appropriate control of fuel injection and air management.[16] However, most studies performed up to now have not dealt with the effects of various operating parameters on the combustion mode transition.

A parametric study on the combustion mode transition in diesel engines is considered indispensable for better understanding of combustion mode transition and hence the practical application of LTC for vehicles. Therefore, the first part of this article studies characteristics of mode transition between LTC and conventional combustion with various operating parameters including rate of EGR change, residual gas, EGR path length, fuel injection pressure and engine speed. It was performed by changing the EGR rate in a light-duty diesel engine. The effects of each operating parameter on the variations of power output, combustion phase, combustion noise and HC emissions during mode transition were then investigated. Furthermore, the combustion mode transition should proceed smoothly so that the vehicle passengers do not experience any shock and noise. To this end, power output variations and noise levels should be acceptable during mode transition. Accordingly, transient injection strategies were proposed for smooth combustion mode transition in the latter part of this study.

Experiments

Experimental setup

A four-stroke, five-cylinder, direct-injection diesel engine was modified to operate in single-cylinder mode and was used for the investigation. A schematic diagram of the research engine system is shown in Figure 1, and its specifications are shown in Table 1. Engine speed was controlled by an eddy current (EC) dynamometer. The four cylinders (nos 1–4) were operated by a programmable electronic control unit (ECU) in the conventional diesel combustion mode at low load for stable engine operation. The mode transition was performed in cylinder no. 5 by independently controlling the fuel injection parameters and EGR rate. The concepts of the control system for mode transition are shown in Figure 2. Fuel injection parameters, including injection timing, injection duration and number of injections, were controlled for the common-rail fuel system by a programmable injector controller (Zenobalti Co., ZB-8010). The EGR rate was controlled by two valves positioned downstream of the EGR cooler (valve 1) and the intake air line (valve 2), both of which were controlled by a valve controller (Zenobalti Co., ZB-2200). Each valve actuated by the electric motor adjusted the flow of the EGR and intake air by controlling the valve opening/closing degree. The performance of the electric motor enables a change of valve from wide open state to closed state within one engine cycle. These two controllers were synchronized by counting the cycle number. This allowed all of the input variables to be controlled on a cycle-by-cycle basis. To create a condition for investigating the effect of residual gas, two valves were installed downstream of the EGR cooler (valve 1) and

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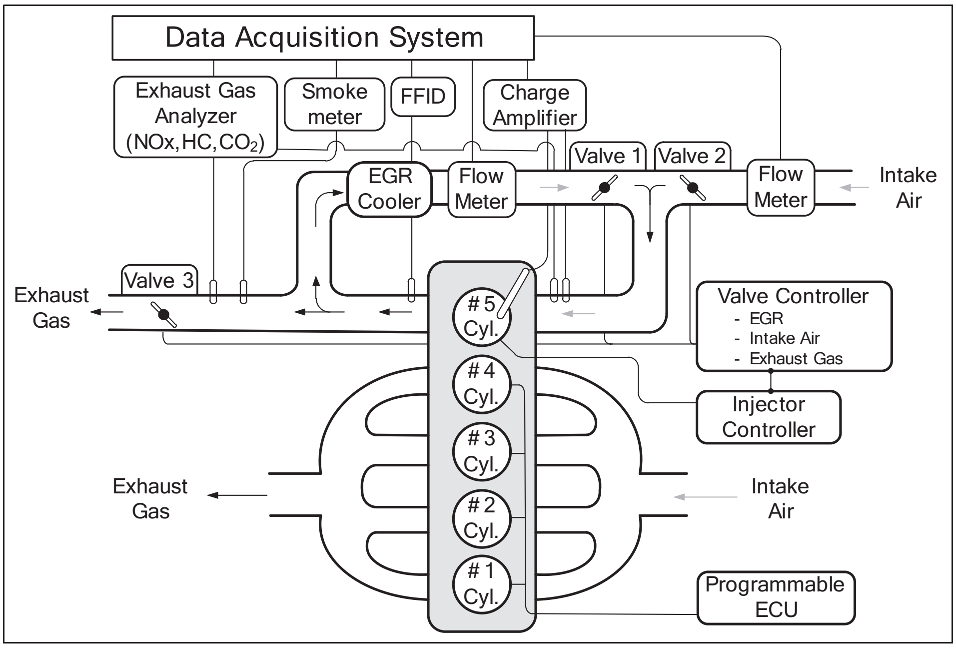

Figure 1. Schematic diagram of the research engine system.

Table 1. Specifications of the research engine.

Engine type	Five cylinders, direct injection, four valves, diesel engine
Bore × stroke	86.2 × 92.4 mm
Displacement per cylinder	539 cm^3
Compression ratio	17.5
Fuel injection equipment	Common-rail injection system

downstream of the exhaust manifold (valve 3) while the intake air line was wide open. A large EGR cooler was installed to supply a large amount of cooled EGR for the reduction of the ambient temperature. Two flow meters were employed to measure the mass flow rates of intake air and EGR, which were used to calculate the EGR rate during mode transition.

The in-cylinder pressure was measured using a piezoelectric pressure transducer (Kistler 6056A), and the intake pressure was measured by a piezoresistive pressure transducer (Kistler 4045A5). All of the measured data were acquired with a resolution of 0.2 crank angle degree (CAD) over 500 engine cycles. The exhaust gases in steady-state conditions were analysed using a gas analyser (Horiba, MEXA-1500d) to measure NOx, HC and carbon dioxide (CO_2) in the exhaust and CO_2 in the intake. A smoke meter (AVL, 415S) was employed for steady-state soot measurement, expressed in filter smoke number (FSN) units, which uses the filter paper method. Primarily the graphitic carbon in the exhaust gas is measured by this smoke meter. The cycle-by-cycle HC measurements during mode transition were made using a fast-response flame ionization detector (FFID, Cambustion, HFR 400) equipped with a constant-pressure system and a heated sampling line. The probe

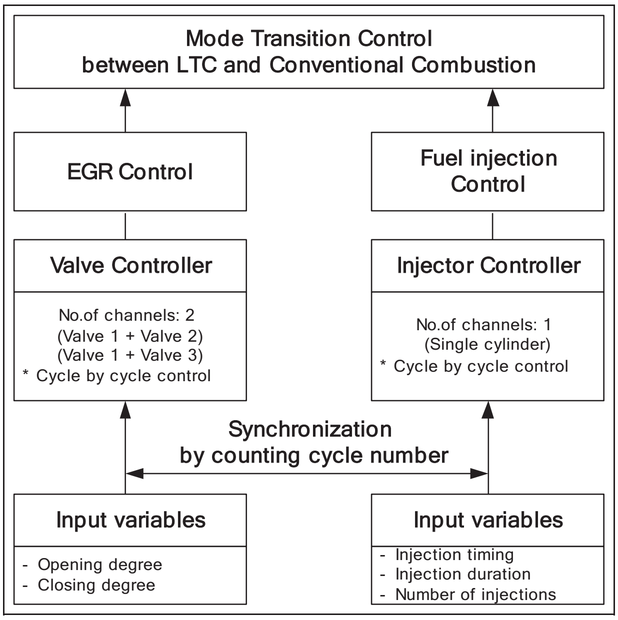

Figure 2. Concepts of the mode transition control system (valves 1, 2 and 3 are shown in Figure 1).

was positioned in the exhaust port 30 mm downstream from the exhaust valve stem. The FFID had a response time of roughly 0.5 ms (3.5 CAD at 1200 r/min) to changes in HC levels, allowing cycle-resolved HC measurements from the exhaust stream. The mean value of the HC for each cycle was taken as the cycle-averaged HC emission for that cycle.

Experimental conditions

Table 2 shows the experimental conditions used in the mode transition test. The combustion mode was divided

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Table 2. Experimental conditions for mode transition.

Operating condition	A (baseline)	B	C	D	E	F
Initial IMEP (MPa)	0.45	0.45	0.45	0.45	0.45	0.45
Engine speed (r/min)	1200	1200	1200	1200	1200	2000
Injection pressure (MPa)	40	40	40	40	120	40
EGR control valve position (1: Intake + EGR line) (2: Exhaust + EGR line)	1	1	1	2	1	1
EGR path length (m)	6.1	6.1	3.8	6.1	6.1	6.1
EGR rate (%)	60 → 0	0 → 60	60 → 0	60 → 0	60 → 0	60 → 0

into two modes represented by the EGR rate. One is the LTC mode, which uses a 60% EGR rate, and the other is the conventional combustion mode, which uses no EGR. These criteria were derived from several studies into LTC.[6,11,17] The EGR rate in the steady-state experiment was defined as the ratio between the CO_2 concentration of the intake air, $X^{in}_{CO_2}$, and the CO_2 concentration of the exhaust gas, $X^{ex}_{CO_2}$

$$\text{EGR rate} = \frac{X^{in}_{CO_2}}{X^{ex}_{CO_2}} \times 100(\%) \tag{1}$$

On the other hand, the EGR rate during mode transition (EGR rate_{MT}) was calculated from the mass flow rate of intake air, $\bar{m}_{\text{Intake air}}$, and the mass flow rate of EGR, $\bar{m}_{EGR}$.[18]

$$\text{EGR rate}_{MT} = \frac{\bar{m}_{EGR}}{\bar{m}_{EGR} + \bar{m}_{\text{Intake air}}} \times 100(\%) \tag{2}$$

The indicated mean effective pressures (IMEPs) in each combustion mode were set to 0.45 MPa before mode transition, representing the maximum load of LTC under naturally aspirated conditions in this research engine. Test conditions were divided into six points according to various operating parameters. Baseline condition A, which features a mode transition from LTC to conventional combustion at the engine speed of 1200 r/min, fuel injection pressure of 40 MPa, EGR path length of 6.1 m and EGR control valves positioned at each EGR and intake line (valves 1 and 2), was compared with the other conditions. In order to investigate the effect of the rate of EGR change on mode transition characteristics, the EGR rate was changed from 0% to 60% in operating condition B. In addition, a different EGR path length of 3.8 m, defined as the length of the EGR line from exhaust manifold to intake manifold, was adopted to understand the effect of the EGR path length in operating condition C. In operating condition D, the positions of EGR control valves were changed from the EGR and intake lines (valves 1 and 2) to the EGR line and downstream of the exhaust manifold (valves 1 and 3) to investigate the effect of residual gas. The effect of fuel injection pressure was explored by evaluating a fuel injection pressure of 120 MPa in operating condition E. Furthermore, an engine speed of 2000 r/min was selected to verify the effect of engine speed in operating condition F. Although the amount of EGR was changed during mode transition, the temperature in the intake manifold during mode transition was kept at 313 K in every test condition. This was because the EGR was sufficiently cooled down to the ambient temperature through a large EGR cooler.

Experimental procedure

In the first part of this study, only changing the EGR rate was used as a way to perform the combustion mode transition. Basically, the combustion mode transition was carried out by changing the EGR rate from 60% to 0% or vice versa in the process of steady-state operation in each combustion mode. The initial operating conditions before mode transition were determined through steady-state experiments. The optimum injection conditions with an IMEP of 0.45 MPa as well as the lowest exhaust emissions were selected through the sweep of injection timings in steady-state experiments for each combustion mode. These chosen injection conditions were used as initial operating conditions in each mode transition test. To guarantee the stable operation of each combustion mode, initial conditions were maintained during 1000 cycles. Subsequently, the EGR rate was changed while injection conditions were maintained constant as initial conditions. In the latter part of the study, mode transition was carried out using cycle-by-cycle injection modulation as well as the change of EGR rate.

Results and discussion

Decision of optimum injection conditions in each combustion mode

The exhaust emission characteristics according to the injection timings were investigated to determine the initial operating conditions of mode transition tests. The exhaust emissions measured for an injection timing sweep are shown in Figure 3. The results shown in Figure 3 from the 60% EGR rate, engine speed of 1200 r/min and fuel injection pressure of 40 MPa condition indicate that NOx and soot emissions were very low, regardless of injection timing. This is because the

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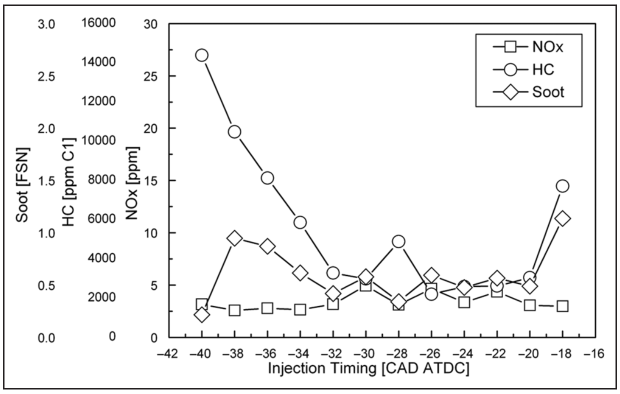

Figure 3. Effects of injection timing on exhaust emissions (IMEP = 0.45 MPa, engine speed = 1200 r/min, injection pressure = 40 MPa, EGR rate = 60%).

low combustion temperature achieved by the high EGR rate suppressed the formation of NOx and soot, irrespective of injection timing. HC emission was minimized at the injection timing of −26 CAD after top dead centre (ATDC). HC emission increased as injection timing was advanced from −26 CAD ATDC, while it increased as injection timing was retarded from that timing. The increase in HC emission with advanced injection timing could be due to the increased amount of liquid fuel that missed the piston bowl.[18] On the other hand, the increased HC emission with retarded injection timing could be owing to reduced mixing and vaporization resulting from shorter ignition delays. Therefore, the injection timing of −26 CAD ATDC was selected as the optimum injection timing in this condition. The optimum injection timings in other operating conditions were determined along the same lines as the previous procedure. The optimum injection timing for the condition of no EGR was retarded near top dead centre (TDC) due to the reduced ignition delay caused by the absence of a dilution effect of EGR. Increased fuel injection pressure also retarded the optimum injection timing due to the improvement of air/fuel mixing by better spray atomization and in turn produced a shorter ignition delay. Meanwhile, optimum injection timing at higher engine speed was advanced to increase the time for premixing. The selected optimum injection conditions are summarized in Table 3.

Table 3. Optimum injection conditions for each operating condition.

Operating condition	Injection pressure (MPa)	Injection duration (μs)	Injection timing (CAD ATDC)
A, C, D (60% EGR rate)	40	860	−26
B (0% EGR rate)	40	817	−4
E (60% EGR rate)	120	547	−14
F (60% EGR rate)	40	872	−36

Effects of rate of EGR change on combustion mode transition

Mode transition between LTC and conventional combustion could be achieved by changing the EGR rate, as shown in Figure 4. NOx and soot emissions (which were measured in steady-state experiments) before and after mode transitions are shown in Figure 4(g). This shows typical characteristics of LTC characterized by low NOx and soot emissions at the EGR rate of 60%. In particular, NOx emissions at the EGR rate of 60% were too low, making the bar in the figure almost nonexistent.

Firstly, mode transition from LTC to conventional combustion was performed by decreasing the EGR rate from 60% to 0% while maintaining the optimized injection conditions as for the 60% EGR rate. Cycle number 0 indicates the cycle at which the movement of the EGR control valves was initiated. Therefore, the negative number cycles were operated in a steady-state condition that represented the condition before mode transition. The EGR $rate_{MT}$, which was calculated from mass flow rates of intake air and EGR measured by flow meters installed far upstream of the intake manifold, was changed immediately after cycle number 0, as shown in Figure 4(a). However, the delay in EGR delivery existed in the early stage of mode transition (approximately 10 cycles) because some time was needed for the gas to travel through the EGR pipe due to the long EGR path length. In this period, no changes were observed due to the unchanged EGR rate in IMEP, in-cylinder maximum pressure (in-cylinder P_{max}), crank angle position where 50% of the total heat release occurs (CA50) and HC emission. After this stage, the EGR rate decreased rapidly until the 20th cycle. This resulted in advancing the combustion phase as represented by CA50, which caused an increase in negative work, and increasing the combustion rate, which allowed an increase in the in-cylinder P_{max}. The combustion phase and in-cylinder P_{max} are the most important parameters that affect the IMEP. IMEP increased in this period because the increase in the in-cylinder P_{max} had a greater influence on IMEP despite the advanced combustion phase. During the following 30 cycles, IMEP decreased rapidly due to a slight advance in the combustion phase caused by hotter residual gas as the combustion mode was switched to conventional combustion. IMEP was then stabilized to a lower level than that of the LTC condition due to the improper injection timing in the conventional combustion. HC emission was maintained until the 10th cycle owing to the EGR delay and decreased rapidly due to the higher combustion temperature caused by reduced EGR rate as mode transition proceeded.

The mode transition from conventional combustion to LTC was also investigated. Rapid changes in IMEP and HC emission, however, were not observed despite the increased EGR rate. This can be explained by the in-cylinder thermal conditions during the mode

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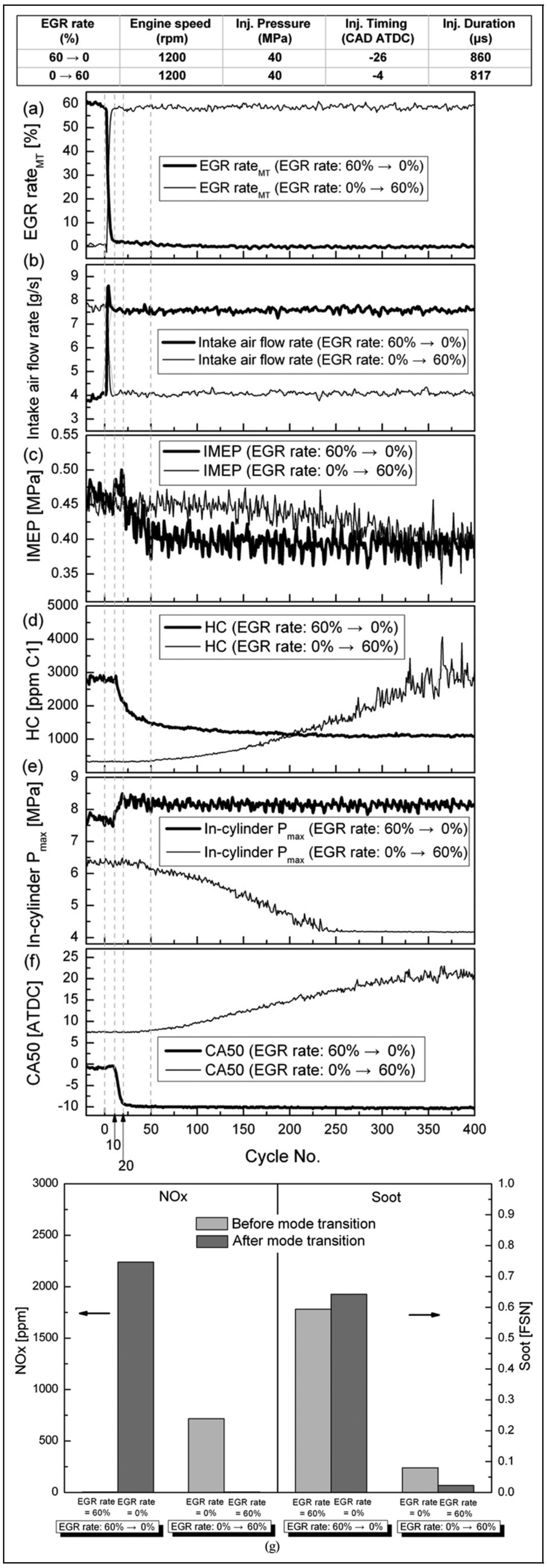

Figure 4. Effects of rate of EGR change on combustion mode transition: (a) EGR rate; (b) intake air flow rate; (c) IMEP; (d) HC emission; (e) in-cylinder P_{max}; (f) crank angle position where 50% of the total heat release occurs (CA50); and (g) NOx and soot emissions before and after mode transition.

transition. Because the residual gas is transferred to the next cycle, the thermal energy of residual gas influences the initial temperature of in-cylinder charge in the following cycle and thus the subsequent combustion process. The burned gas temperature from conventional combustion is higher than that from LTC. The hotter residual gas from conventional combustion made the initial in-cylinder charge temperature higher and thus prevented not only a rapid decrease in the peak cylinder pressure, but also drastic retard in the combustion phase. Therefore, the IMEP decreased slowly, and HC emission increased gradually without a sudden change.

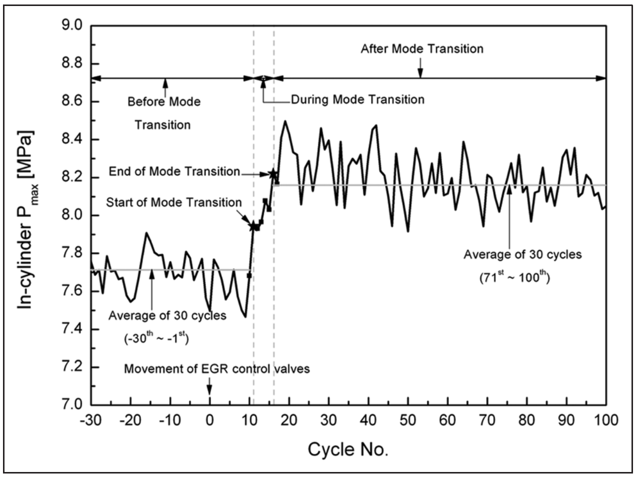

Figure 5. Definition of mode transition interval (baseline condition A (specified in Table 2)).

Effects of EGR path length on combustion mode transition

From this section, the mode transition interval is defined for better understanding, as can be seen in Figure 5. The variation of in-cylinder P_{max} was selected as a criterion for dividing the combustion mode because it is believed to be most sensitive to the change of in-cylinder charge composition. The start of mode transition was defined as the cycle at which in-cylinder P_{max} just exceeds the end tip of the average line that depicts the average of the in-cylinder P_{max} during 30 cycles from -30 to -1. The end of mode transition was defined in the same way with different cycle numbers from 71 to 100.

In the above experiment, the EGR path length from exhaust manifold to intake manifold was found to be too long to achieve fast mode transition. Therefore, the combustion mode transition with shorter EGR path length was tested for the purpose of reduction of the EGR delay. Figure 6 compares the mode transitions from LTC to conventional combustion with different EGR path lengths of 6.1 m and 3.8 m. The volumes of the EGR pipes were 12.2 times (EGR path length of 6.1 m) and 7.6 times (EGR path length of 3.8 m) the engine displacement volume, respectively. EGR control strategies were the same in both conditions, with different EGR path lengths. It can be seen that mode

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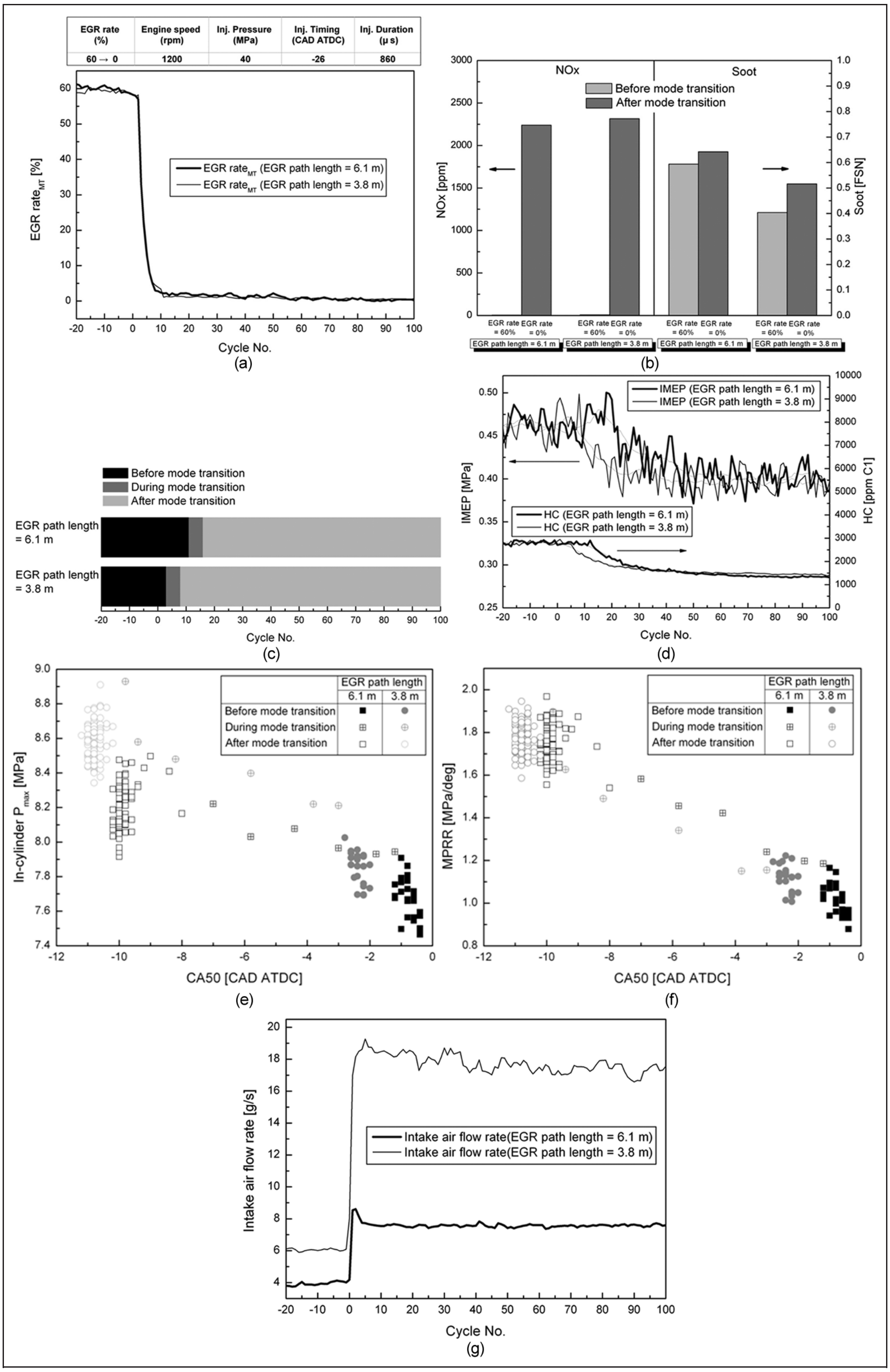

Figure 6. Effects of EGR path length on combustion mode transition: (a) EGR rate; (b) NOx and soot emissions before and after mode transition; (c) classification of combustion mode; (d) IMEP and HC emission; (e) in-cylinder P_{max} versus CA50; (f) MPRR versus CA50; and (g) intake air flow rate.

transition with the shorter EGR path occurred earlier because the time for the gas traveling through the EGR pipe was reduced, as shown in Figure 6(c). The shorter EGR path reduced the pressure drop, which results from friction between the gas and the pipe, resulting in increased volumetric efficiency. It allowed more intake air flow into the cylinder, as shown in Figure 6(g). However, the EGR rates were kept the same in both

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test conditions, as can be seen in Figure 6(a). This was because the EGR flow rate into the cylinder was also increased due to reduced pressure drop. Therefore, the increased intake air flow advanced the combustion phase and increased in-cylinder P_{max}, as shown in Figure 6(e). Maximum pressure rise rates (MPRRs) in both cases, on the other hand, show similar levels in Figure 6(f). Although the combustion phase and in-cylinder P_{max} were totally different in both cases, the IMEPs show similar levels due to the compensation of the above two contradicting effects. HC emission also shows the faster transition with a shorter EGR path.

Effects of residual gas on combustion mode transition

In-cylinder mixture conditions such as composition and temperature have significant effects on the combustion process and exhaust emissions. The EGR gas, consisting of the external EGR and the internal EGR, i.e. residual gas, which can be adjusted by the EGR control system has a strong influence on in-cylinder mixture conditions.

In order to investigate the effect of residual gas on mode transition from LTC to conventional combustion, two kinds of EGR control systems were applied. In one system, the EGR rate was controlled by two EGR control valves positioned downstream of the EGR cooler (valve 1) and the intake air line (valve 2). In the other system, EGR control valves were installed downstream of the EGR cooler (valve 1) and downstream of the exhaust manifold (valve 3) to control the exhaust back pressure and thus adjust the EGR rate (refer to Figure 1). In the latter system, the intake air side maintained a wide open condition. The change of exhaust back pressure, which affects the residual gas concentration and hence the composition and temperature of the in-cylinder mixture, has the potential to influence the combustion process.

Figure 7 shows a comparison of the mode transition characteristics between two EGR control systems. Although the difference of EGR rate between the two EGR control systems was negligible, as shown in Figure 7(a), the difference in the amount of residual gas was substantial due to the different exhaust back pressures. Thus, the effective EGR rate (external EGR rate + internal EGR rate) was higher in the system that controlled the exhaust back pressure.[19] This phenomenon appears noticeably at the higher EGR rate condition due to the higher exhaust back pressure, while it diminishes at the lower EGR rate condition because of the reduced exhaust back pressure. Generally, the increase in residual gas provides a greater charge dilution effect and increase of charge temperature.

The higher effective EGR rate in the exhaust back pressure control system led to a decrease in the in-cylinder P_{max} and the MPRR and retard in the combustion phase before and during mode transition. This indicates that the charge dilution effect had a greater influence on the combustion process than the thermal effect of hot residual gas. The combination of the decrease in the in-cylinder P_{max} and retard in the combustion phase had contradictory effects on the IMEP, and thus IMEP showed similar levels in both cases. However, the differences in the in-cylinder P_{max}, MPRR and CA50 between the two conditions were reduced as the mode transition proceeded since the difference in the effective EGR rate between each control system was gradually reduced. HC emission for the exhaust back pressure control system under LTC was higher due to the increase in charge dilution. In addition, the difference of HC emissions between the two cases was reduced as the mode transition proceeded because of the reduced dilution effect.

Effects of fuel injection pressure on combustion mode transition

The air–fuel mixing condition, which is significantly affected by fuel injection pressure, is an important factor in the combustion process. Figure 8 shows the mode transition process from LTC to conventional combustion at two different fuel injection pressures, 40 MPa and 120 MPa. There are no significant differences between IMEP variation trends of two fuel injection pressures. The reason for this is that the IMEP variation trend is governed mainly by the rate of EGR change, which was exactly the same in both cases. With regard to HC emissions, it shows clear differences between the two fuel injection pressures. HC emissions were lower at the higher fuel injection pressure. This is attributed to the fact that the higher fuel injection pressure improves the atomization of injected fuel and enhances evaporation of the fuel droplets and the mixing process with the ambient air, which are favourable for complete combustion.[20]

It is noted that differences in the MPRR and the in-cylinder P_{max} between the two fuel injection pressures show opposite trends along the mode transition process. Before and during the mode transition, in-cylinder P_{max} at the fuel injection pressure of 120 MPa was lower, while the level of MPRR was almost the same for both fuel injection pressures. Even though the combustion phase was retarded by EGR during this period, the amount of burned fuel before TDC was larger in the case of 40 MPa due to earlier injection timing, as can be seen in Figure 8(g). This resulted in higher in-cylinder P_{max} at the fuel injection pressure of 40 MPa. The reason for the similar level of MPRR in both cases was that CA50 at the fuel injection pressure of 120 MPa occurred after TDC due to relatively late injection timing and the charge dilution by EGR, despite the improved air–fuel mixing, which can accelerate the combustion rate.

On the other hand, in-cylinder P_{max} was similar at both fuel injection pressures, while MPRR at the fuel injection pressure of 120 MPa was higher after the mode transition. In this period, the combustion rate

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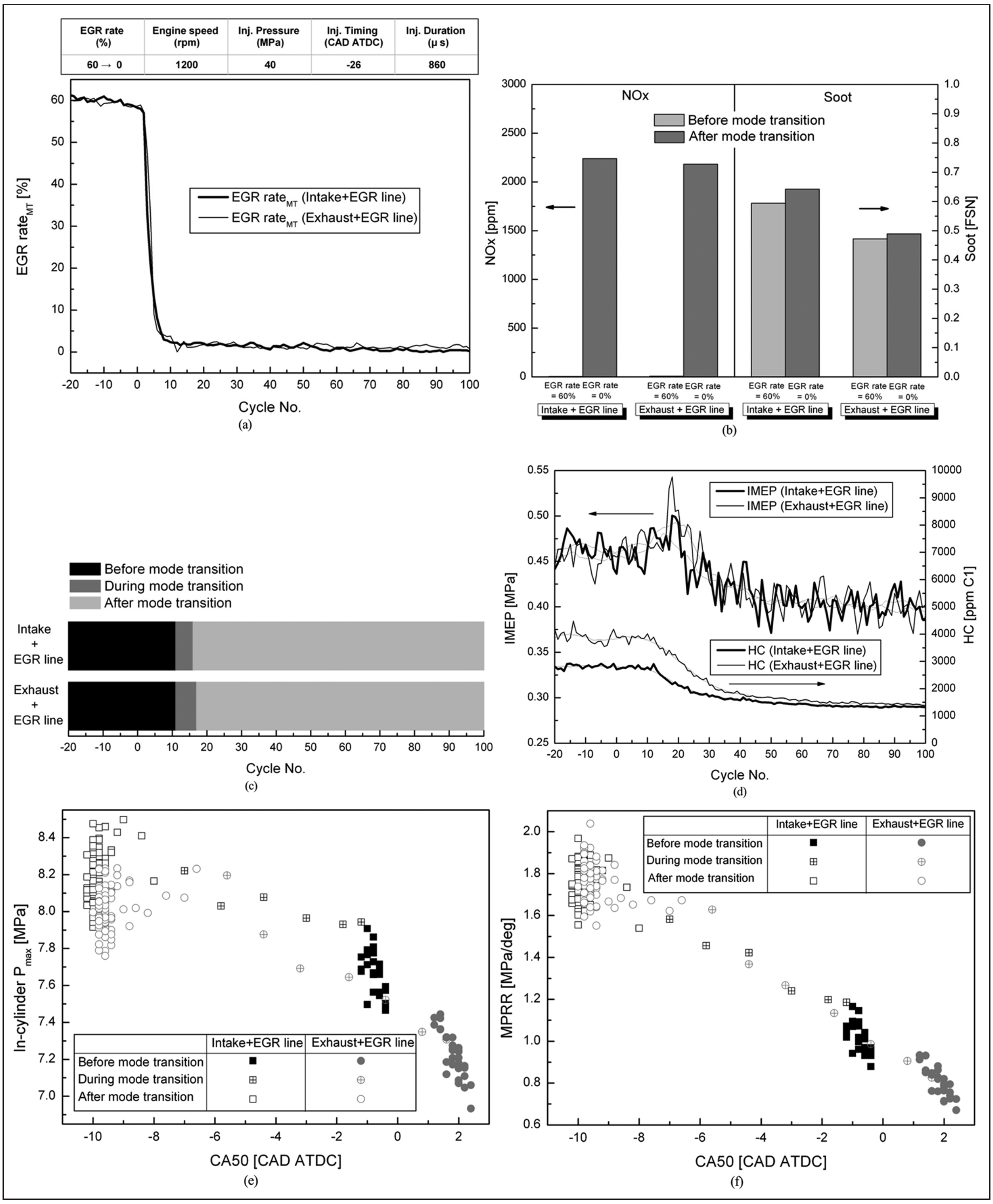

Figure 7. Effects of residual gas on combustion mode transition: (a) EGR rate; (b) NOx and soot emissions before and after mode transition; (c) classification of combustion mode; (d) IMEP and HC emission; (e) in-cylinder P_{max} versus CA50; and (f) MPRR versus CA50.

was fast due to the absence of EGR. Thus, CA50 in both cases occurred before TDC, which means that the amounts of burned fuel before TDC were large at both fuel injection pressures. Consequently, in-cylinder P_{max} at both injection pressures occurred at TDC and showed similar levels. Higher MPRR at the fuel injection pressure of 120 MPa, meanwhile, resulted from the improved air–fuel mixing, as well as the CA50 before TDC. It can be concluded that levels of in-cylinder P_{max} and MPRR are determined by not only in-cylinder mixing conditions, but also the combustion phase.

Effects of engine speed on combustion mode transition

Mode transition from LTC to conventional combustion was performed at different engine speeds of 1200 r/min and 2000 r/min, as can be seen in Figure 9. The start of

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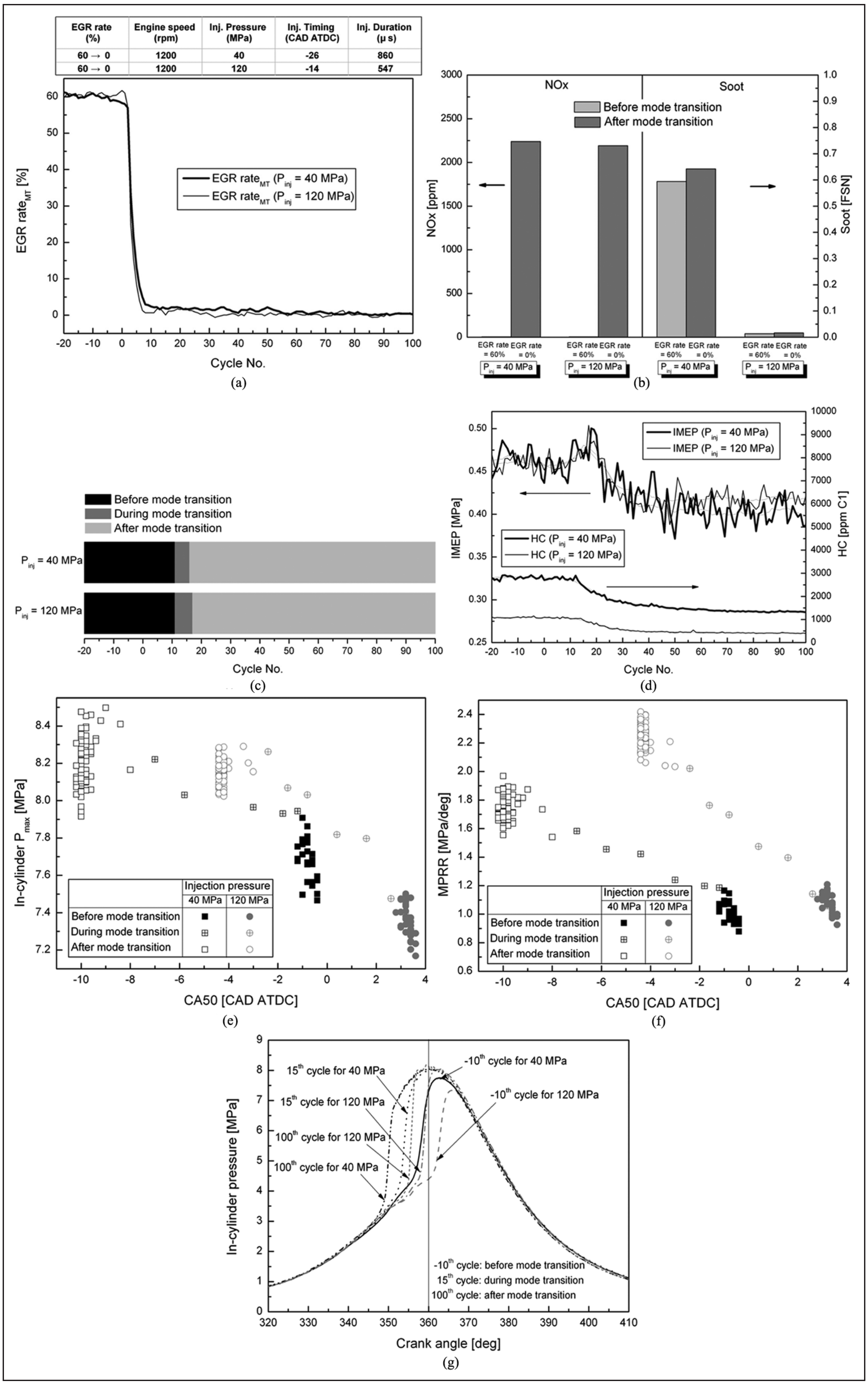

Figure 8. Effects of fuel injection pressure on combustion mode transition: (a) EGR rate; (b) NOx and soot emissions before and after mode transition; (c) classification of combustion mode; (d) IMEP and HC emission; (e) in-cylinder P_{max} versus CA50; (f) MPRR versus CA50; and (g) in-cylinder pressure.

mode transition at the higher engine speed was earlier due to a reduced EGR delay as a consequence of higher flow velocity, while the end of mode transition was almost the same with the lower engine speed. In other words, it took more cycles to complete the mode transition process at the higher engine speed. However, the duration of mode transition at the engine speed of 2000 r/min (0.48 s) was slightly shorter than that at the

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engine speed of 1200 r/min (0.50 s) in terms of absolute time. Although the mode transition occurred earlier at higher engine speed, IMEP variations in both cases showed similar trends because the combustion phase was also changed according to the change of in-cylinder P_{max}.

Figure 9(g) shows the ignition delay for different engine speeds during the mode transition. The ignition

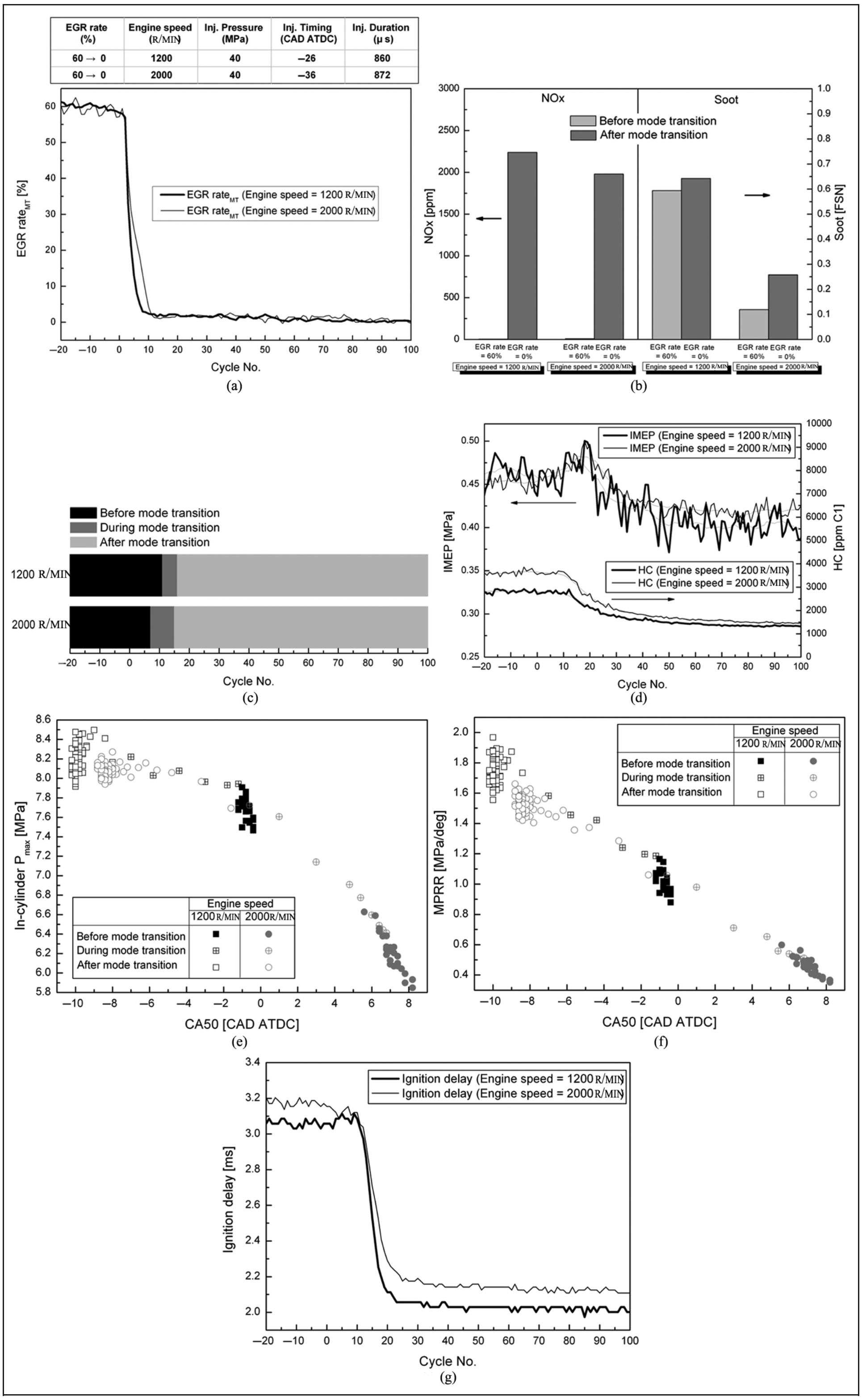

Figure 9. Effects of engine speed on combustion mode transition: (a) EGR rate; (b) NOx and soot emissions before and after mode transition; (c) classification of combustion mode; (d) IMEP and HC emission; (e) in-cylinder P_{max} versus CA50; (f) MPRR versus CA50; and (g) ignition delay.

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delay was defined as the time between the start of injection and the start of combustion, which is the crank angle position where 10% of the total heat release occurs (CA10).[21] In general, an increase in engine speed results in a slight decrease in ignition delay because the peak compression temperature increases with increasing engine speed due to smaller heat loss during the compression stroke.[18] The engine speed of 2000 r/min, however, showed somewhat longer ignition delay compared to the engine speed of 1200 r/min due to earlier injection timing. Even if there was little difference in ignition delays in absolute time, the difference in terms of CAD was quite huge. This largely affects the combustion phase and thereby in-cylinder P_{max} and MPRR. The combustion phase at the engine speed of 2000 r/min was more retarded than that at the engine speed of 1200 r/min, even though the injection timing was earlier. Therefore, the amount of burned fuel before TDC was less than that at the engine speed of 1200 r/min. This led to slowing the combustion rate and also reducing the MPRR and the in-cylinder P_{max}.

The longer ignition delay at the engine speed of 2000 r/min allowed more time for premixing. Furthermore, the higher engine speed promotes higher turbulence, which enhances the mixing of air, fuel and EGR. These two reasons resulted in a globally lean mixture and more homogeneous EGR distribution inside the combustion chamber. Therefore, HC emissions were higher at the engine speed of 2000 r/min. This is particularly noticeable before mode transition, i.e. at the high EGR rate condition. In other words, the difference in HC emissions between the two engine speeds was reduced as mode transition proceeded. The reason for this was that the effect of homogeneous EGR distribution diminished during mode transition because the amount of EGR was reduced.

Smooth combustion mode transition by cycle-by-cycle injection modulation

It is clear from the above discussion that a rapid decrease of IMEP and an increase of noise level represented by MPRR are unavoidable when the mode transition from LTC to conventional combustion occurs by changing the EGR rate only. The reason for these problems is that the injection timing and injection duration were fixed to the optimized conditions for one combustion mode, despite the switched combustion mode. These types of rapid variations during mode transition make the vehicle passengers uncomfortable. Therefore, four kinds of injection strategies were applied to achieve a smooth mode transition when the combustion mode switched from LTC to conventional combustion. In this section, variations of IMEP and MPRR were the focus of attention.

Rapid variations of IMEP and MPRR occurred due to improper injection conditions during mode transition. Therefore, matching the injection conditions for the start and end of mode transition with optimized conditions for each combustion mode was used as a basic injection strategy. Baseline condition A (specified in Table 2) was chosen for this test. The optimized injection timing and injection duration for each combustion mode were chosen from steady-state tests. The optimized injection timing and injection duration were −26 CAD ATDC and 860 μs for LTC and −4 CAD ATDC and 817 μs for conventional combustion, respectively. These conditions satisfied the lowest emissions and fuel consumption as well as an IMEP of 0.45 MPa for each combustion mode. Next, cycles where the injection strategy was applied were found from the mode transition test without an injection strategy. The 15th–21st cycles in which MPRR increased rapidly when the mode transition occurred without an injection strategy were chosen as the cycles during which to apply the injection strategies. On the other hand, injection conditions for the case without injection strategy were fixed as the optimized injection conditions for LTC, namely the injection timing of −26 CAD ATDC and injection duration of 860 μs.

Figure 10(a) shows the first injection strategy and the comparison with the case without injection strategy in terms of IMEP and MPRR. The first injection strategy gradually retarded injection timing from −26 CAD ATDC to −4 CAD ATDC while maintaining injection duration (860 μs) during selected cycles. The aim of this strategy is to prevent the advance in combustion phase and thus reduce the MPRR. Although the MPRR after mode transition was significantly reduced, a high peak of MPRR appeared during mode transition. This is because injection timings from the 15th to the 17th cycle were still early, despite the gradually retarded injection timing. Therefore, the effect of retarding injection timing on retarding combustion phase was weak, and thus MPRR was still high in this period. After the 17th cycle, MPRR was reduced gradually until the 21st cycle due to the decreases in ignition delay and the amount of burned fuel before TDC that resulted from retarding the injection timing. Meanwhile, IMEP increased during mode transition owing to the retard in the combustion phase.

The second injection strategy was to gradually decrease injection duration from 860 μs to 817 μs while maintaining injection timing (−26 CAD ATDC) during selected cycles, as can be seen in Figure 10(b). As a result of decreasing injection quantity, IMEP and MPRR were more reduced during and after mode transition. Therefore, the difference in IMEP before and after mode transition was larger compared to the case without injection strategy, while that in MPRR was smaller.

The third injection strategy shown in Figure 10(c) was the combination of the two injection strategies above. In other words, injection timing was gradually retarded while injection duration was gradually decreased at the same time. The rapid decrease in IMEP disappeared and IMEP maintained a constant

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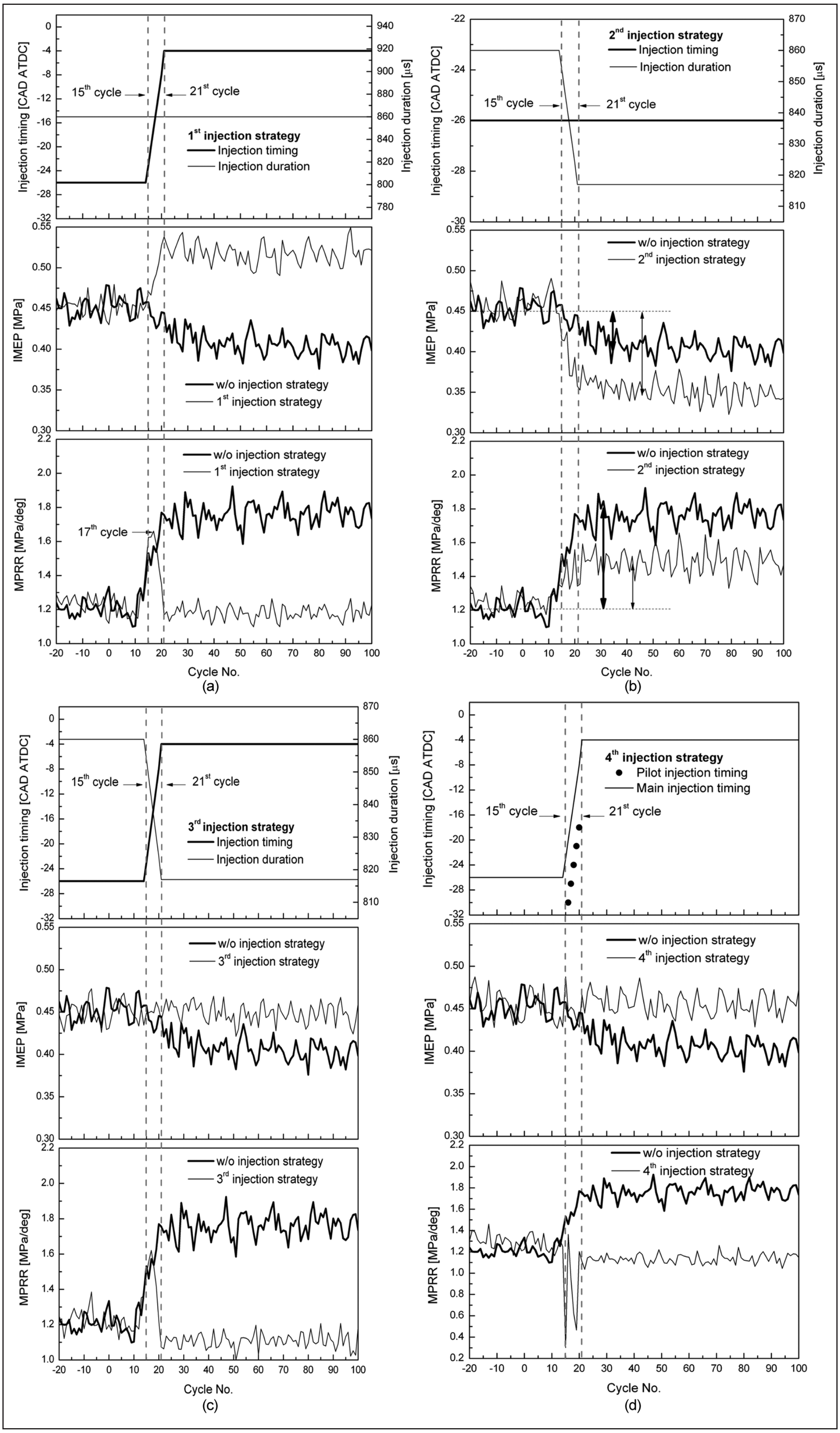

Figure 10. Effects of transient injection control on combustion mode transition (baseline condition A). (a) First injection strategy: gradually retarding injection timing, (b) second injection strategy: gradually decreasing injection duration, (c) third injection strategy: gradually retarding injection timing and decreasing injection duration and (d) fourth injection strategy: pilot injection.

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Table 4. Injection conditions of the pilot injection strategy.

Cycle no.	Pilot injection			Main injection			Pilot ratio (m_{pilot}/m_{total})
	Timing (CAD ATDC)	Duration (μs)	Mass (mg)	Timing (CAD ATDC)	Duration (μs)	Mass (mg)	
15th	-	-	-	−23	854	11.80	-
16th	−30	600	6.00	−20	700	8.40	0.41
17th	−27	600	6.00	−17	736	9.24	0.39
18th	−24	600	6.00	−14	728	9.06	0.40
19th	−21	600	6.00	−11	700	8.40	0.41
20th	−18	600	6.00	−8	642	7.16	0.45
21st	-	-	-	−4	817	11.04	-

level due to the combination of contradictory effects: the retarded combustion phase, which had a positive effect, and the reduced injection quantity, which had a negative effect, in terms of increase in IMEP. However, the high peak of MPRR, which was mainly influenced by injection timing, still remained.

Finally, a pilot injection strategy was applied to reduce the high peak of MPRR. A pilot injection strategy is well known as an effective way to reduce the combustion noise by shortened ignition delay.

The pilot injection conditions applied in this strategy are summarized in Table 4. The main injection timings were the same as those of the third injection strategy, and pilot injection timings were 10 CAD earlier than each main injection timing. Pilot injection durations were fixed at 600 μs. Since the pilot injections were added, main injection durations were less than those of the third injection strategy to maintain the IMEP.

As a result of the pilot injection strategy, the high peak of MPRR disappeared while IMEP remained stable. On the other hand, the low peak of MPRR, which has no influence on the loud noise, occurred during mode transition, as can be seen in Figure 10(d). Nevertheless, it is not possible to conclude that this strategy was optimized for smooth mode transition. It is expected that mode transition can be further improved through feedback control. However, the potential for smooth combustion mode transition by cycle-by-cycle injection modulation was shown.

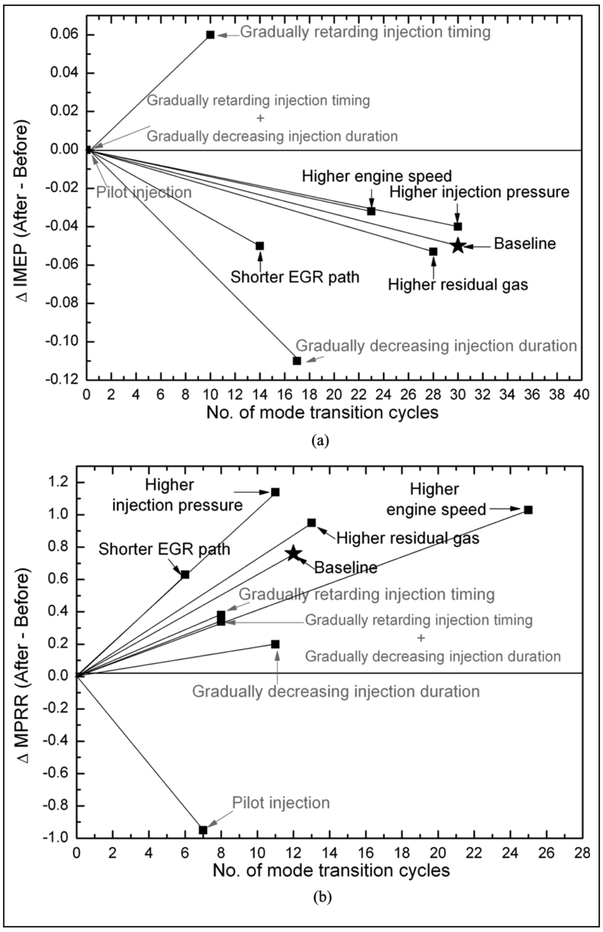

Figure 11. Effects of operating parameters on the mode transition response in number of cycles and differences before and after mode transition as a function of: (a) IMEP and (b) MPRR.

Criteria for desirable combustion mode transition

In general, faster and smoother mode transition, which can be achieved by a smaller difference in IMEP and MPRR before and after mode transition, is desirable. Based on the experimental results discussed above, it can be revealed how each operating parameter has an influence on the desirable combustion mode transition. Figure 11 shows the mode transition response in number of cycles and differences before and after mode transition as a function of IMEP and MPRR for each operating parameter. It indicates which parameters are favourable for a desirable mode transition. The use of the shorter EGR path was favourable to faster mode transition in terms of both IMEP and MPRR. For the smooth mode transition, the higher engine speed and the higher fuel injection pressure conditions were favourable to the smaller difference in IMEP, while the use of shorter EGR path was advantageous to the smaller difference in MPRR. Nevertheless, cycle-by-cycle injection modulation is positively necessary to achieve smooth mode transition. Basically, the

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combination of gradually retarding injection timing and decreasing injection duration are essential in the mode transition from LTC to conventional combustion. Furthermore, the addition of a pilot injection in this strategy was an effective way to reduce the combustion noise.

Conclusions

Mode transitions between LTC and conventional combustion in a light-duty diesel engine were investigated with various operating parameters. Injection control strategies were also developed for a smooth combustion mode transition. The major findings from these investigations are summarized as follows.

1. A rapid decrease in IMEP and increase in MPRR occurred due to improper injection conditions as the mode transition proceeded from LTC to conventional combustion by decreasing the EGR rate. On the other hand, IMEP and MPRR in the case of mode transition from conventional combustion to LTC by increasing the EGR rate were changed slowly without rapid change due to the thermal effect of hot residual gas from conventional combustion.
2. Faster mode transition could be achieved by the use of shorter EGR paths due to the reduction of EGR delay, which depends on the time needed for the gas to travel through the EGR pipe.
3. In the case of EGR rate control by means of controlling the exhaust back pressure, the charge dilution effect of EGR was greater due to the larger amount of residual gas resulting from the higher exhaust back pressure. However, this effect diminished as mode transition proceeded from LTC to conventional combustion due to reduced exhaust back pressure.
4. Higher fuel injection pressure resulted in lower HC emission due to improved atomization of spray and air–fuel mixing. However, the effect of fuel injection pressure on the IMEP variation trend was negligible despite the different combustion processes that resulted from different mixing conditions corresponding to fuel injection pressure.
5. With the faster engine speed, EGR delay was reduced due to the higher flow velocity, while the number of mode transition cycles increased. However, the absolute time for mode transition was shorter at faster engine speeds.
6. Rapid changes of IMEP and MPRR that occurred when the combustion mode switched from LTC to conventional combustion when only changing the EGR rate could be solved by cycle-by-cycle injection modulation. As a consequence of gradually retarding injection timing, gradually decreasing injection duration and adding a pilot injection, MPRR was significantly reduced, while IMEP maintained stable.
7. As a result of cycle-by-cycle injection modulation, the differences in IMEP and MPRR before and after mode transition, which have to be small enough for desirable combustion mode transition, were reduced by up to 100% and 225%, respectively, compared to baseline conditions. In addition, the response lags of IMEP and MPRR were also reduced by up to 100% and 42%, respectively.

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Appendix 1

Notation

ATDC	after top dead centre
CA10	crank angle position where 10% of the total heat release occurs
CA50	crank angle position where 50% of the total heat release occurs
CAD	crank angle degree
CAI	controlled auto ignition
CO	carbon monoxide
CO_2	carbon dioxide
EC	eddy current
EGR	exhaust gas recirculation
FFID	fast-response flame ionization detector
FSN	filter smoke number
HC	hydrocarbon
IMEP	indicated mean effective pressure
LTC	low-temperature combustion
MPRR	maximum pressure rise rate
NOx	nitrogen oxide
PCCI	premixed charge compression ignition
PCI	premixed compression ignition
SI	spark ignition
TDC	top dead centre
VVA	variable valve actuation

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