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Effects of Exhaust Throttling on Engine Performance and Residual Gas in an SI Engine

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ABSTRACT

Combustion in engines can be controlled by the amount of residual gas, which has high temperature and heat capacity compared with fresh charge. Residual gas also acts like a dilution gas during combustion period. Accordingly, combustion duration increases, while the peak combustion temperature and nitrogen oxides (NOx) decreases. Amount of residual gas is affected by pressure difference between exhaust and intake, valve timing and engine speed.

The main objective of this work is to identify the effects of exhaust throttle, valve timing and load conditions on residual gas fraction and engine performance. The intake valve open timing was varied freely under fixed exhaust valve close (EVC) timing. Additionally, exhaust throttle has been installed in the exhaust manifold to build up the exhaust back-pressure allowing extra amount of exhaust gases to be admitted into the cylinder during the valve overlap duration.

Operation of exhaust throttle also increases exhaust gas temperature and pumping loss. However, it reduces NOx and hydrocarbon (HC) emissions. The results show that the variation of valve overlap duration with exhaust throttle provides a wider range of residual gas fraction (RGF) without much influence on engine performance.

INTRODUCTION

During the exhaust period, all of the burned gas can't be eliminated from the cylinder. Some gases remain in the cylinder as residual gas. The residual gas takes part in combustion of next cycle as dilution gas. Sometimes this gas is a very important factor affecting burning velocity and ignition delay [1-4].

Residual gas fraction (RGF) is defined as the mass fraction of in-cylinder remained gas from the previous cycle to the total mixture. RGF tends to increase as engine speed decreases, and either load or valve overlap duration increases. If engine speed is increased, the time duration of back-flow and intake pressure is decreased at the same engine operating condition. Therefore, RGF decreases under a high engine speed condition. On the other hand, if valve overlap duration extends, the duration of back-flow increases leading to the increase of RGF. At the higher load conditions, pressure difference between exhaust and intake becomes less and total induction mass becomes more resulting in lower value of RGF.

Recently, one of the new engine technologies to improve specific fuel consumption (SFC) and emissions is controlled auto-ignition (CAI) system, in which fuel is spontaneously ignited during compression stroke. The auto-ignition timing relies heavily on the energy transferred from the residual gas to the fresh charge to ignite the mixture. Due to its relatively high temperature, RGF is one of the most effective factors to control the combustion phase in CAI engine.

The test engine can freely vary the intake valve open (IVO) timing to achieve various valve overlap duration. The effects of various IVO timing on the RGF have been reported [3]. When IVO timing is set before TDC, the residual gas flows from the combustion chamber and exhaust system into the inlet manifold where it mixes with the fresh charge. The fresh charge, diluted with residual gas, begins to flow into the cylinder when the cylinder pressure is reduced to some level below the intake pressure, while the piston moves downwards.

Most EMV (Electro Magnetic Valve) systems can freely vary valve timing, duration and lift, which enable residual gas to be controlled by regulating gas flow. However, in case of mechanical variable valve timing (VVT), valve motion can't be changed freely, which restricts the amount of residual gas. So, exhaust throttle was installed. The exhaust throttling has been used previously for regeneration of particulate trap and improving fuel economy through the increase of exhaust temperature and pressure [7-9]. According to those studies, exhaust throttling generates the exhaust overpressure due to the reduction of exhaust pipe area. Exhaust throttle was installed to take into account these effects and thus extra residual gas could be added to the fresh charge compared to that of the conventional engine system.

In-cylinder sampling method has been widely used to measure the RGF. Fast response flame ionized detector (FRFID), nitrogen oxides (NOx) and carbon dioxide (CO_2) analyzers could analyze the in-cylinder sampled mixture based on assuming the in-cylinder mixture is homogeneous before sampling. However, in a conventional spark ignition (SI) engine, except for a gas-fueled engine, there would be non-homogeneity in the mixing state. Though using the in-cylinder hydrocarbon (HC) concentration measurement with FRFID is a very convenient and fast method, it has some drawback; in that the HC concentration is affected by the sampling location in cylinder. Furthermore, a misfiring cycle is needed for the measurement of HC concentration, which can damage the engine at higher engine loads and speeds. HC analysis from the mixture in exhaust pipe with FRFID was proposed to overcome this drawback [5]. There is no influence of the sampling location on the CO_2 concentration [6], thus measuring method of CO_2 concentration among in-cylinder mixture was used in this study.

In this study, the RGF and engine performance were identified not only by changing the valve timing but also by the exhaust throttle, which was installed before the close coupled catalyst. The operation of the exhaust throttle reduces the effective area of exhaust pipe, causing the exhaust gas pressure and temperature to increase allowing more exhaust gases to enter the cylinder during the valve overlap duration.

EXPERIMENTAL SYSTEM AND CONDITIONS

ENGINE – Figure 1 shows a schematic diagram of the experimental setup. A detailed specification of engine is given in Table 1. It is a four-stroke, water-cooled, four-cylinder double over-head camshaft (DOHC) gasoline engine with a VVT unit and sprocket of exhaust camshaft. Engine speed and load are controlled by an eddy current (EC) dynamometer. Figure 2 shows a schematic diagram of intake and exhaust camshaft, and a VVT unit, which is operated by oil pressure. Using the VVT unit, intake camshaft rotates freely under fixed exhaust camshaft and thus valve overlap duration could be varied from –5 to 35 CAD (Crank Angle Degree) as shown in Fig. 3. Exhaust cam sprocket was installed to change exhaust valve timing. The exhaust camshaft is connected to the intake camshaft by chain as shown in Fig. 2. Therefore, if exhaust camshaft rotates, intake camshaft also rotates the same crank angle degree. And thus intake valve open timing is changed and valve overlap center varies. Exhaust throttle was located before CCC for increasing the exhaust pressure, as shown in Fig. 1. When the exhaust throttle closes, the sectional diameter of the exhaust pipe is decreased from 60 mm to 15 mm.

A piezoelectric pressure transducer (Kistler, 6117b) was used for the measurement of in-cylinder pressure. Two piezo-resistive pressure transducers (Kistler, 4045A5) were installed to measure the intake and exhaust pressures, while UEGO (Universal Exhaust Gas Oxygen) sensors were fitted on the exhaust pipe of the 4th cylinder and before CCC (Close Coupled Catalyst) for the measurement of equivalence ratio.

SAMPLING SYSTEM – For the in-cylinder sampling, a sampling line was implemented on the 4th cylinder wall directly. The gas sample was admitted and analyzed with a gas chromatograph (GC). The sampling system consists of a high-speed solenoid valve, an electronic controller, a vacuum pump, a sampling chamber and sampling lines as shown in Fig. 1. The high-speed solenoid valve has a delay time of approximately 2 ms. The sampling chamber and lines were evacuated before measurement by a vacuum pump. During the compression stroke, the in-cylinder mixture was sampled by the pressure difference between in-cylinder pressure and sampling chamber pressure. The sampling duration was about 70 CAD at 1500 rpm. To get rid of gas in the dead volume, initial sampled gas was passed by. The concentration of CO_2 was measured by GC using the thermal conductivity detector.

Although RGF is defined in terms of mass fraction, it is experimentally determined based on the CO_2 mole concentration in the sampled gas $\left(\bar{x}_{CO_2}\right)_C$ during the compression stroke relative to the CO_2 concentration in the exhaust gas $\left(\bar{x}_{CO_2}\right)_E$ as defined by Heywood[11] and expressed by Eq. (1)

$$\mathrm{RGF} = \frac{\left(\bar{x}_{CO_2}\right)_C}{\left(\bar{x}_{CO_2}\right)_E} \quad (1)$$

EXHAUST GAS – Exhaust gases were analyzed with a gas analyzer (HORIBA MEXA 1500d) to detect CO_2, NOx. Exhaust gas temperature was also measured by a thermocouple (K-type).

Table 1 Engine specification

Bore (mm)		82
Stroke (mm)		93.5
Compression ratio		10.1
Intake/ Exhaust duration		228/228
Valve timing (CAD)	IVO (BTDC)	-11~29
	IVC (ABDC)	59~19
	EVO BBDC)	36~42
	EVC ATDC)	0~6
Valve Lift (mm)	Intake/Exhaust	8.5/8.4

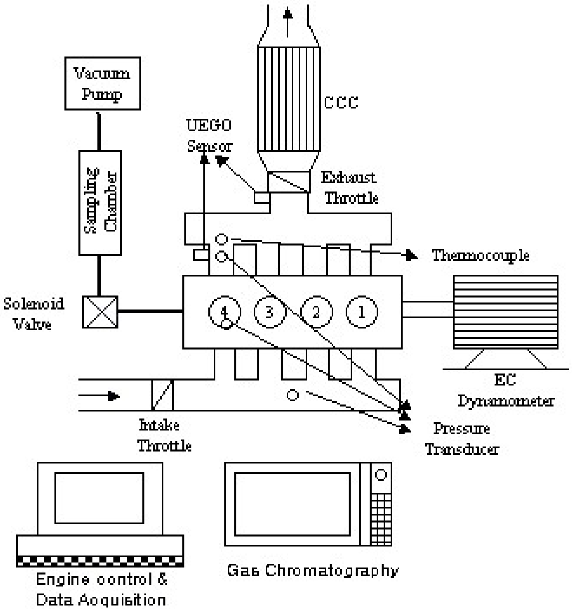


Figure 1 Experimental apparatus

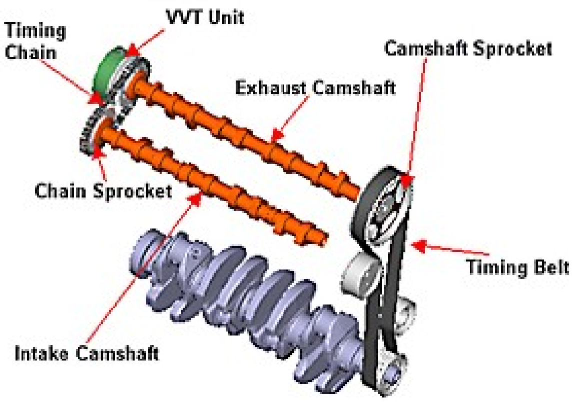


Figure 2 Schematic diagram of valve train

ENGINE OPERATING CONDITIONS – Table 2 shows the engine operating conditions. The engine was operated at 1500 rpm with various IVO and two different EVC timings (EVC at 0 and 6 CAD BTDC), two different torque conditions (20 and 50 N·m), and with exhaust throttling and without exhaust throttling conditions. Throughout the experiments, engine was operated at the stoichiometric condition with MBT (Minimum spark advance for Best Torque) ignition timing.

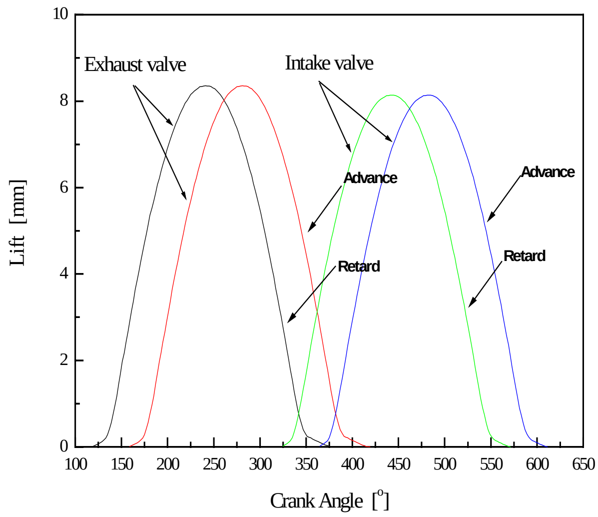


Figure 3 Valve lift diagram

Table 2 Engine operating conditions

Engine speed (rpm)	1500
Valve overlap duration (CAD)	-5, 5, 15, 25, 35
Exhaust valve close timing (CAD ATDC)	0, 6
Torque(N·m)	20, 50
Exhaust throttle	Open, Close(75 %)
Excess air ratio	1
Ignition timing	MBT timing
Fuel	Gasoline (RON 93)

RESULTS AND DISCUSSION

RESIDUAL GAS FRACTION – For the two different EVC conditions, RGF was calculated as a function of IVO timing at 1500 rpm, two different torque conditions and EVC timings with exhaust throttling and non-exhaust throttling. The figures show that the residual gas fraction increases as valve overlap duration extends or IVO timing advances which allowing more exhaust gas to go back into the cylinder. Comparing the results of 20 N·m and 50 N·m torque as shown in Fig. 4 (a) and (b), RGF was found to be decreasing with increasing the torque. This is attributed to the increase in the intake pressure, which reduces the driving force for back-flow and allows more fresh charge to be admitted into the cylinder. Both effects lead to a reduction in RGF.

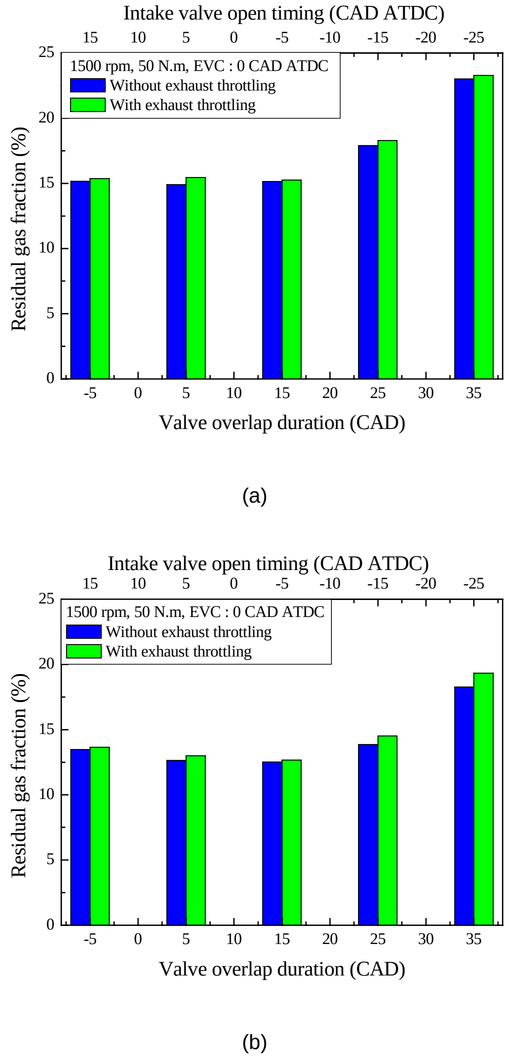


Figure 4 Residual gas fraction as a function of valve overlap duration and exhaust throttle close at 1500 rpm, EVC at 0 CAD ATDC (a) 20 N·m torque condition (b) 50 N·m torque condition

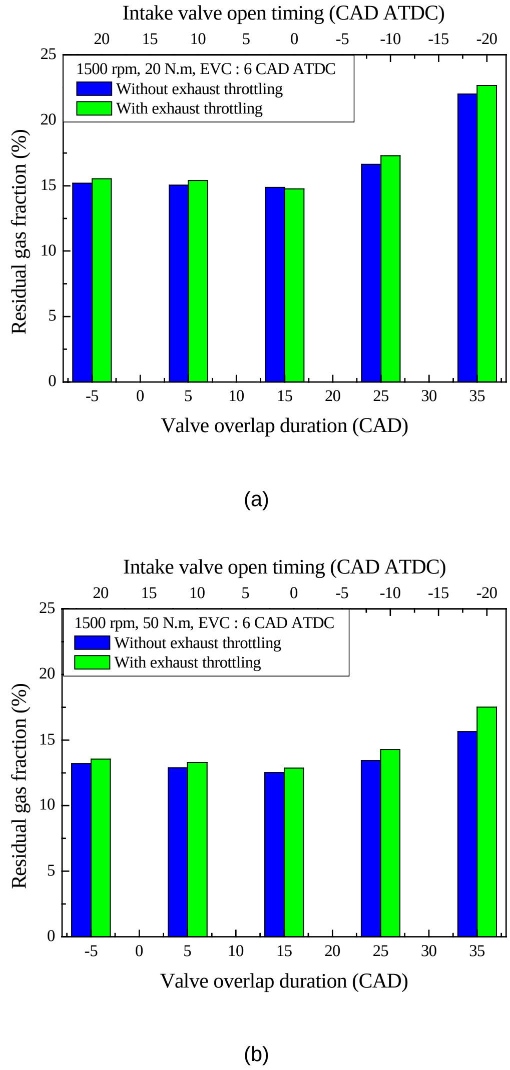
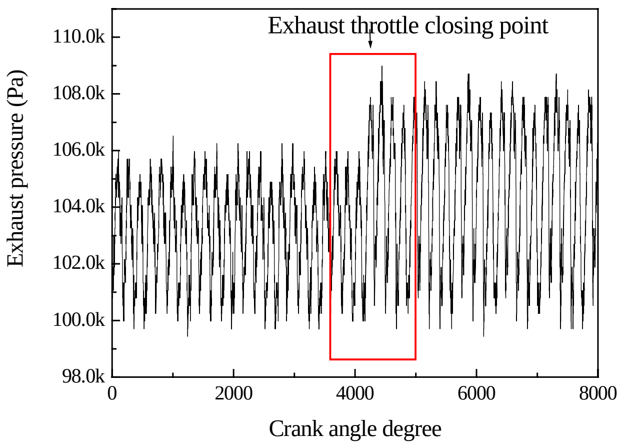


Figure 5 Residual gas fraction as a function of valve overlap duration and exhaust throttle close at 1500 rpm, EVC at 6 CAD ATDC (a) 20 N·m torque condition (b) 50 N·m torque condition

To compare the case of exhaust throttling with that of non-exhaust throttling, RGF was found slightly increased under the condition of exhaust throttle closing. This is attributed to the additional increase in the exhaust pressure. As valve overlap duration and torque increase, the effects of exhaust throttle become more pronounced with more RGF by up to 10% than the opened exhaust throttle.

RGF amount is larger for EVC at 0 CAD ATDC than EVC at 6 CAD ATDC, particularly for longer overlap duration. This is due to the advanced shift of the valve overlap center for the case of EVC at 0 than EVC at 6. This provides the confirmation of previous finding [3].

PRESSURE RESULTS – Figure 6 shows the exhaust gas pressure trace of 50 cycles. When the exhaust pipe was throttled, the gas flow was obstructed and the average exhaust pressure increased by about 3 kPa at the 20 N·m torque condition and by about 5 kPa at the 50 N·m torque condition. These results are similar to that obtained by Pattas et al. [7] who showed that exhaust pressure increased by about 1~2 bar with a reduction in sectional diameter by of 5~18 mm in a diesel engine. Figure 7 shows the exhaust pressure ($P_{exhaust}$) and the pressure difference between $P_{exhaust}$ and intake pressure (P_{intake}) as a function of valve overlap duration.



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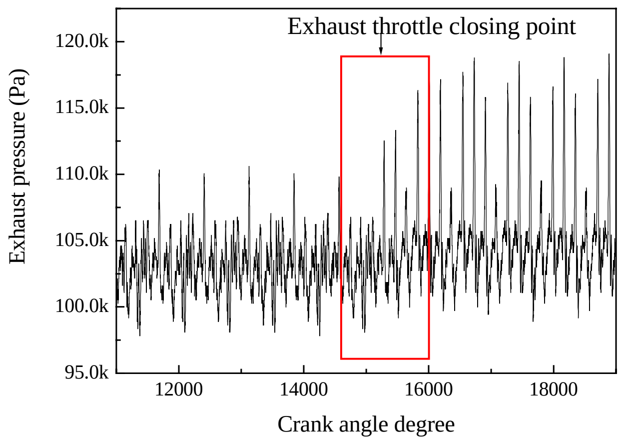
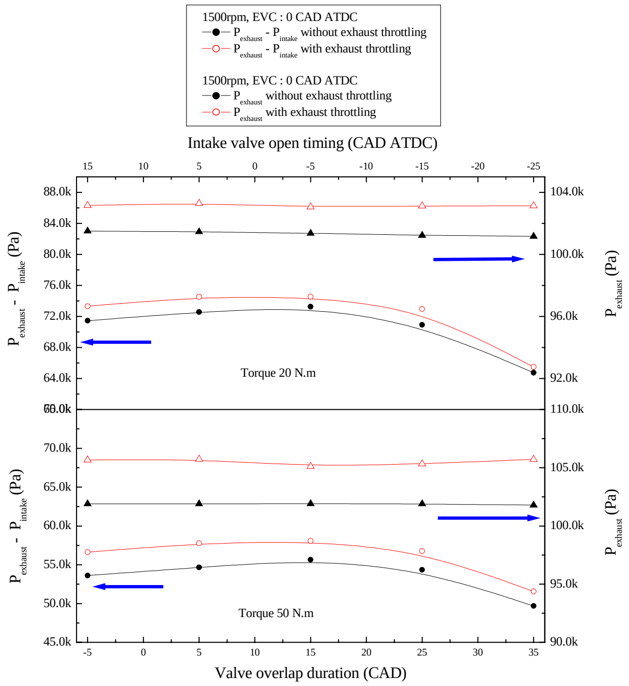


(b)

Figure 6 Exhaust pressure variation at the moment of exhaust throttle close, 1500 rpm, 5 degree valve overlap duration, EVC at 0 CAD ATDC
(a) 20 N·m (b) 50 N·m torque condition

Figure 7 shows that the pressure difference was smaller when valve overlap duration is longer than 25 CAD, though RGF is higher at this condition, as shown previously in Fig. 4 and 5. This implies that the effect of valve overlap duration is more significant at longer valve overlap duration.



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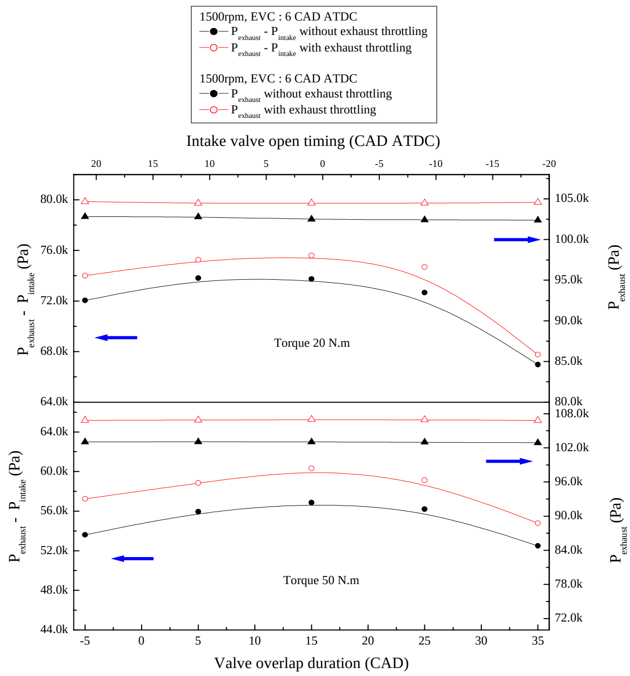
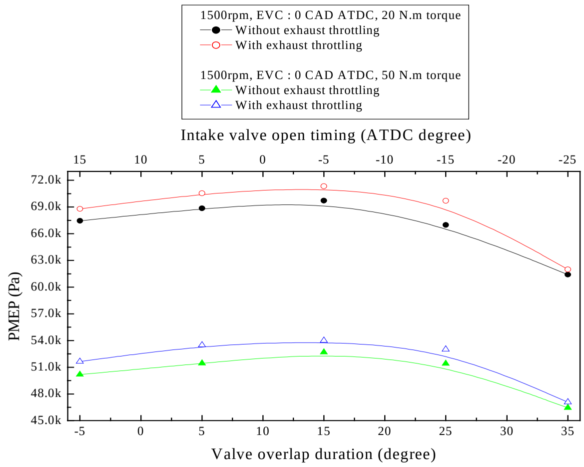


(b)

Figure 7 Pressure difference between intake and exhaust pressure at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC, (b) EVC at 6 CAD ATDC condition



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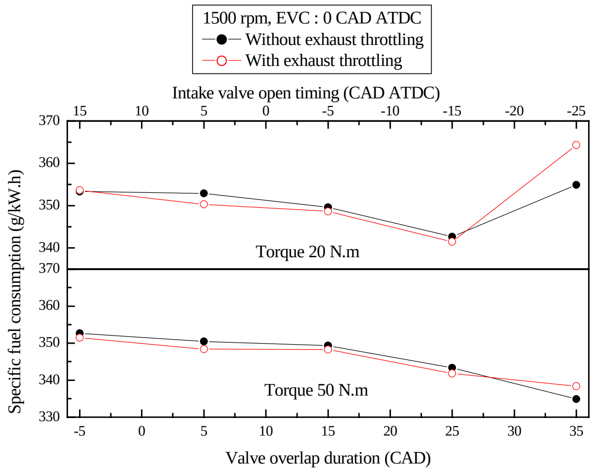

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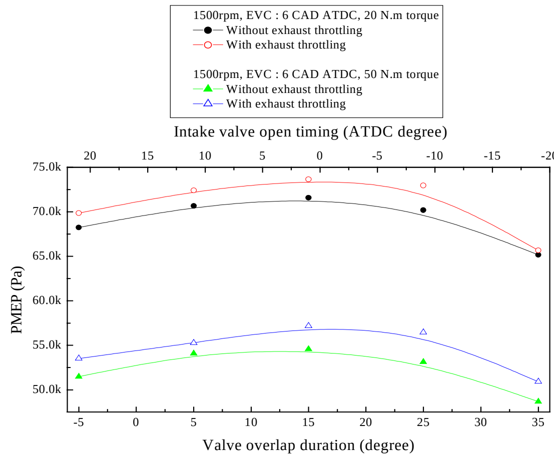

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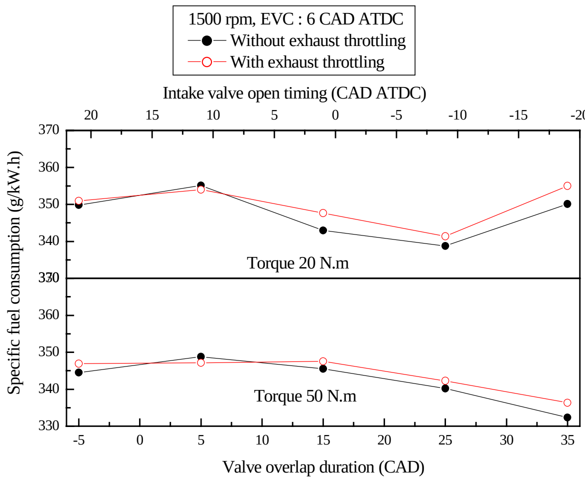


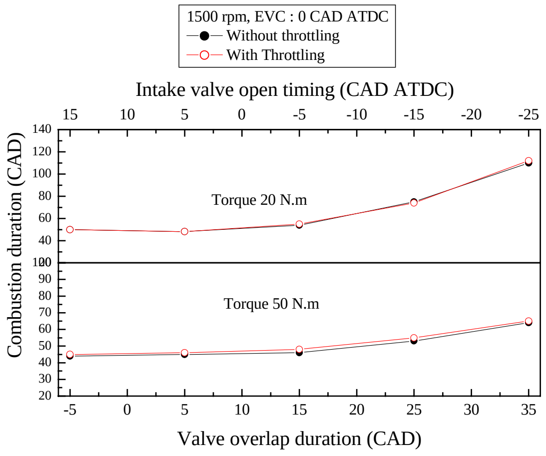
(b)

Figure 8 Pumping mean effective pressure at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC condition (b) EVC at 6 CAD ATDC condition

Figure 9 Specific fuel consumption at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC condition (b) EVC at 6 CAD ATDC condition

Figure 8 shows the pumping mean effective pressure (PMEP), which can be obtained by integrating around the p-V diagram over exhaust and intake strokes, decreases as valve overlap duration increases, torque increases and exhaust throttle opens. In case of exhaust throttling, PMEP increases by up to 6% due to exhaust pressure build-up, compared with the non exhaust-throttling case. As valve overlap duration increases, IVO advances, so that, intake valve opens further away before piston reaches TDC during exhaust cycle. This causes PMEP to decrease.

Figure 9 shows the variation of specific fuel consumption (SFC) at each operating condition. Generally, SFC is found to be improving as valve overlap duration increases at all conditions except at low load with 35 CAD valve overlap duration. At low load 35 CAD valve overlap duration, the combustion duration gets longer because of the higher RGF, and thus high SFC is expected as shown in Fig. 9. SFC was not much affected by the exhaust throttling in most cases. Only at long overlap duration, exhaust throttling leads the higher SFC than non-throttling.



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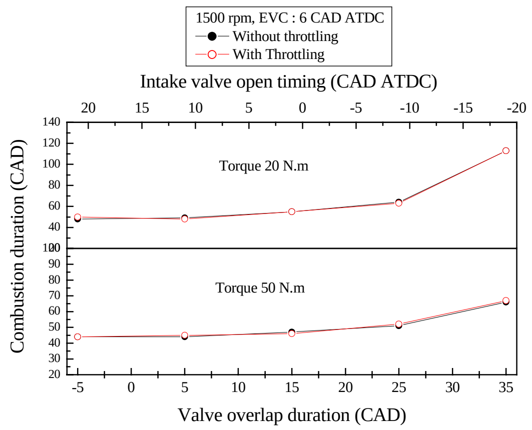
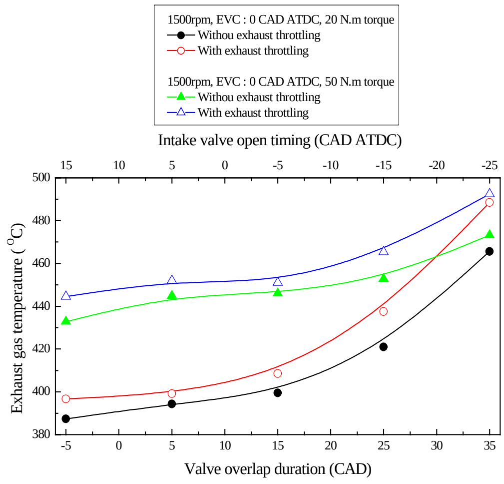


(b)

Figure 10 Combustion duration at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC condition (b) EVC at 6 CAD ATDC condition

Figure 10 shows the combustion durations as a function of valve overlap duration at two different EVC and torque conditions. As valve overlap duration extended, combustion duration takes longer time due to higher RGF. The effect of exhaust throttle on combustion duration was negligible at all cases.

Figure 11 shows the variation of exhaust gas temperature for different operating conditions. As valve overlap duration increases, exhaust gas temperature becomes higher for all the different EVC timings and torque conditions. Exhaust gas temperature increases by 90 ^{o}C at 20 N·m and 40 ^{o}C at 50 N·m. Exhaust gas temperature increases by 5~20 ^{o}C due to the exhaust throttling. If RGF increases, combustion takes longer time as shown in Fig. 10. Due to that longer combustion duration, it is expected that combustion chamber exposes to high temperature for longer time during the expansion stroke.



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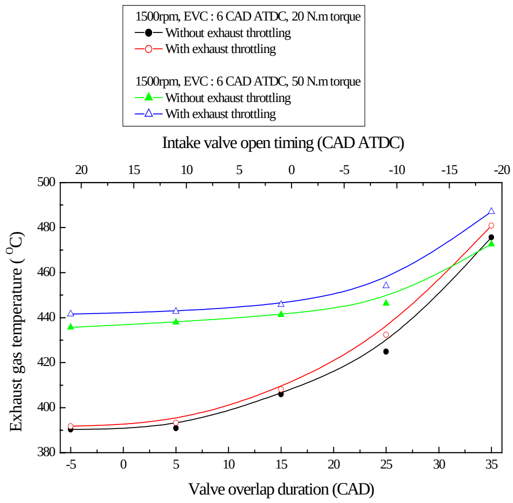


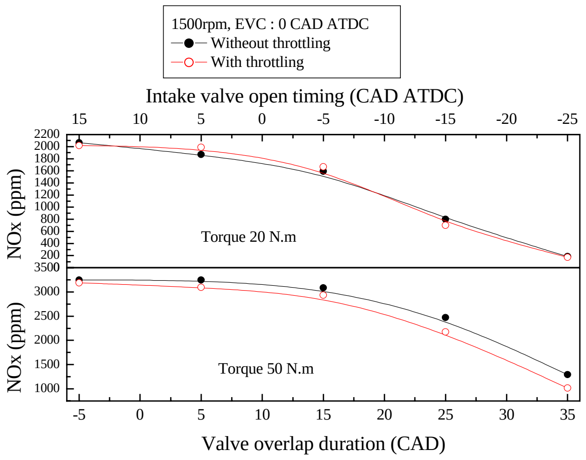
(b)

Figure 11 Exhaust gas temperature at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC condition (b) EVC at 6 CAD ATDC condition

Figure 11 also shows that the exhaust gas temperature at high torque (50 N·m) is higher than at low torque case (20 N·m) at small overlap duration. However, as the overlap duration becomes longer, the rate of temperature increase in the low torque case becomes more significant than in high torque cases. This cause the exhaust gas temperature for the two torque conditions to reach almost the same value at overlap duration of 35 CAD.

Recently, for improving SFC and emission, new engine technology was investigated like a controlled auto-ignition (CAI), which was ignited by auto-ignition during compression stroke. In CAI engine, the auto-ignition timing is mainly determined by the history of pressure and temperature of the in-cylinder mixture during compression stroke. Variation of in-cylinder temperature due to exhaust throttle and change of valve timing could be utilized as a controlling factor of auto-ignition timing. Pre-determined RGF could control the CAI auto-ignition timing.

Figure 12 shows exhaust NOx concentrations as a function of valve overlap duration at two different EVC and torque conditions. This shows a reduction in NOx emissions with the valve overlap duration. At lower torque and valve overlap duration, the effect of exhaust throttle on NOx emission was not significant, compared to higher torque and valve overlap duration conditions.



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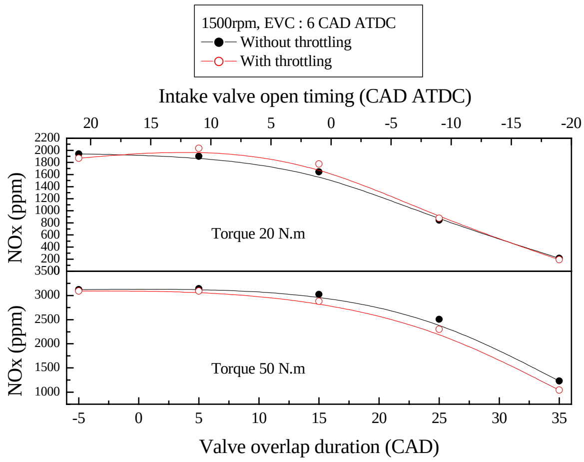

(b)

Figure 12 NOx at 1500 rpm , various valve overlap duration
(a) EVC at 0 CAD ATDC condition (b) EVC at 6 CAD ATDC condition

CONCLUSIONS

Exhaust throttle was installed before CCC (Closed Coupled Catalyst) in a SI engine equipped with a VVT unit. The sectional diameter of exhaust pipe was decreased from 60mm to 15mm by exhaust throttling valve. It was confirmed that exhaust throttling has a potential to increase residual gas. The following conclusions were drawn from this study.

1. The residual gas fraction is increased by exhaust throttling due to the rise in the exhaust pressure, especially at the longer valve overlap and higher torque conditions. The increase in the pressure is estimated to be 3 kPa at 20 N·m torque and increases to 5 kPa at 50 N·m torque.

2. Exhaust gas temperature increases by 90 ^{o}C at 20 N·m and 40 ^{o}C at 50 N·m. Exhaust gas temperature increases by 5~20 ^{o}C due to the exhaust throttling. The effect of exhaust throttling is significant in a longer valve overlap duration.

3. NOx emission obviously decreases by higher residual gas fraction at higher torque and longer valve overlap duration.

4. As valve overlap duration increases, Combustion duration takes longer time due to the higher RGF amount. However the exhaust throttle showed no effect on combustion duration.

5. Although the exhaust valve provides additional way to control the RGF, it increases the pumping mean effective pressure by 6 % due to the throttling process compared to the non-throttling case.

The results of this work provide basic information to control the ignition timing in the controlled auto-ignition (CAI) engine using the pre-determined RGF.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

Roman

$(x_{CO2})_C$	CO2 concentration of sampling gas
$(x_{CO2})_E$	CO2 concentration of exhaust gas
$P_{exhaust}$	exhaust pressure
P_{intake}	intake pressure

Abbreviation

CAD	crank angle degree
CCC	close coupled catalyst
DOHC	double overhead camshaft
EVC	exhaust valve close
EVO	exhaust valve open
EVC at 0	exhaust valve closed at 0 CAD ATDC
EVC at 6	exhaust valve closed at 6 CAD ATDC
GC	gas chromatography
IVC	intake valve close
IVO	intake valve open
MBT	minimum spark advance for best torque
PMEP	pumping mean effective pressure
RGF	residual gas fraction
RON	research octane number
UEGO	universal exhaust gas oxygen
VVT	variable valve timing

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