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Correspondence to:
Jinwook Lee
immanuel@ssu.ac.kr

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Analysis on possibility of expanding low-load operation area for homogeneous charge compression ignition in CI engine with variable exhaust valve actuation

Insu Cho[1], Wootaek Kim[2] and Jinwook Lee[2]

[1]Department of Mechanical Engineering, Graduate School, Soongsil University, Seoul 06978, Korea,
[2]Department of Mechanical Engineering, Soongsil University, Seoul 06978, Korea

Abstract Various technologies are being studied for the advancement of diesel passenger cars and associated environmental regulations. Effective compression ignition combustion in diesel engines is highly dependent on the cylinder charging temperature, composition, and cylinder pressure during valve train operation. The application of variable valve control in diesel engines has several potential advantages. In this study, we applied the variable valve actuation system to a single-cylinder engine model using a GT-POWER simulation and analyzed the effects of the recompression and rebreathing valve profiles, and fuel-injection pressure on the combustion characteristics of a compression ignition engine. As a result, NOx emissions were reduced by more than 90 %, while those of indicated mean effective pressure were reduced by up to 35 %. The benefits of recompression strategies in terms of NOx emissions reduction were confirmed.

1. Introduction

The existing variable valve strategy is an important factor that directly affects the fuel consumption and performance of gasoline engines. And it is necessary to use a variable valve actuation (VVA) system that enhances the freedom of control by using an electronic or hydraulic method. So, the development of mechanisms has also been paralleled [1]. The homogeneous charge compression ignition (HCCI) engine combines the advantages of a gasoline engine with diesel auto-ignition combustion concept, and is generally operated at a low equivalent ratio, thereby lowering the combustion temperature and preventing the production of nitrogen oxides. It also burns pre-mixed gas, so it can reduce smoke generation at the same time [2].

The load of the HCCI engine is controlled by the equivalent ratio, as in a diesel engine. As shown in Fig. 1, the equivalent ratio is too low in the low-load region. Thus, spontaneous ignition cannot occur, or incomplete combustion takes place in the cylinder when spontaneous ignition does occur. In the high-load region, there is a problem in that ignition occurs too early or the pressure in the cylinder increases rapidly, causing knocking as an abnormal phenomenon [3, 4].

Various technologies are being studied regarding the expansion of the technology and environmental regulations for diesel passenger vehicles. The densely compressed ignition combustion in diesel engines depends significantly on the cylinder charging temperature, composition and cylinder pressure during valve train operation [5]. Among them, variable valve actuation (VVA) technology refers to a technique or combination of techniques that changes the valve timing, valve duration, valve lift, etc., depending on the operation conditions and actuation strategy.

The compression ignition engine features the required characteristics of a variable valve system capable of controlling the valve opening period.

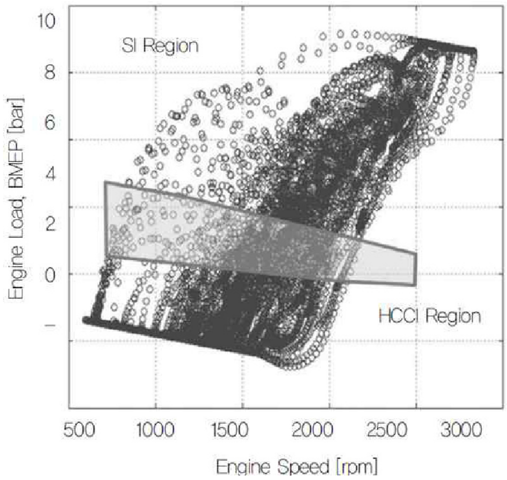


Fig. 1. Comparison of HCCI and SI operation areas [3].

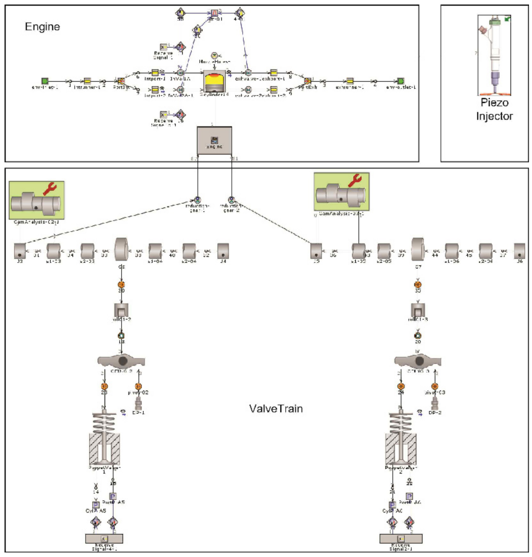
This VVA mechanism has limited valve lift and timing control, depending on the type of mechanism and method that is being operated. Therefore, it is desirable to design a variable valve actuation mechanism in which the ability to control the range of valve lift and opening timing is set at the development stage of the actual technology [6].

The application of variable valve control to diesel engines has several potential advantages. By applying a variable valve function to a compression ignition engine, the temperature of the exhaust gas can be controlled, and the compression ratio can also be effectively controlled through the intake or exhaust air amount [7, 8]. Based on these functional characteristics, the variable valve system is expected to be essential as a core technology for improving the combustion of diesel engines.

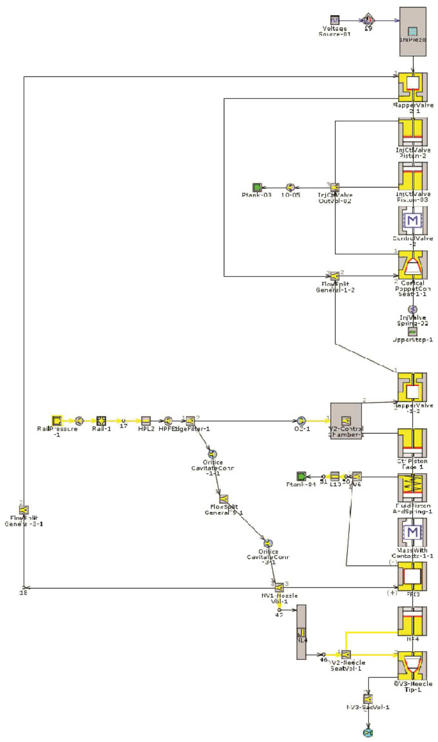
In this study, we applied the two VVA profiles based on an exhaust valve to a single-cylinder engine model using a GT-POWER simulation and analyzed the effect of the recompression and rebreathing valve profiles, and fuel-injection pressure on the combustion characteristics of a compression ignition engine. We also aimed to verify the possibility of expanding the HCCI operating range through a compression ignition engine with variable exhaust valve control.

2. Numerical modeling

GT-POWER (Gamma Technologies), used for numerical analysis, is a commercial engine analysis program that is an industry standard for vehicle and powertrain simulation software. This Integrated CAE application is becoming increasingly important in the industry owing to engine evaluation functions and powertrain system evaluations of activities such as fuel consumption and emission under any driving conditions. In particular, there is also an advantage to including detailed information on intake and exhaust valves, cylinders, and combustion behavior (predictive or data-based functions) for engine combustion calculations.



(a) Full-circuit model for valvetrain system



(b) Piezo-driven injector model

Fig. 2. Analytical model for variable valve actuation and injection.

2.1 Engine model for compression ignition

Fig. 2(a) shows the GT-POWER model developed in this study, which is a valvetrain analysis model for diesel valve control. DOHC type R engine specifications were applied, and the valvetrain system was modeled by measuring the actual dimensions of cylinder head. The camshaft lobe of the valvetrain model was created in association with the engine crank, and the valve profile value was generated and input to

Table 1. Specifications for CI diesel engine.

DOHC type R engine		
Bore × stroke	84 mm × 90 mm	
Displacement	499 cc	
Valve per cylinder	4 (2 intake and 2 exhaust)	
Compression ratio	16	
Intake valve	IVO	bTDC 10 CAD
	IVC	aBDC 28 CAD
Exhaust valve	EVO	bBDC 54 CAD
	EVC	aTDC 4 CAD

Table 2. Specifications for third generation piezo injector.

Item	Unit	Injection feature
Driving mechanism	-	Servo hydraulic
Actuator type	-	Piezo-driven
Number of holes	ea	8
Diameter of hole	mm	134

Table 3. Operating condition of actual R-engine.

Low-load fuel injection quantity : 12 mg(↓)	Low and middle load-fuel injection quantity : 12 & 19.8 mg
Idle + low speed engine speed : 800 rpm	Middle-speed engine speed : 1200-2000 rpm

the valve parameter of the engine model part. A direct-injection diesel multi-pulse combustion model (DIPULSE) with improved NOx prediction accuracy was used for the diesel combustion model, and Woschni-GT was used for the heat transfer model.

Fig. 2(b) shows the analytical model of the third generation piezo high-pressure injector (Bosch) applied to the CI diesel engine. The internal hydraulic circuit was designed with bypass diagram, and the number of injection nozzle holes is 8. The results of the fuel injection model were verified by comparing with experimental data obtained by the Bosch-tube method [9].

Table 1 shows the specifications of the CI diesel engine applied to the engine analysis model, and Table 2 shows the specifications of the piezo-driven injector used in this study. Referring to Table 3, which is the actual operating condition of the R engine, the numerical analysis was carried out at 1200 rpm and 12 mg along five points on the operating map of the CI diesel R engine.

2.2 Analytical condition for VVA

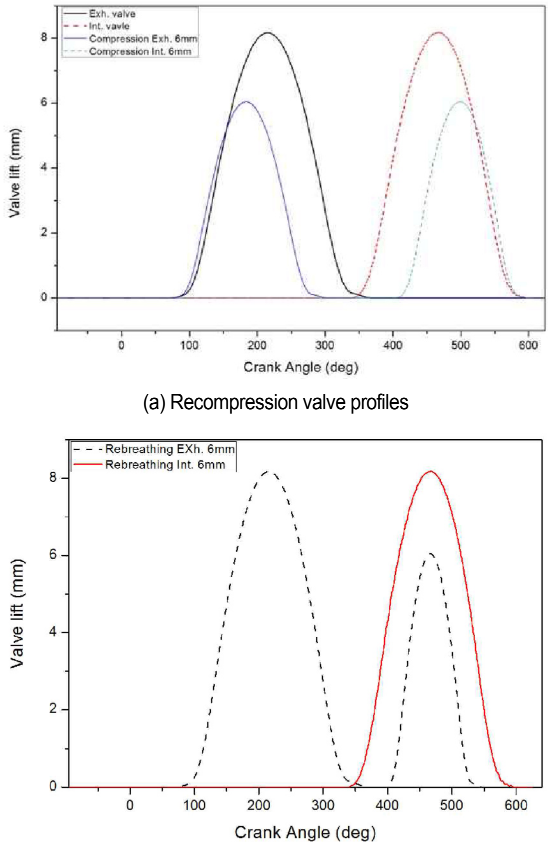
As shown in Fig. 3(a), to implement the recompression strategy, the valve profile was applied under the condition that the intake and exhaust openings were fixed at a maximum lift of 6 mm as compared to the base profile of 8 mm. The valve opening period was reduced by applying a reduced rate of

Table 4. Engine operating condition.

Item	Unit	Specification
Engine speed	rpm	1200
Fuel injection quantity	mg	12
Fuel injection pressure	bar	600
Intake & exhaust valve lift	mm	8

Table 5. Fuel injection condition.

Item	Unit	Pilot	Main
Fuel injection timing	degree	15 ATDC	5 ATDC
Fuel injection quantity	mg	1.4	11.1



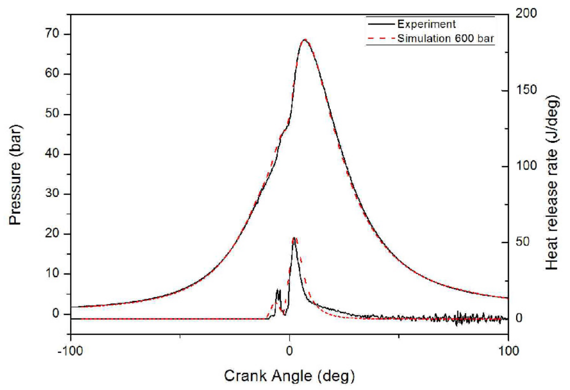
(a) Recompression valve profiles

(b) Rebreathing valve profiles

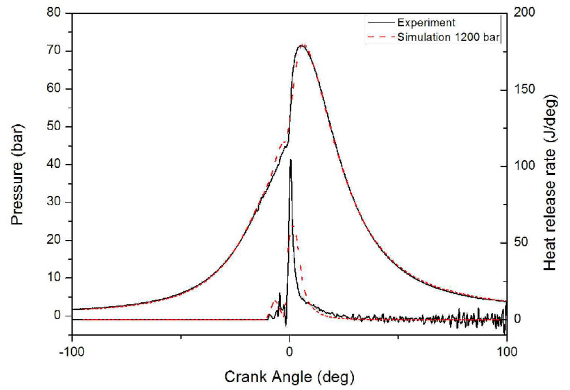
Fig. 3. Two variable exhaust valve profiles used in this study.

6 mm for valve lifting in the base profile. Under the above conditions, the fuel injection pressure was 600 bar and 1200 bar at 1200 rpm. Fig. 3(b) shows the rebreathing profile that allows the exhaust gas to flow back into the cylinder by opening the exhaust valve again during the intake stroke. This can be confirmed by the exhaust valve profile, indicated by the dotted line.

Tables 3 and 4 show the analytic conditions for engine operation and fuel injection used in this study.



(a) Injection pressure : 600 bar



(b) Injection pressure : 1200 bar

Fig. 4. Comparison of heat release rate and combustion pressure between experiment and simulation analysis.

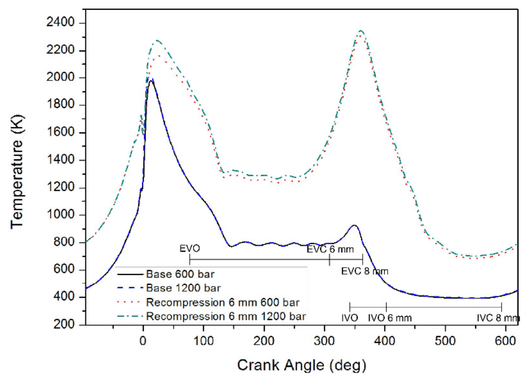


Fig. 5. Comparison of temperature variation.

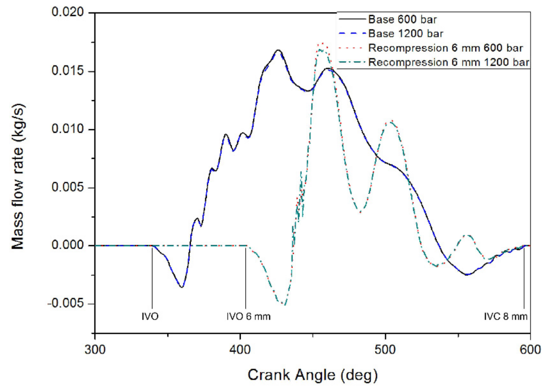


Fig. 6. Variation of intake flow rate.

3. Results and discussion

3.1 Verification of analytic model

Fig. 4 shows a comparison between the experiment and analysis under the same engine specifications used in this study. As seen, the numerical reliability was verified by the similarity between the internal pressure and the heat release rate. In addition, the results were consistent with the experimental results at 600 bar, and the error was confirmed at the maximum heat release rate peak at 1200 bar.

3.2 Effects of recompression valve control

Fig. 5 shows the internal cylinder temperature according to the fuel injection pressure under the recompression condition. It can be confirmed that when the recompression strategy is performed, the engine operates at a relatively high internal temperature. Regardless of the recompression condition, the maximum temperature immediately after the ignition of 0° fuel at 1200 bar is higher; this is considered to be due to improved fuel atomization and increased thermal efficiency. It is confirmed that the temperature increase rate owing to the high-pressure fuel injection is higher when the recompression condition is 1200 bar. And it can be confirmed that the temperature is maintained at a level similar to the ignition timing because the amount of exchange of combustion gas with incoming gas during the intake and exhaust strokes is reduced by negative overlap under the recompression condition.

Fig. 6 shows that the recompression strategy increases the fluctuation of the intake flow because the intake air flow changes during the intake stroke, and the backflow at the beginning of the intake valve opening increases owing to the increased temperature of the internal gas in the cylinder. In the case of 600 bar fuel injection, the maximum flow rate is slightly higher than 1200 bar.

Fig. 7 shows the mass captured inside the cylinder during engine operation. This can be defined as the amount of all kinds of chemical components including air, fuel, and residual gas.

At 6-mm of valve lift, 1200 bar condition, the temperature of the internal gas was higher than that of the 600 bar condition during the recompression period. However, the total amount trapped in the engine at the end of the intake closure was high at 600 bar, and the gas exchange rate was at a relatively low combustion temperature.

Fig. 8 shows the pressure inside the cylinder under the recompression condition. The negative overlap shows a marked difference in the pressure rise in the intake and exhaust strokes compared to the existing positive overlap. This

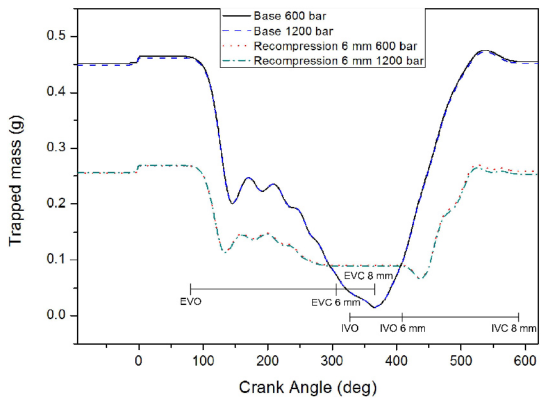


Fig. 7. Variation of trapped mass in cylinder.

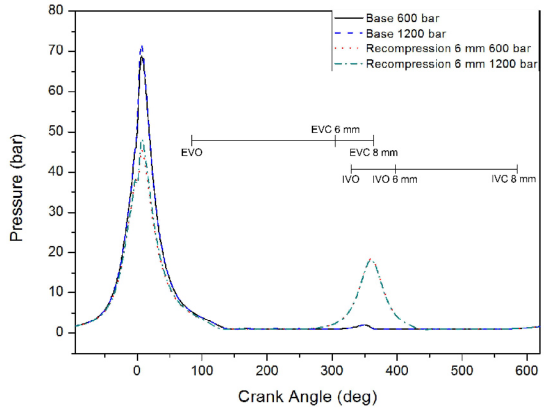


Fig. 8. Combustion pressure inside cylinder.

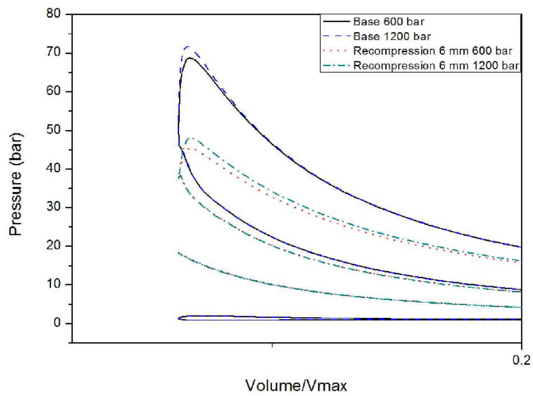


Fig. 9. Pumping loss under recompression control.

was owing to the recompression of the internal combustion gases. The pressure during recompression was constant and independent of the temperature because of the difference in the fuel injection pressure.

Fig. 9 shows the pumping loss. Using the recompression strategy, it was confirmed that the effective work was reduced to a lower value and the fuel injection pressure was increased to 1200 bar to improve the combustion efficiency to some extent. However, the pumping loss owing to the internal heat source increased during the overlap period.

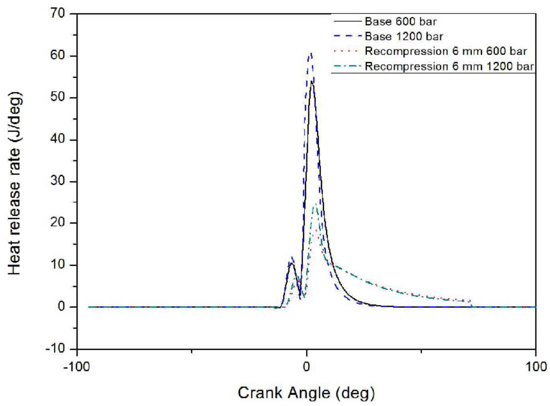


Fig. 10. Change of heat release rate inside cylinder.

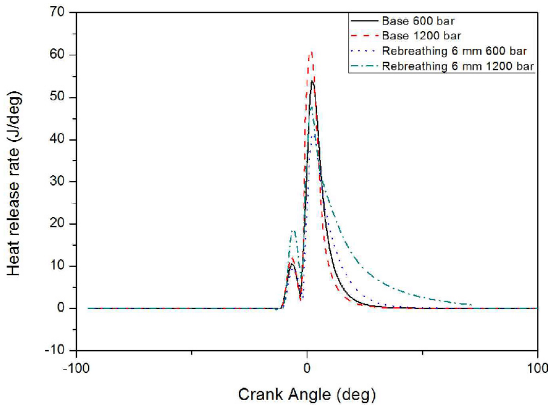


Fig. 11. Comparison of temperature variation at rebreathing.

Fig. 10 shows the heat release rate under the recompression condition. As a result of using the 6 mm recompression strategy, ignition delay occurred during the pilot combustion owing to the lowered fresh air volume and the internal pressure of the engine. The main injection heat release rate decreased but the range and area of diffusive combustion increased, and the material exhibited a characteristic of burning at a low heat release rate.

3.3 Effects of rebreathing valve control

Fig. 11 shows the temperature inside the cylinder under the rebreathing condition, which indicates a higher temperature distribution than the conventional method. In particular, when the injection pressure of the fuel was increased, the temperature increased significantly. The reason for the rise in temperature inside the cylinder was confirmed to be the high-temperature gas flowing backward to the exhaust valve during the intake valve operation in Fig. 12.

Fig. 13 shows the gas trapped inside the cylinder under the rebreathing condition. The relatively high temperature under the 6 mm, 1200 bar condition shows a low trapping amount of gas backwashing when the intake valve was closed. However, it was confirmed that there was a relatively large amount of heat source abandoned owing to high gas exchange in the

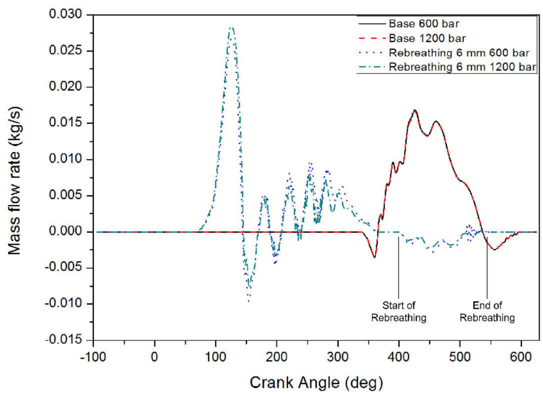


Fig. 12. Variation of intake flow rate at rebreathing.

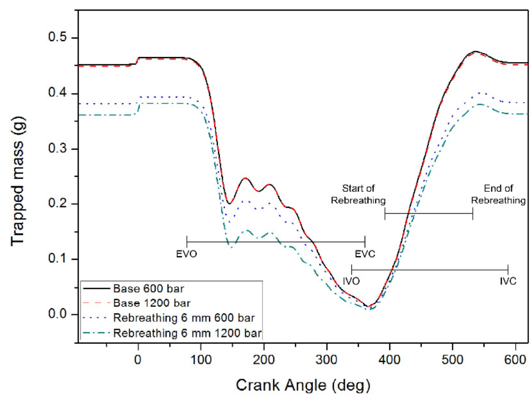


Fig. 13. Variation of trapped mass in cylinder during rebreathing.

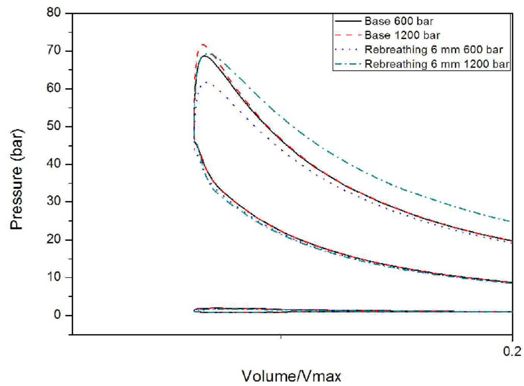


Fig. 14. Pumping loss during rebreathing.

exhaust stroke.

Fig. 14 shows the pumping losses, with rebreathing control and existing valve profiles showing equivalent losses. It is considered that this is owing to the flexible gas flow that resulted from the opening of the exhaust valve during the intake stroke as well as the existing overlap period, unlike the recompression control with negative valve overlap (NVO).

Fig. 15 shows the heat release rate under the rebreathing condition. The area of diffusive combustion increases with the rebreathing strategy and the maximum heat release rate decreases in the main combustion stage.

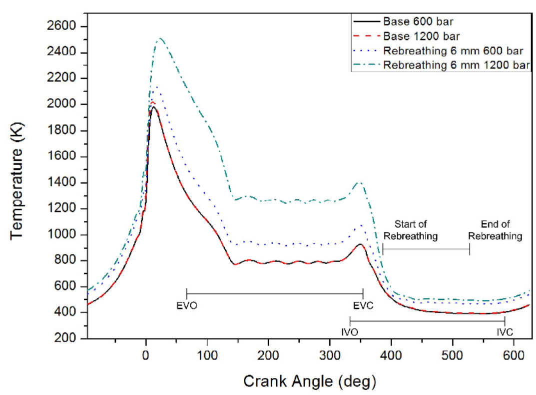


Fig. 15. Change of heat release rate during rebreathing.

In particular, when the fuel injection pressure is 1200 bar, the increase of diffusive combustion is prominent, and it is considered that the combustion performance of the pilot injection timing improves in conjunction with the increased amount of internal combustion gas.

3.4 Quantitative comparison

To confirm the possibility of expansion in the low load operation region of HCCI combustion, the 1200 bar pressure results were examined to study the effects of two valve strategies and the fuel injection pressure.

Figs. 16 and 17 show the indicated mean effective pressure (IMEP) and NOx emissions with two types of valve control. The maximum combustion temperature compared with the existing profile had the advantage of the lowest value under the recompression condition but the change in the IMEP according to the fuel injection pressure was small. In the case of rebreathing control, the difference in the IMEP value according to the injection pressure of the fuel was significant, and it was found that the rebreathing strategy using high-pressure fuel injection was advantageous to improving the power efficiency.

The NOx emissions showed the lowest results under recompression conditions. This is believed to be owing to the efficient utilization of the thermal energy of the combusted gases under relatively low combustion temperatures. Unlike other conditions, rebreathing control increased the injection pressure of the fuel, so the maximum combustion temperature and NOx emissions tended to decrease.

Fig. 18 shows the volumetric efficiency with recompression and rebreathing valve control. As mentioned, it was confirmed that the maximum combustion temperature decreased owing to an increase in the fuel injection pressure under the rebreathing condition. In addition, the changes in the fuel injection pressure did not significantly affect the volumetric efficiency when the valve control was applied at the same time.

With rebreathing, recompression control is appropriate if only considerable reductions in NOx emissions are considered. However, recompression control is necessary to improve fuel efficiency through supercharging or post fuel injection, as

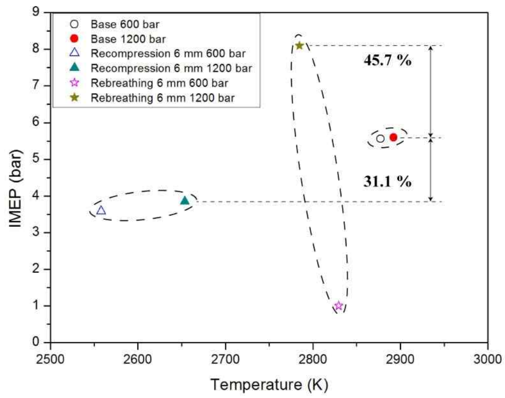


Fig. 16. Comparison of IMEP with two types of valve control.

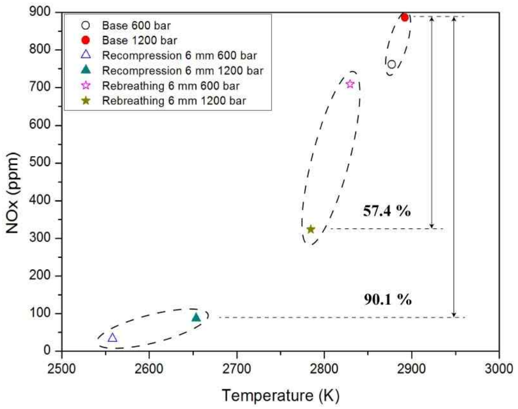


Fig. 17. Reduction of NOx emission by variable valve control.

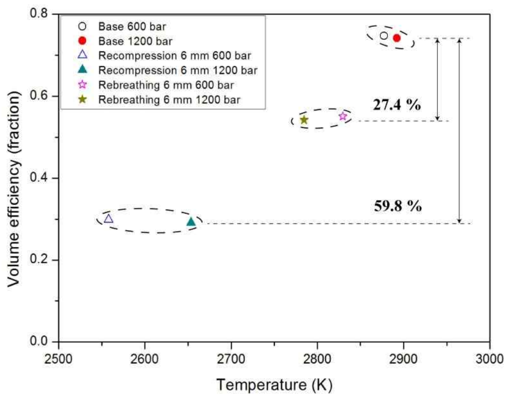


Fig. 18. Comparison of volume efficiency by valve control.

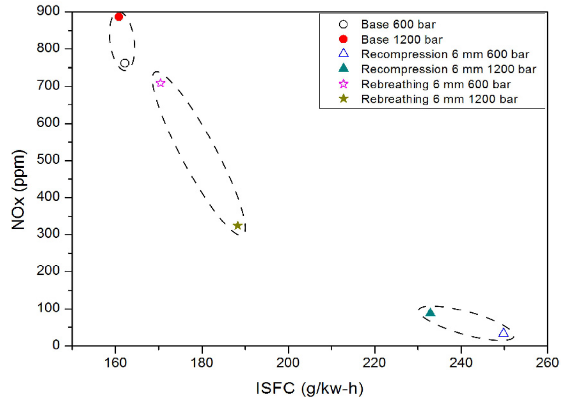


Fig. 19. Comparison of NOx vs. ISFC.

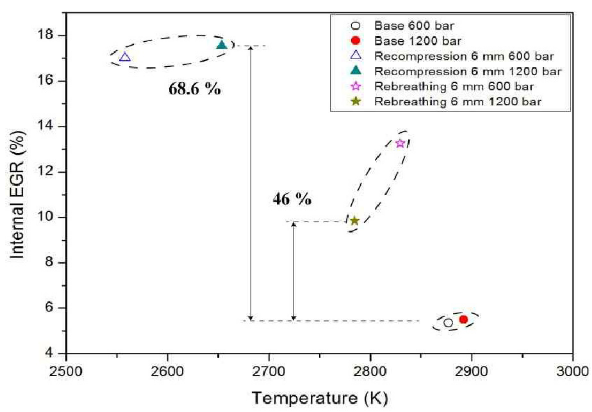


Fig. 20. Effect of internal EGR rate on combustion temperature.

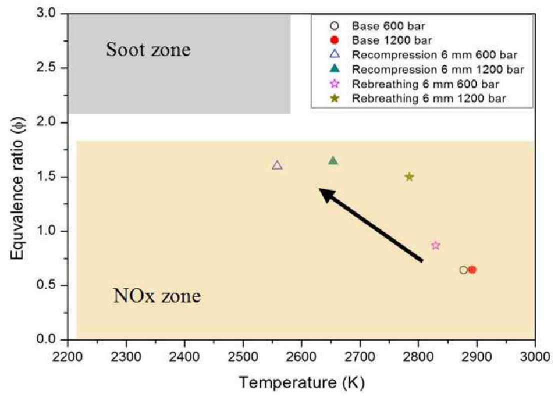


Fig. 21. Comparison result of NOx reduction by applying two different variable exhaust valve control strategies.

shown in Fig. 19.

Fig. 20 shows the internal exhaust-gas recirculation (EGR) ratio, which is the ratio of the cylinder volume after excluding the captured oxygen from the total gas collected at the start of combustion. The recompression with an NVO interval shows the advantage of having the highest internal EGR rate. The rebreathing condition exhibited a relatively low EGR rate increase because the high-temperature gas flowed backward.

Fig. 21 shows a comparison of the NOx reduction between recompression and the rebreathing strategy in this study. As a result of the application of recompression, it gradually deviated from the NOx zone as indicated by an arrow in the figure.

4. Conclusions

An analytical model of a CI diesel engine with a piezo-driven injection concept was developed for a variable exhaust valve system using the GT-POWER analysis program.

Recompression and rebreathing strategies were applied to control the opening and closing timing of the exhaust valves.

The combustion and flow characteristics of the compression ignition engine at a fuel injection pressure of 600 bar and 1200 bar were analyzed. The following conclusions were drawn.

(1) The rebreathing condition showed a higher temperature distribution than the conventional method. In particular, when the injection pressure of the fuel was increased, the temperature increased significantly. And rebreathing control and existing valve profiles showed equivalent losses. It is considered that this is owing to the flexible gas flow as a result of opening the exhaust valve during the intake stroke, as well as the existing overlap period, unlike the recompression control with NVO.

(2) For the recompression condition, it was confirmed that the combustion gas collected in the cylinder owing to negative overlap was recompressed, and the effectiveness decreased by the lowered fresh air amount. It was confirmed that the combustion efficiency was partially compensated by an increase in the fuel injection pressure. As a result of using the 6 mm recompression strategy, ignition delay was observed in the pilot combustion owing to the lowered fresh air volume and the internal pressure of the engine. However, the main injection heat release rate was lowered.

(3) The recompression strategy lowered the rate of gas exchange during the intake and exhaust strokes, indicating that IMEP and the combustion temperature were simultaneously falling. However, with a reduced internal pressure and fresh air volume, the reduction rate of NOx emissions decreased by more than 90 %, while IMEP decreased by up to 35 %. This confirmed the benefits of recompression strategies in terms of NOx emissions reduction.

(4) The recompression strategy showed that the NOx generation point of the engine operation analyzed in this paper gradually deviated from the NOx zone. This result is considered to be due to a decrease in the air amount owing to the internal EGR and the lowered maximum pressure and heat release rate. However, incomplete combustion such as with HC and CO owing to the lowering of oxygen also increased. It is considered necessary to use the combustion gases effectively. NOx reduction by recompression application means that not all of the fuel is converted to effective work, but the engine is operated at low efficiency. Therefore, additional measures for increasing the thermal efficiency, like a turbocharger, are needed.

Acknowledgments

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Nomenclature

VVA : Variable valve actuation
HCCI : Homogeneous charge compression ignition
NVO : Negative valve overlap
IMEP : Indicated mean effective pressure
EGR : Exhaust gas recirculation

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Jinwook Lee received his Ph.D. in Mechanical Engineering from Seoul National University, Korea. He is currently a Professor of Mechanical Engineering, Soongsil University. His research interests are automotive powertrain system and clean energy technology for ICEV, HEV and FCEV.

Insu Cho received his Ph.D. (2020) in Mechanical Engineering from Soongsil Uni-versity, Korea. He now serves as an Assistant Researcher in the Automotive Powertrain Laboratory at Soongsil University. His research interests are fuel injection system, HEV powertrain system, and vehicle system simulation.

Wootaek Kim is an undergraduate in Mechanical Engineering, Soongsil University, Korea. He now serves as a Bachelor-Master's degree student in the Automotive Powertrain Laboratory at Soongsil University. His research interests are fuel injection system, low emission powertrain system, and vehicle system simulation.