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STUDY ON COMBUSTION CHARACTERISTICS AND APPLICATION OF RADIAL INDUCED IGNITION METHOD IN AN ACTUAL ENGINE

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ABSTRACT−This experimental study was executed to obtain basic data for actual engine operation using radical induced ignition method (RI) which can achieve emission reduction and high efficiency due to the rapid bulk combustion. In this study, a direct injection diesel engine was converted into SI type engine with a sparkplug. The modified SI type engine can be divided into two classes. One is the SI engine with a sparkplug only at the cylinder head, and the other is the SI engine with the sparkplug which is enveloped in a sub-chamber. Also, a basic experimental was conducted in order to investigate combustion mechanism of radical induced injection before the experiment execution for actual engine using the modified SI engine. The bulk combustion phenomenon of radical induced ignition method was analyzed from the basic experiment by using a constant volume chamber. Volume value of sub-chamber used in this experiment is approximately 0.2% of one of the main combustion chamber. In this paper, combustion characteristics using radical induced injection method was compared with that of using spark ignition method according to change in the engine speed and equivalence ratio. As a result, in the case of the radical induced injection engine, the combustion duration and cycle variation were respectively reduced ranged from Φ (equivalence ratio)=0.8 (lean mixture ratio) to Φ =1.0 (stoichiometric ratio).

KEY WORDS : Radical induced ignition (RI), Spark ignition (SI), Sub-chamber, Passage holes, Constant volume chamber

1. INTRODUCTION

In the internal combustion engine, the new combustion techniques are to supply lean mixture and to obtain simultaneous ignition in the whole space of combustion chamber for the best operation performance with high efficiency and low emission. HCCI (Homogeneous Charge Compression Ignition) of the diesel engine (Ohyama, 2001) and CAI (Controlled Auto-Ignition) of gasoline engine (Montagne and Duret, 2001) can be typically stated as the new combustion techniques. The object of related studies is located on measurement method of combustion state control because there is no direct means to control the start of combustion in new combustion processes. Main method of combustion state control is to adjust the passive operation parameters e.g. inlet temperature, EGR rate (Aoyama *et al.*, 1996; Olsson *et al.*, 2000), and compression ratio (Christensen *et al.*, 1999). Radical injection method for multi point ignition was reported by previous researchers as one of methods of combustion state control. Radicals are essential factors and activated chemical species that lead to the elementary reaction in the combustion process. If the radical seeding method can be applied to formation of the homogeneous mixture combustion, the bulk combustion can be realized. Pascal (Pascal, 1999) developed a radical induced injection method by using sub-chamber in a gasoline engine. Rich mixture around spark plug is made by supplementary injection into the sub-chamber. The incomplete combustion of rich mixture at the sub-chamber creates lots of radicals of intermediate. It is reported that, when using the method, the burning velocity become faster, and cycle variation is much lower than that of conventional spark ignition. However, it is not clear how the radical induced injection method affects combustion characteristics according to engine load variation, speed, and air-fuel ratio. The main purpose of this study is the application of radical induced injection technique to an actual engine. As first step, the fundamental works to acquire data of radical induced injection combustion were conducted in a constant volume combustion chamber (Park *et al.*, 2004). Also, we applied the radical induced injection method to a modified actual engine. As second step, we investigated optimum condition of the

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geometry shape (volume, diameter and number of passage hole) of the sub-chamber in a actual engine. In this paper, it is purpose to compare combustion characteristics using radical induced injection method with that using spark ignition method according to engine speed and equivalence ratio.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The combustion chamber of modified engine using the radical induced injection type is divided into a sub-chamber and a main chamber like an indirect diesel engine with a sub-chamber. The sub-chamber is located in the upper main chamber. The fuel supply method is different from the indirect diesel engine, and the pre-mixture inducted from the main chamber to the sub-chamber is ignited by spark discharge. The pressure of the working gas in the sub-chamber increases in accordance with the flame develops and the many radicals in combustion gases are propagated into the main chamber through the passage hole. The working principle of the engine with radical induced injection system is shown in Figure 1.

In an SPI (Single Point Injection) type engine, some fuels injected into intake manifold come into the main chamber with air during intake stroke. Pre-mixture enters at the sub-chamber during compression stroke as shown in Figure 1(a). Pre-mixture is ignited by spark discharge near the TDC as shown in Figure 1(b). Burned gases with the radicals are injected into the main chamber by the pressure rise at the sub-chamber, and then multi point ignition by the radicals causes the bulk combustion simultaneously as shown in Figure 1(c). The experimental apparatus for application the radical induced ignition method to a real engine consists of a test engine, an electronic engine control unit, a part of dynamometer, and an analysis system of combustion. A direct injection diesel engine with single cylinder (ND80DI) was used as a base engine which was modified to a spark ignition type engine.

The specifications of the modified engine are shown in

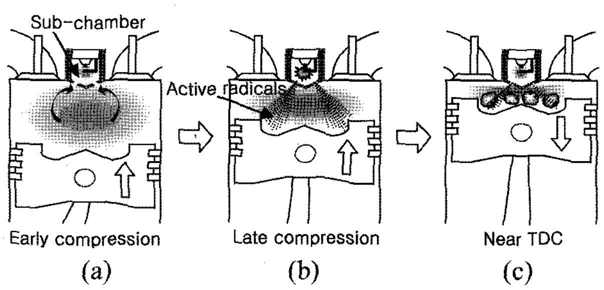


Figure 1. Schematic diagram for working principle of the engine using radical induced ignition method.

Table 1. Specification of test engine.

Specification	Base engine	Modified engine
Type	4 strokes DI diesel engine	→
Bore and stroke (mm)	92 × 95	→
Displacement (cc)	632	→
Compression ratio	19	10.5
Shape of piston head	Toroidal	Flat
Maximum output (PS/rpm)	11/2200	

Table 1. The spark ignition (SI) and the radical induced ignition (RI) engines have a structural difference in the combustion chamber of engine cylinder. The RI engine means the SI engine having a sub-chamber in this study. The compression ratio of modified engine was adjusted from 10.5 to 19 by using a copper gasket of 7 mm in thickness. The piston cavity was exchanged from toroidal type to flat type in order to reduce S/V ratio of the main chamber.

In the prior fundamental experiment by using a constant volume chamber, the relationship between volume of sub-chamber and area (the diameter and the number) of passage holes was expressed in A_h/V_s ratio by Park (Park *et al.*, 2004). Park (Park *et al.*, 2004) carried out a experiment on the geometrical optimization of sub-chamber in the lean burn.

In this chapter, three sub-chambers for manufacture of the RI engine were designed as shown in Figures 2 and 3. Their volume is fixed with 1cc smaller, the number of passage holes (N_h) is varied from 4 to 8. The diameter of passage hole is respectively adopted with 2 mm or 1.3 mm based on the optimum A_h/V_s ratio (0.11 cm^{-1}) from the fundamental study (Park *et al.*, 2002). The passage holes were radially arranged. Also the shape of ground electrodes was changed into two types in consideration of

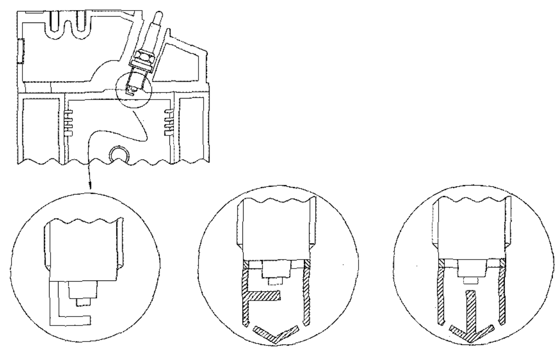


Figure 2. Schematic diagram of SI and RI engine combustion chamber.

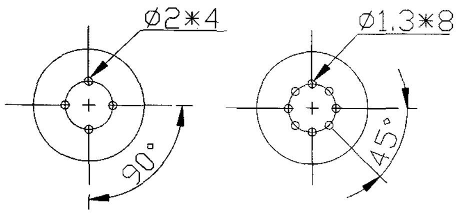


Figure 3. Detailed drawing of RI type A and type B.

Table 2. Experimental conditions.

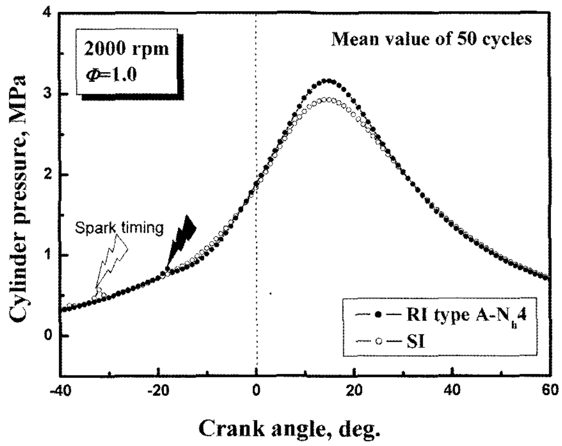
Specification		Condition
Fuel		Gasoline
CWT (°C)		70
Equivalence ratio (Φ)		0.8, 1.0
Engine speed (rpm)		1200, 2000
	Type	$D_h \times N_h$
V_s=1cc (RI engine)	RI type A-N_h4	ϕ 2.0 mm × 4
	RI type A-N_h8	ϕ 1.3 mm × 8
	RI type B-N_h4	ϕ 2.0 mm × 4

the flame kernel grows. Type A is the same shape as a conventional spark plug and type B has a projected ground electrode toward center electrode.

The experimental conditions are shown in Table 2. Used fuel is gasoline, the temperature of cooling water in data acquisition is constantly maintained as 70°C, and spark timing is adjusted on the basis of MBT to keep TDC in the center of pressure rising period. The equivalence ratio of mixture is set as lean (Φ=0.8) and stoichiometric (Φ=1.0). The rpm of the engine is regulated between low speed of 1200 rpm and the middle speed of 2000 rpm. Also the results from RI engine were compared with one from SI engine.

3. RESULTS AND DISCUSSION

In the cases of SI and RI (type A-N_h4) engines, the curves of cylinder pressure versus crank angle are given in Figure 4. Each pressure value is average value calculated from the data measured for 50 cycles at 2000 rpm. The cylinder maximum pressure (P_{max}) by RI engine under Φ=1.0 is slightly higher than that by SI engine, and such difference can be shown greatly under Φ=0.8. This fact can be treated as a characteristic of RI method possessing higher pressure than that of SI in spite of the relative MBT delay. In particular, the problems of the SI engine are the flame propagation of slow velocity, the increasing time loss, and the unstable combustion under lean condition. However, in the case of RI engine, the time loss is decreased in disregard to the existence of sub-chamber because the increase in the effect of reaction



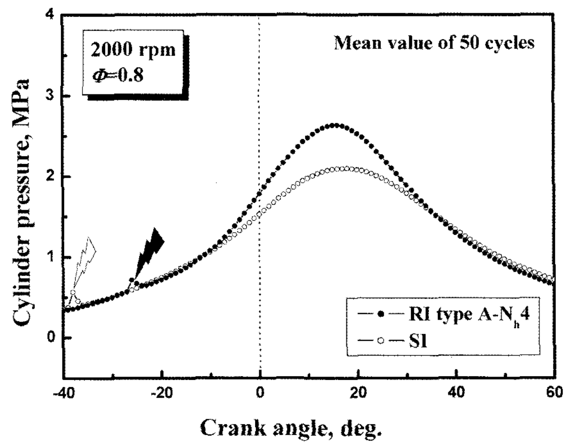


Figure 4. Cylinder pressure versus crank angle for RI and SI engines at 2000 rpm.

zone takes place owning to the spout of active radicals and multi-ignition, and then pressure increment rate, that is, the heat release rate is also increased.

Figure 5 shows the accumulated frequency rate of the overall combustion duration corresponding to a period from spark timing to P_{max} for 50 cycles at 1200 and 2000 rpm. The result implies the fluctuations of P_{max} occurrence timing (CA P_{max}) because ignition timing is fixed in each condition. It can be speculated that the more rapid burning velocity and the higher intensiveness of frequency induce the stable combustion and engine operation. Duration and fluctuation of combustion are wholly reduced for RI as compared with those for SI, regardless of equivalence ratio and rpm. There is no difference in air-fuel ratio at the main chamber and sub-chamber theoretically if air and fuel mixed homogeneously. However, it can be expected that burned gas partially remains at the sub-chamber because gas exchange terminated during a short time in the engine. In other words, the generation of flame kernel becomes merit demerit in the case of the RI as compared with one of the SI because of

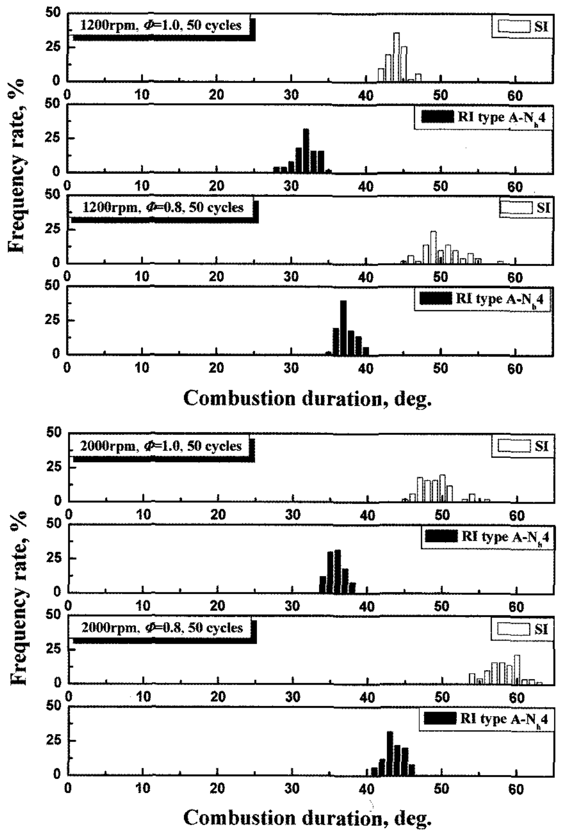


Figure 5. Frequency rate of combustion duration for 50 consecutive cycles by each SI and RI type A-N$_h$4.

burned gas in the vicinity of the spark plug. Nevertheless, the decrement and stability of combustion duration can be explained from effect of the radical induced ignition. Optical measurement should be performed to clear the effects of radical ignition in the engine.

In Figure 6, the change tendency of the mass fraction

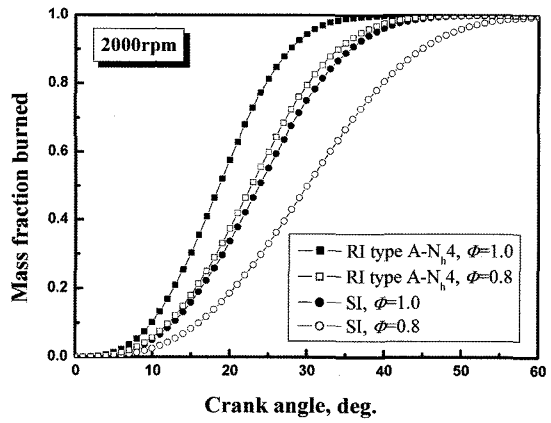


Figure 6. Mass fraction burned versus crank angle for 50 consecutive cycles for each SI and RI type A-N$_h$4.

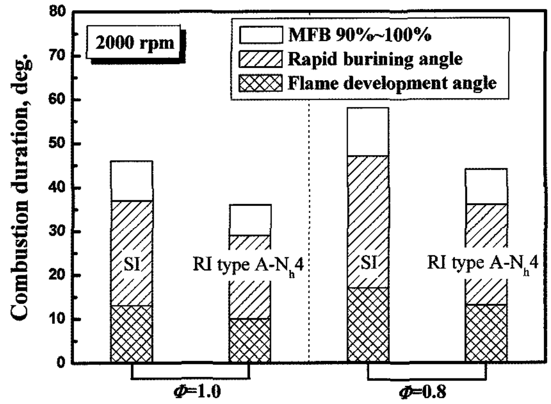


Figure. 7. Each section in a whole combustion duration with equivalence ratio.

burned in the RI engine shows sharp gradients in comparison with that of SI engine under the same equivalence ratio as a whole, and the difference under Φ=0.8 appears more remarkably than that under Φ=1.0. Consequently, it can be concluded that RI method has an advantage in lean burn. Also, such fact can be minutely observed in Figure 7, where the overall burning angle of RI type A-

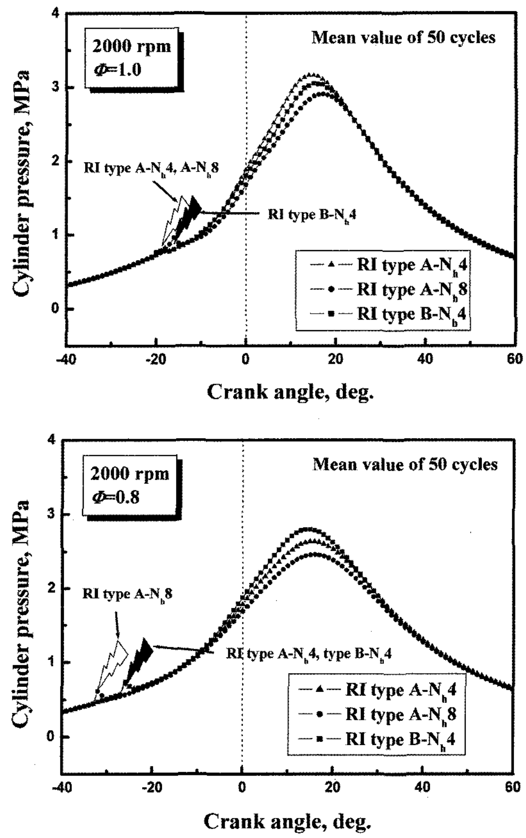


Figure 8. Cylinder pressure versus crank angle each RI engine at 2000 rpm.

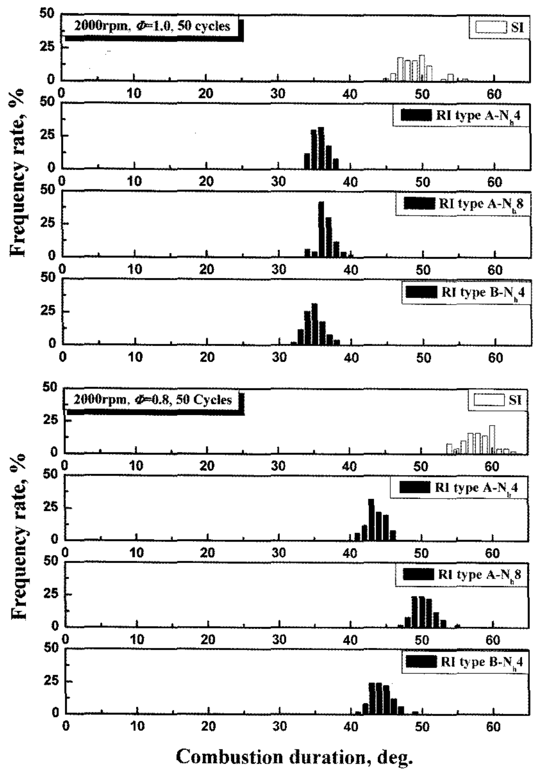


Figure 9. Frequency rate of combustion duration for 50 consecutive cycles at 2000 rpm by each RI condition.

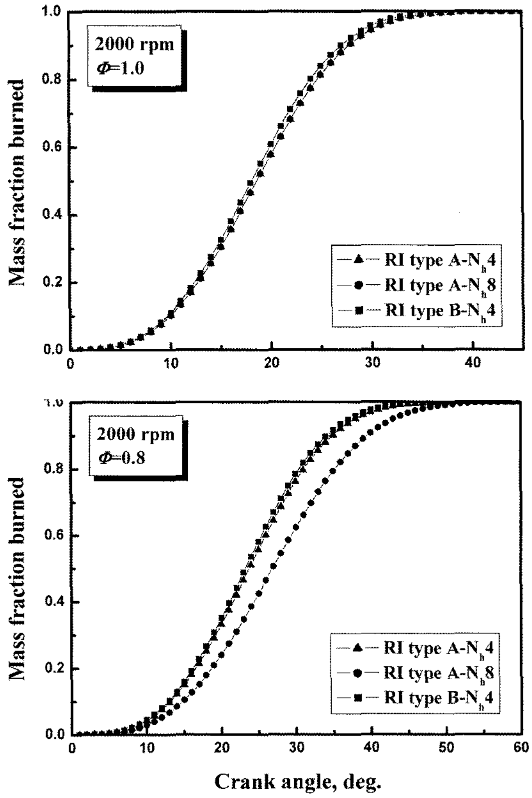


Figure 10. Mass fraction burned versus crank angle for 50 consecutive cycles for each RI condition.

N_h4 under Φ=1.0 and Φ=0.8 is shown with a falling tendency due to a decline of the flame development angle and the rapid burning angle. Then, the case of RI under Φ=0.8 indicates the same tendency as that of SI under Φ=1.0 over almost all sections. It seems reasonable to conclude that the development of initial flame fronts in RI engine has a considerable influence on the next section and the overall burning duration.

Figure 8 shows the mean values of combustion pressure with P-θ diagram according to sub-chamber geometries (type A-N_h4, A-N_h8, and B-N_h4) at 2000 rpm in the RI engine. The cases of type A-N_h4 and A-N_h8 have equal spark timing under Φ=1.0, but nevertheless that of A-N_h4 has some higher P_{max}.

Under Φ=0.8, though the cases of A-N_h4 and B-N_h4 have a synchronously delayed spark timing, they indicate an upward tendency in P_{max} in contrast to the case of A-N_h8. The reason for this is as follows. The wall surface area increases as the hole number increases, which is considered to result in the increase of an instantaneous heat loss into the walls of passage holes during the ejection of products from a sub-chamber. Therefore, the quality of the material building passage holes as well as the number of passage holes should be considered when designing a sub-chamber. Besides, in the cases of the RI, the formation and growth of flame kernel is different with the shape of ground electrode in the same sub-chamber, which also will affect the combustion of the main chamber.

Figure 9 shows the frequency rate of the overall combustion duration during 50 cycles for the SI and each RI engine at 2000 rpm. Under both Φ=0.8 and Φ=1.0. The overall combustion duration in all the RI engines remarkably reduced and the distribution also is dense compared with that in the SI engine. From the viewpoint of rapid combustion and burning stability, the case of B-N_h4 shows the most excellent feature in sub-chamber design.

Figure 10 presents mass fraction burned with crank angle (CA) and figure 11 expresses this as distribution by stages, respectively. From Figure 10, it can be shown that the gradient aspect of mass fraction burned is steeper due to the rapid combustion progress in the case of RI type B-N_h4. Also, as shown in Figure 11, all the burning sections in the cases of RI type A-N_h4 and B-N_h4 are slightly reduced compared with that of A-N_h8 under Φ=1.0 and Φ=0.8. This shows that the difference of flame development stage governs the overall burning duration as in Figure 7, which becomes marked under Φ=0.8. In

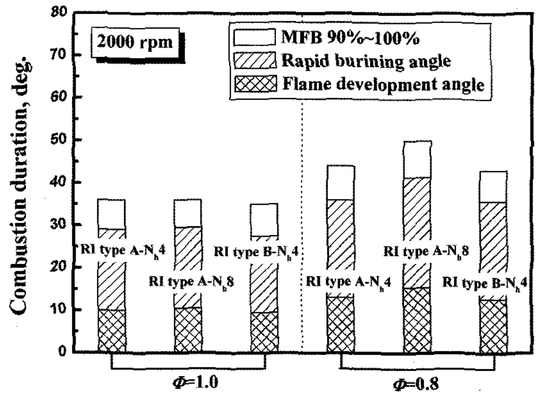


Figure 11. Each section in a whole combustion duration with equivalence ratio.

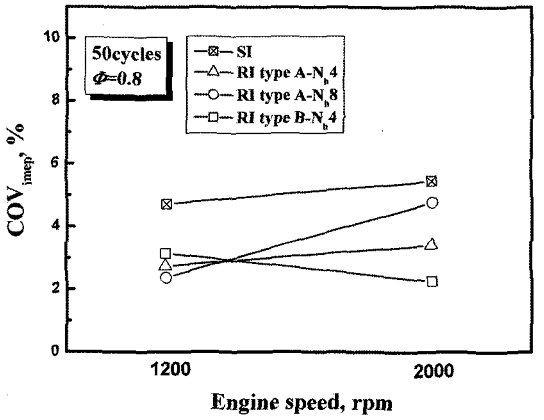


Figure 12. Variation of COV_{imep} for each engine.

addition, each overall burning angle in the cases of RI type A-N_h4, A-N_h8, and B-N_h4 is reduced by 24.2%, 13.8%, and 25.9% in comparison with that of SI engine under Φ=0.8.

Figure 12 shows COV_{imep} in the lean range at 1200 rpm and 2000 rpm, respectively. COV (Coefficient of variation in indicated mean effective pressure) is generally used to understand the cycle variation of the engines. As shown in the figure, the cases of RI engine show lower values having about 2%–3% except for A-N_h8 than that of SI.

Figure 13 shows the decrement rate of brake power as an imep value of a power property when the condition of stoichiometric ratio shifts to lean range. The decrement rate of brake power of the RI (A-N_h4 : 14.29%, A-N_h8 : 17.74%, B-N_h4 : 9.84%) is smaller than that of the SI (18.33%), where it is considered that the case of B-N_h4 of the smallest decrement rate will remarkably expand the lean operation range of an engine.

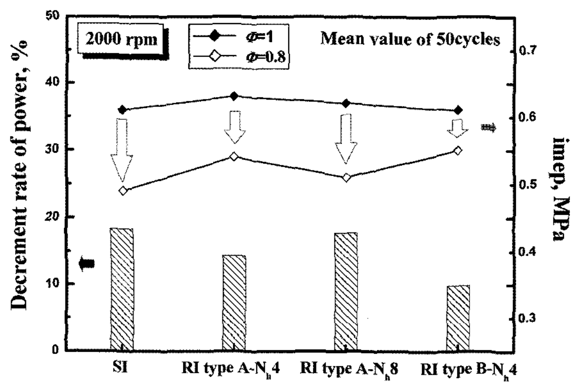


Figure 13. Decrement rate of power from Φ=1 to Φ=0.8.

4. CONCLUSIONS

The results of this experimental study can be summarized as follows:

(1) The cylinder maximum pressure (P_{max}) obtained by RI engine under Φ=1.0 is slightly higher than that by the SI engine, and this difference greatly appears under Φ=0.8 in spite of the relative delay of MBT. Also, though the cases of A-N_h4 and B-N_h4 have the same delayed spark timing under Φ=0.8, the P_{max} of the cases is higher than that of the A-N_h8. Besides, the combustion duration and fluctuation are wholly reduced for RI as compared with those for SI regardless of change in the equivalence ratio and rpm.

(2) The combustion characteristics of the RI engine under Φ=0.8 is partly the same as that of SI engine under Φ=1.0. It seems reasonable to conclude that the development of initial flame by RI method in a engine has the similar influence on the next section and the overall burning as in the constant volume chamber.

(3) The combustion of RI engine proceeds differently with the shape of ground electrode in the same sub-chamber, and type B-N_h4 in RI engines shows the most excellent behavior characteristics in the viewpoint of rapid combustion and burning stability. COV_{imep} in RI engines show lower values having about 2%–3% except for A-N_h8 than that in SI engine at 1200 pm and 2000 rpm.

(4) When the condition of stoichiometric ratio shifts to lean range, the case of RI type B-N_h4 has the smallest decrement rate of brake power (SI : 18.33%, RI type A-N_h4 : 14.29%, A-N_h8 : 17.74%, B-N_h4 : 9.84%).

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REFERENCES

Aoyama, T., Hattori, Y., Mizuta, J. and Sato, Y. (1996).

An experimental study on a premixed-charge compression ignition gasoline engine. *SAE Paper No.* 960081.

Christensen, M., Hultqvist, A. and Johansson, B. (1999). Demonstrating the multi fuel capability of a homogeneous charge compression ignition engine with variable compression ratio. *SAE Paper No.* 1999-01-3679.

Montagne, X. and Duret, P. (2001). What will be the future combustion and fuel-related technology challenges?. *IFP Int. Congress*, 175–179.

Ohyama, Y. (2001). Engine control using combustion model. *Int. J. Automotive Technology* **2**, **2**, 53–62.

Olsson, J-O., Erlandsson, O. and Johansson, B. (2000). Experiments and simulation of a six-cylinder homogeneous charge compression ignition (HCCI) engine. *SAE Paper No.* 2000-0102876.

Park, J. S., Choi, H. S., Lee, M. J., Lee, T. W., Ha, J. Y. and Chung, S. S. (2002). A fundamental study to improve the ignitability of lean mixture by radicals induced injection in a constant volume combustor. *FISITA,* Helsinki.

Park, J. S., Lee, T. W., Ha, J. Y. and Chung, S. S. (2004). A study on the combustion characteristics of lean mixture by radicals induced injection in a constant volume combustor (1). *Trans. Korean Society of Automotive Engineers* **12**, **2**, 45–53.

Park, J. S., Kang, B. M., Lee, M. J., Ha, J. Y. and Chung, S. S. (2004). A study on the combustion characteristics of lean mixture by radicals induced injection in a constant volume combustor (2). *Trans. Korean Society of Automotive Engineers* **12**, **3**, 19–26.

Pascal, H. (1999). New combustion concept for internal combustion engines. *Proc. the 15th Internal Combustion Engine Symp., Korea, No.* 9935761.