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COMPARISON OF THE COMBUSTION CHARACTERISTICS BETWEEN S.I. ENGINE AND R.I. ENGINE

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ABSTRACT–This experimental study was carried out to obtain both low emissions and high thermal efficiency by rapid bulk combustion. Two kinds of experiments were conducted to obtain fundamental data on the operation of a RI engine by a radical ignition method. First, the basic experiments were conducted to confirm rapid bulk combustion by using a radical ignition method in a constant volume chamber (CVC). In this experiment, the combustion velocity was much higher than that of a conventional method. Next, to investigate the desirable condition of engine operation using radical ignition, an applied experiment was conducted in an actual engine based on the basic experiment results obtained from CVC condition. A sub-chamber-type diesel engine was reconstructed using a SPI type engine with controlled injection duration and spark timing, and finally, converted to a RI engine. In this study, the operation characteristics of the RI engine were examined according to the sub-chamber's specifications such as the sub-chamber volume and the diameter and number of passage holes. These experimental results showed that the RI engine operated successfully and was affected by the ratio of the passage hole area to the sub-chamber volume.

KEY WORDS : Ultra lean mixture, Lean burn, Radical, Passage hole area/sub-chamber volume ratio, Bulk combustion

1. INTRODUCTION

Many experimental studies on the lean mixture combustion of an internal combustion engine have been conducted (Lee, 1996; 1998). The results from these studies have been used to obtain the low emissions and high thermal efficiency in the internal combustion engines. The combustion of a lean mixture, however, does not always have good results: for example, a slow combustion velocity adversely affects the engine's emissions and efficiency. In relation, some specific techniques on lean mixture combustion have been developed and used in an actual engine such as the Lean-Burn and the GDI engine. Considering the regulations of Euro-V against emissions and carbon dioxide in the categories by the Kyoto protocol (Jeong, 1998), however, the rapid combustion technique of ultra lean mixture has become inevitable. The combustion in the main chamber starts from the ejection of a flame kernel created in the sub-chamber with spark discharge, and then, the flame kernel propagates across the cylinder to the wall of the combustion chamber. At the walls, the flame is extinguished because of low heat transfer and destruction of active chemical species (Kito *et al.*, 2000). In these combustion processes of a hydrocarbon mixture, chemical reactions are generated due to the rising temperature and pressure by the spark discharge energy, up to TDC, and radicals such as OH, CH, H, and C_2 participate in these processes as the principal factors. These radicals, in the combustion process, are essential factors and activated chemical species that lead to elementary reactions. If the radicals are generated by the initiative chain reaction in the combustion process, the numbers of radicals rapidly increase because other elementary reactions are actively induced (Hirano, 1986; Park *et al.*, 2001). From the above results, the mixture is changed to the combustion products of high enthalpy. If the radical seeding method can be applied to hydrocarbon mixture combustion, simultaneous combustion can be obtained (Park *et al.*, 2005; 2006). Since a method using combustion is expected to produce rapid combustion and high thermal efficiency, a radical ignition engine that uses a method such as the radical seeding method as mentioned above was developed by Pascal (1999), who obtained better results for emissions and efficiency than the expected values. In this experimental study, the radical ignition method was applied to develop a clean and high efficiency engine. Consequently, the fundamental experiments to obtain data on the radical ignition combustion were conducted in a constant volume combustion chamber operated by an electrically controlled system. Moreover,

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to apply the data obtained from the fundamental experiments to an actual engine, a test engine with a radical ignition system was constructed. In the analysis of the combustion characteristics of the engine with a sub-chamber, the volume of the sub-chamber, and the number and diameter of the passage holes were selected as experimental parameters in this experiment.

2. EXPERIMENTAL APPARATUS AND TECHNIQUE

2.1. Fundamental Experiments in a Constant Volume Combustion Chamber

Figure 1 shows the schematic diagram of apparatus constructed for the rapid bulk combustion experiment. It consists of a constant volume combustion chamber with a main chamber and a sub-chamber, intake and exhaust devices, heating devices, electronic control system, pressure measurement system, visualization system.

To produce the rapid combustion of the premixture by radical ignition, a constant volume chamber, as shown in Figure 2, is used. The combustion chamber is divided into the main chamber and the sub-chamber. The volume of main chamber is 487cc and the chamber has quartz glass windows, which allows the luminosity of the flame to be photographed. The sub-chamber is placed onto the upper part of the main chamber. The sub-chamber volume and the number and diameter of the passage holes are used as the experimental parameters. In this experiment, the 4cc sub-chamber and twelve passage holes of 1.8 mm in diameter are used to obtain the fundamental data on the radical ignition method. The fuel injectors of the GDI engines are used to form the mixture inside the sub and main chambers. As the ignition occurs in the sub-chamber by spark discharge, burned and unburned gases, which include many radicals, are injected into the main chamber; the simultaneous combustion of the mixture then occurs.

Table 1 shows experimental conditions for the fundamental experiment with CVC. N-heptane (C_7H_{16}) was used as the test fuel and the initial pressure and ambient temperature were fixed at 0.5 MPa and 403 K, respectively. In this study, the effects of the diameter of the passage holes and the volume of a sub-chamber on the combustion characteristics were investigated.

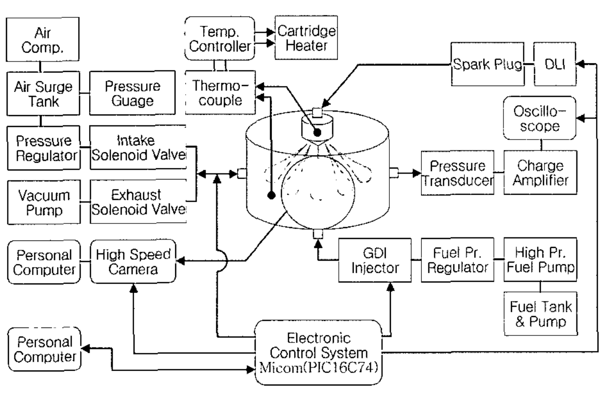


Figure 1. Schematic diagram of experimental apparatus with a constant volume combustion chamber.

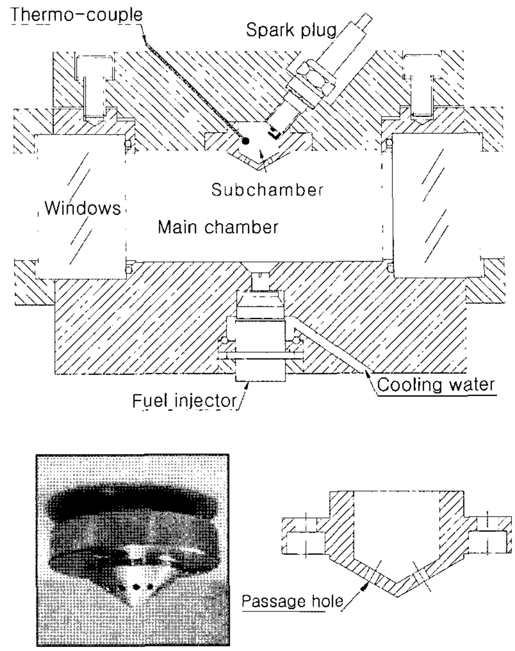


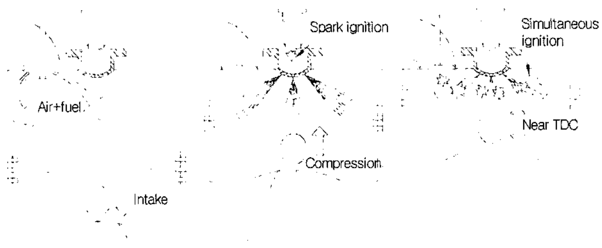
Figure 2. Cross sectional view of the constant volume chamber and photography of sub-chamber.

Table 1. Experimental conditions in a CVC.

Initial pressure (MPa)		0.5
Initial temperature (K)		403
Equivalence ratio (ER)		0.55~1.0
Diameter of passage holes (mm)		1.0~2.4
Volume of combustion chamber	Main (cc)	487
	Sub (cc)	2.2, 4, 7

2.2. Description of the RI Engine Concept

The combustion chamber of the RI engine is divided into a sub-chamber and a main chamber like an indirect injection diesel engine with a pre combustion chamber. The sub-chamber is located over the main chamber. The premixture flowed from the main chamber enters into the sub-chamber and is ignited by spark discharge. Since the pressure of the working gas in the sub-chamber increases with the flame development, the gas including the many radicals is injected into the main chamber through the passage holes. The working principle of the engine with



(a) (b) (c)

Figure 3. Schematic diagram of the operation principle of an engine using radical ignition method.

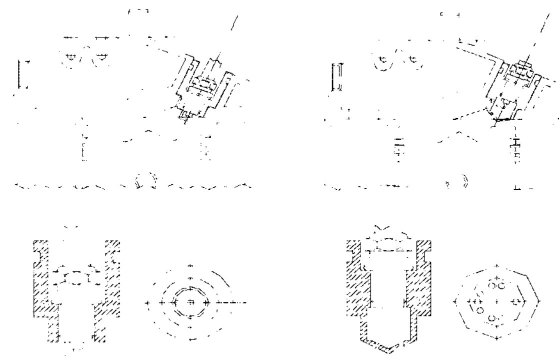
the radical ignition system is shown in Figure 3. As an SPI (Single Point Injection) type engine, the fuel injected into the intake manifold is lead by air to the main chamber during the intake stroke, shown in Figure 3(a). The premixture flows to the sub-chamber during the compression stroke and is ignited by the spark discharge near the TDC, as shown in Figure 3(b). Burned and unburned gases, which contain many radicals, are injected into the main chamber by the pressure rise in the sub-chamber, and then simultaneous combustion causes the multi point ignition of the premixture in the main chamber at the same time, as shown in Figure 3(c). The engine piston is forced down to BDC by the pressure increase due to the bulk combustion.

2.3. Configuration of the RI Engine

The RI engine is based on the indirect injection diesel engine with a single cylinder, and the specifications are shown in Table 2. The base engine was modified into an SPI type engine for the experiment, and the modified SPI engine was reconstructed into a RI type engine. The GDI injector, throttle valve, and air flow sensor were installed in a engine manifold. The compression ratio was 7.7 in the RI engine. There were only structural differences in

Table 2. Specifications of the test engine.

Specification	Test engine
Based engine	4 stroke, IDI diesel engine with a single cylinder
Bore (mm) and stroke (mm)	92×95
Displacement (cc)	623
Compression ratio	19
Maximum output (PS/rpm)	11/2200
Intake valve open & close	BTDC 20° ATDC 224°
Exhaust valve open & close	ATDC 136° ATDC 20°



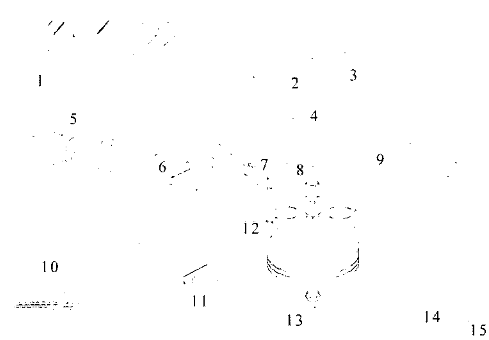
(a) SPI engine (b) RI engine

Figure 4. Schematic diagram of SPI and RI engine combustion chambers.

the combustion chamber between the SPI and the RI engine. RI engine has a sub-chamber with eight passage holes, which is set over the main combustion chamber and offset by about 30 mm from the central-axis of a piston, because the spark plug is installed at the fuel injector position of the existing diesel engine. Therefore, a knock may occurs by this geometry of the RI engine. The types of combustion chamber of the SPI and the RI engine are shown in Figure 4. In order to examine the effect of the geometrical shape of the sub-chamber on combustion characteristics, two kinds of sub-chamber of 2.2cc and 4cc volumes, respectively, were used in this experiment.

2.4. Experimental System in an Actual Engine

The main object of this study is to examine the feasibility of operation of a conceptual RI engine, as explained in Figure 3. In the experiment, the p-θ diagrams were obtain-



1. Controller	2. Fuel pump	3. Fuel tank
4. CDI ignitor	5. Air flow sensor	6. Throttle valve
7. Injector	8. Spark plug	9. λ sensor
10. PC	11. Amplifier	12. Pressure transducer
13. Engine	14. Dynamometer	15. Encoder

Figure 5. Schematic diagram of experimental apparatus with a modified diesel engine.

ed mainly from SPI and RI engine and were compared with each other. Specifically, the p-θ diagrams were obtained by changing the volume and passage hole diameter of the sub-chamber in an SPI and a RI engine. To obtain fundamental data, the amount of fuel injection per cycle was fixed at 17.77 mg in the engine experiments. As the engine speed increased, the equivalence ratio was reduced because of the increase in the sucked air quantity. The ignition timing and the degree of the throttle valve opening were controlled by the experimental system shown in Figure 5. In the experiment, the cooling water temperature was kept at a constant of 403 K. A dynamometer and a λ sensor were used in an actual engine. The combustion pressure of the in-cylinder was measured by a piezo-electric type pressure transducer and recorded in an oscilloscope.

3. RESULTS AND DISCUSSION

3.1. Experimental Results in CVC (Constant Volume Chamber)

Figure 6 shows the relationship between the combustion pressure and the time elapsed after spark discharge. In this experiment, n-heptane was used as the test fuel, and initial experimental conditions were fixed at 403 K and 0.5 MPa. The SI indicates that the spark plug alone was used for the combustion of mixture without the sub-

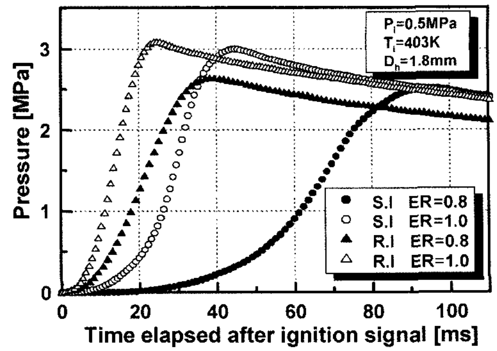


Figure 6. Comparison of combustion pressure characteristics between RI and SI method.

chamber. The RI means the case using the RI method, and an equivalence ratio was expressed as the ER, respectively. Assuming that the combustion finished at the maximum combustion pressure in Figure 6, as reported in the previous study (Park *et al.*, 2002), the whole combustion period of the RI is shortened 45% and 52% more compared with that of SI because of the simultaneous combustion due to the radical seeding.

Figure 7 shows a combustion pressure diagram in the RI method in the case of the passage hole diameters (D_h) of 1.0 mm~2.4 mm. The period from the rising point of pressure to P_{max} is shorter than that of the SI method under

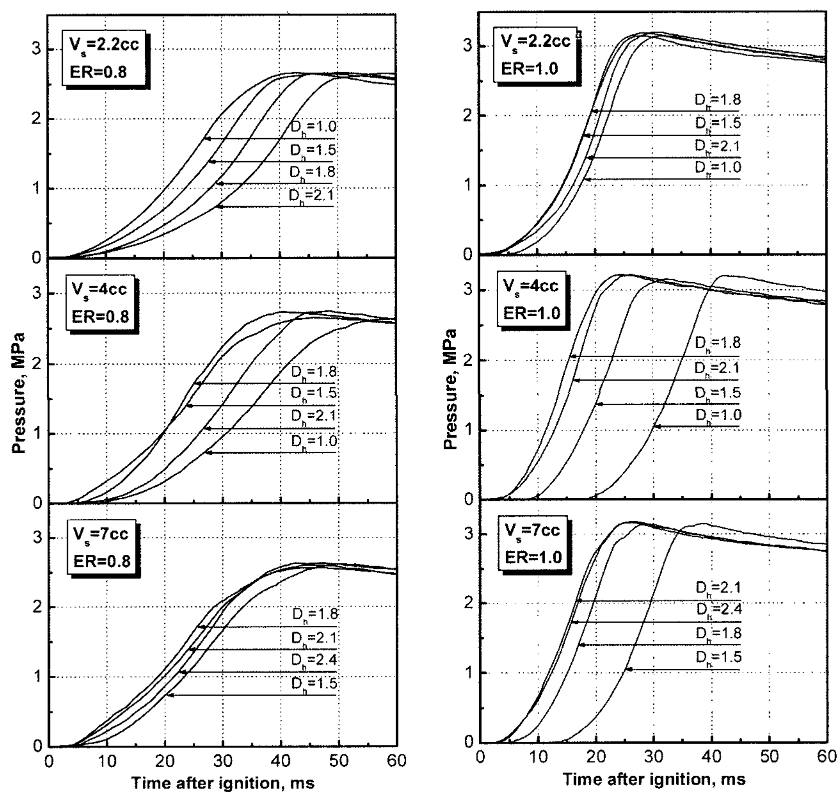


Figure 7. Characteristics of combustion pressure with diameter of passage hole.

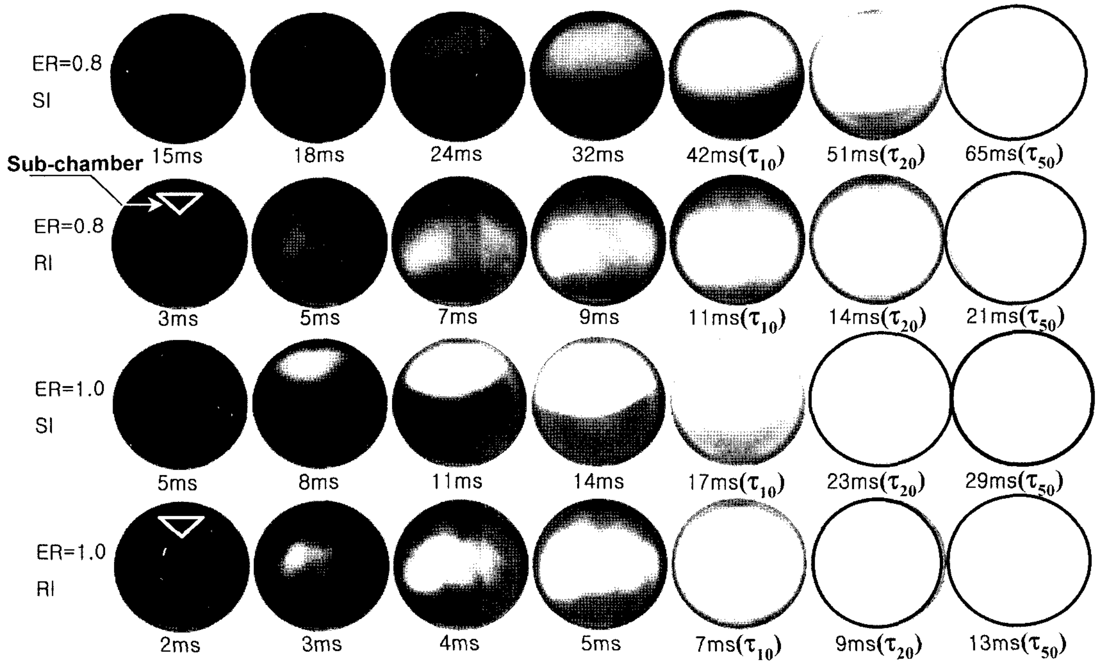


Figure 8. Comparison of flame visualization with SI and RI.

the equivalence ratio of 0.8 and a sub-chamber volume of 4cc. The combustion period is similar between the two methods for D_h=1.5 mm~2.1 mm except for D_h=1.0 mm. In the case of D_h=1.0 mm, the combustion period is shorter than that of the SI method, but such a tendency is clearer than in cases of other diameters. This is the reason why the case of D_h=1.0 mm has the demerits such as the insufficient quantity of burned gases due to the throttle loss, the heat loss at the chamber wall, and the cooling loss by long mixing time. The ignition delay is long in the case of D_h=2.1 mm. This means that the case becomes similar to that of a typical SI method, as the diameter of the sub-chamber increases.

Figure 8 shows the photographs of flame luminosity taken by SI and RI for the same conditions shown in Figure 6. In Figure 8, the combustion velocity is defined by the flame shapes in each image. As shown in Figures 6 and 8, the rapid combustion of the mixture is obtained by the radical ignition. In the case of the SI method, a flame propagates to the front with the laminar flame of a smooth spherical shape. In the case of the RI method, a strong turbulent flow is formed during the early combustion period by the injecting radicals when the flame jets through the passage holes from the sub-chamber to the main chamber, and the combustion is continued to the end by the jetted flame in the main chamber. In the RI method, the direction of flame propagation is opposite that in the SI method and spreads rapidly from seeded points of the bottom to the top. A well-dispersed burning zone of explosion and reaction should be formed because

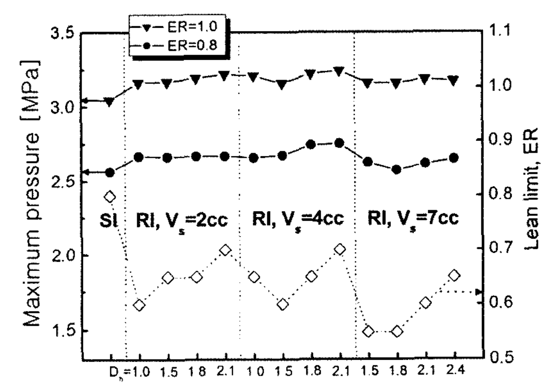


Figure 9. Effects of hole diameter and sub-chamber volume on maximum pressure and lean limit.

the burned gases, which include many radicals, are dispersed from the sub-chamber to the whole area of the main chamber, and the zone results in high rising rate of pressure and low heat loss by rapid combustion.

Figure 9 shows the effect of sub-chamber volume and passage hole diameter on P_{max} and the lean limit. As shown this figure, the equivalence ratio is extended about 0.15~0.2 in the lean limit of the RI method. Because the heat loss and pressure decrease due to the increase of pressure and the decrease of flame propagation distance in the sub-chamber, combustion and heat utilization efficiencies increase in the sub-chamber. Therefore, the energy of the combustion products jetted from the sub-chamber becomes concentrated at local, premixed areas of the

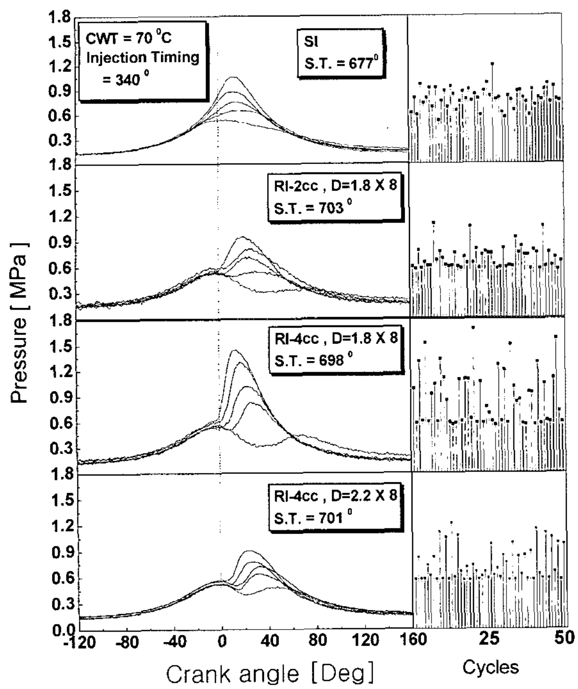


Figure 10. p-θ and comb-type diagrams at 900 rpm.

main chamber.

3.2. Experimental Results in an Actual Engine

The p-θ and comb-type diagrams (Iida, 1993) of the SPI and the RI engine are shown in Figure 10. The C.W.T. and S.T. represent the coolant water temperature and the spark discharge timing, respectively. Gasoline was used as the experiment fuel, and the ignition timing was controlled to obtain the maximum speed at fixed conditions of injection quantity and degree of throttle valve opening. The sub-chamber volume and the number of passage holes were changed, as shown in each figure. In Figure 10, the comb-type diagrams were continuously measured fifty times, and the p-θ diagrams were randomly selected from these diagrams. Although the degree of fluctuation of combustion in the comb-type diagrams is not small, because the swept volume is large and the compression ratio is low comparatively, the p-θ diagrams of the SI method show a combustion phenomenon typical of SPI engines. A base diesel engine shown in Table 2 was successfully converted into a SPI type engine. However, unlike the SI, the RI occasionally has an irregular combustion phenomenon. The appearance point of the maximum pressure on p-θ diagrams is not regular, and the fluctuation degree on comb-type diagrams is not little also. Nevertheless, since the RI combustion characteristics varied due to the change in the sub-chamber volume or diameter, it can be speculated that the RI method is significantly affected by sub-chamber design like the sub-chamber volume and diameter. Therefore, the undesirable factors must be improved to obtain the optimum design of the sub-chamber.

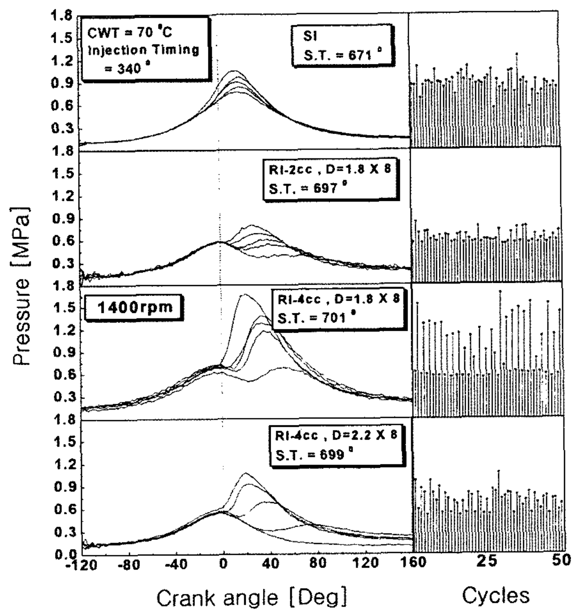


Figure 11. p-θ and comb-type diagrams at 1600 rpm.

The feasibility of using the radical ignition in the RI engine was verified by the results shown in Figure 10. In the case of d_h=1.8 mm in the RI, the combustion pressure for the sub-chamber volume of 2cc fluctuates very differently from that of 4cc. The pressure fluctuation in case of the 4cc is more remarkable than that of the 2cc. From the above result, the A/V ratio (the total area of passage holes to sub-chamber volume) is a very important factor in the design of a RI engine. In both cases of 2cc and 4cc, the A/V ratios were calculated to be 0.1/cm and 0.05/cm, respectively. The fluctuation degree of the A/V ratio of 0.05/cm was larger than that of the A/V ratio of 0.1/cm because the scavenging process of the exhaust gas by the induction of a new mixture was not sufficient in the sub-chamber.

In the Figure 11, to set the engine speed to 1,600 rpm, only the throttle valve was adjusted to the condition in the Figure 10, and therefore, the equivalence ratio in Figure 11 was lower than that in the Figure 10. However, in the sub-chamber of 4cc and passage holes of d_h=1.8 mm in diameter, particularly, the speed of the RI engine cannot reach 1,600 rpm due to the A/V ratio in the sub-chamber. Though the mixture in the sub-chamber has a ignitable range with spark discharge, the mixture is not sufficient for ignition because of the small A/V ratio. The combustion process of the engine is irregular in the comb-type diagram. Such a combustion phenomenon is in contrast with that of the case in which the passage hole diameter is 2.2 mm, because, the physical and chemical factors

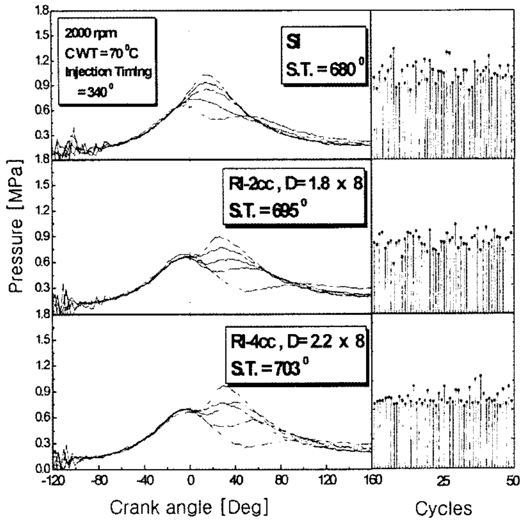


Figure 12. p-θ and comb-type diagrams at 2000 rpm.

injected into the main chamber in each cycle change. As stated above, since the A/V ratio is a considerable factor in a RI engine, it will be necessary to analyze the A/V design in the future.

Figure 12 shows the p-θ and the comb-type diagrams. The experimental conditions are fixed at 2,000 rpm, d_h= 1.8 mm and 2.2 mm in passage hole diameter, 2.2cc and 4cc in sub-chamber volume, respectively. Under these conditions, particularly the stable comb-type diagrams were obtained to compare with those of the previous conditions (Figs. 10 and 11) because scavenging of the sub-chamber is gradually improved by the increase in volumetric efficiency with the degree of opening of the throttle valve. Therefore, a value of the optimized A/V ratio is needed for the RI engine design.

4. CONCLUSIONS

In order to obtain a RI engine of low emission and high thermal efficiency, a radical ignition method was introduced for the rapid combustion of lean mixture as a bulk combustion in CVC. Also, the experiment of the RI engine with the radical ignition method in an actual engine was performed. The obtained results are as follows:

(1) The rapid combustion of hydrocarbon fuel mixture can be realized by using the RI method in a constant volume chamber and an actual engine.
(2) At the equivalence ratio of 1.0, the RI method reduced the combustion period by a maximum of 50%, which was more than that by the SI method. It was more effective in lean mixture combustion. Therefore, the application of the RI method to engines can extend the combustible lean limit of the mixture.
(3) The combustion characteristics shown in RI method were significantly affected by the geometric characteristics of the sub-chamber such as the volume and the diameter.
(4) The test engine using the RI method was successfully operated with the change of engine speed, although the engine had occasional irregular combustion pressures at low speeds compared with that of the SI engine.
(5) The A/V ratio of the sub-chamber is the most significant factor of the optimum design of a RI engine.

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