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EXPERIMENTAL STUDY ON HOMOGENEOUS CHARGE COMPRESSION IGNITION ENGINE OPERATION WITH EXHAUST GAS RECIRCULATION

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ABSTRACT–This paper is concerned with the Homogeneous Charge Compression Ignition (HCCI) engine as a new concept in engines and a power source for future automotive applications. Essentially a combination of spark ignition and compression ignition engines, the HCCI engine exhibits low NOx and Particulate Matter (PM) emissions as well as high efficiency under part load. The objective of this research is to determine the effects of Exhaust Gas Recirculation (EGR) rate on the combustion processes of HCCI. For this purpose, a 4-cylinder, compression ignition engine was converted into a HCCI engine, and a heating device was installed to raise the temperature of the intake air and also to make it more consistent. In addition, a pressure sensor was inserted into each of the cylinders to investigate the differences in characteristics among the cylinders.

KEY WORDS : Equivalence ratio, EGR (Exhaust Gas Recirculation), Fuel flow rate, Combustion efficiency, Rate of heat release, IMEP (Indicated Mean Effective Pressure)

1. INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) engine combines the combustion characteristics of a compression ignition engine and a spark ignition engine. HCCI engines take advantage of the high compression ratio and heat release rate and thus exhibit high efficiency found in compression ignition engines. Also, due to the lean air/fuel ratio, HCCI engines produce low NOx and Particulate Matter (PM) emissions even without the use of after treatment devices. Because HCCI combustion is not the result of a single spark, a traditional flame front does not exist, and in general, a localized high-temperature area is not present (Campbell *et al.*, 1999; Christensen *et al.*, 1998; Akagawa *et al.*, 1999).

Najt *et al.* (1983) produced a model based on the Cooperative Fuel Research (CFR) engine and called it Compression-Ignited Homogeneous Charge (CIHC) combustion.

HCCI combustion, in general, results in a high heat release rate; through appropriate timing, it can approach an ideal Otto cycle. Distributed low-temperature reaction and non-luminous combustion reduce the amount of lost heat, and HCCI combustion therefore has a higher thermodynamic efficiency (Canakci *et al.*, 2003; Mogi *et al.*, 1999; Shimazaki *et al.*, 1999).

Aoyama *et al.* (1996) have conducted a visual study of HCCI. They concluded that HCCI combustion is initiated simultaneously in different parts of the combustion chamber, and a clear flame front does not exist. Even without a flame front, they argued that HCCI heat release rate is not simply a function of the chemical reaction rate, but it depends on pressure, temperature, and species concentration in the cylinder.

Another big drawback of the HCCI engine is the increase in HC and CO emissions. One of the reasons for such increase in HC and CO emissions is that the increase in EGR rate required for HCCI reduces the cylinder temperature. It is already well known that temperature reduction within the cylinder leads to lower post-combustion oxidation rate and higher HC and CO emissions (Au *et al.*, 2001; Flowers *et al.*, 2000; Odaka *et al.*, 1999).

Another important aspect of HCCI research is the control of combustion phases. Unlike a spark ignition or a diesel engine, there is currently no method for controlling the start of combustion (Lanzafame *et al.*, 2003). Instead, combustion is started by auto-ignition of the air/fuel

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mixture. However, the mixture's auto-ignition is affected by its properties and time/temperature history.

In their research, Christensen *et al.* (1999) proposed that the combustion phases of the HCCI engine are affected by the following: auto-ignition properties, fuel concentration, residual rate, residual reactivity, mixture homogeneity, compression ratio, intake temperature, latent heat of the fuel, engine temperature, heat transfer to the engine, and various other parameters. In order to solve the problems of HCCI, a new HCCI engine should be developed, but for now, numerical and experimental studies show that HCCI combustion (Au *et al.*, 2001; Flowers *et al.*, 2000; Odaka *et al.*, 1999) is possible.

Some researchers (Morimoto *et al.*, 2001; Stanglmaier *et al.*, 1999) have proposed that HCCI engines have lower NOx reduction properties compared to spark ignition engines. In the future, through the development of a HCCI fuel and advancements in the control of combustion phases, an air/fuel ratio and an increase in power similar to those of diesel engines are expected. For now, HCCI combustion applicable for automotive use should utilize a dual-mode engine. In other words, HCCI combustion would be used in low loads to increase fuel efficiency and emission reduction properties, and spark ignition or diesel combustion would be used under high loads to solve the power problem. In engines used for low load and part load conditions, HCCI combustion would be very useful.

Through the use of high air/fuel ratio and/or EGR, a high dilution state can be achieved. If the mixture is very rich, the combustion rate can become too high and cause problems such as knocking. On the other hand, if the fuel is too lean, incomplete combustion or misfire can occur. For the SI engine, mixture inhomogeneity around the spark plug and variations in flow rate can cause significant changes in the early development of the flame front, resulting in large cycle-to-cycle variations.

Ryan *et al.* (1996) proposed that while both 2-stroke and 4-stroke engines can be used, the operating conditions are restricted by the need for a mixture that can reduce the combustion rate. Consequently, 2-stroke engines must have rich residual gases under part load, and 4-stroke engines, which generally do not have large amounts of residual gas, must make use of an ultra-lean mixture or large EGR fractions. HCCI combustion is achieved by compressing the mixture of air, fuel, and recirculated gas before auto-ignition. In general, HCCI combustion is a homogeneous and rather rapid low-temperature process.

Morimoto *et al.* (2001) stated that, in general, combustion timing of HCCI depends on inlet charge temperature, intake manifold pressure, type of fuel, compression ratio, equivalence ratio, EGR rate, engine speed, and engine load.

As many of the researchers above mentioned, much research regarding the relationship between EGR and HCCI is needed. Therefore, the purpose of this research is to determine how greatly the EGR rate affects the HCCI combustion process. Also, the effects of the EGR rate on NOx and PM reductions as well as on improving efficiency and various other properties will be investigated. Data from this study should be a valuable source of information in the development of the HCCI engine.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

The engine's specifications are listed in Table 1.

For the experiment, a 4-cylinder Volkswagen Turbo Direct Injection engine was modified accordingly. The schematic diagram of the experimental engine is shown in Figure 1.

Figure 2 shows a photo of the experimental setup. A Turbo Charged (TC) engine was modified into a Naturally Aspirated (NA) engine, and a 18 kW preheat device was used to obtain an intake charge.

Energy required for preheating was not included in the efficiency calculations. A 30 kW dynamometer was used to control and measure the load applied to the engine, and thermocouples were inserted into the intake pipe, exhaust pipe, coolant, engine oil, and various other parts to measure the temperature. The experimental engine was

Table 1. Engine specifications.

Item	Specification
Displacement	1.896L
Bore	79.5 mm
Stroke	95.5 mm
Connecting rod length	144.0 mm
Compression ratio	18.8:1
Piston geometry	Bowl

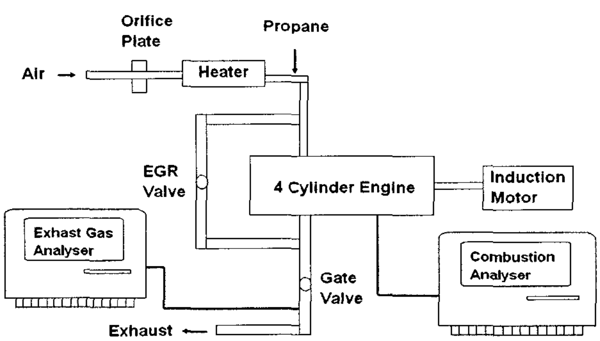

Figure 1. Schematic diagram of experimental apparatus.

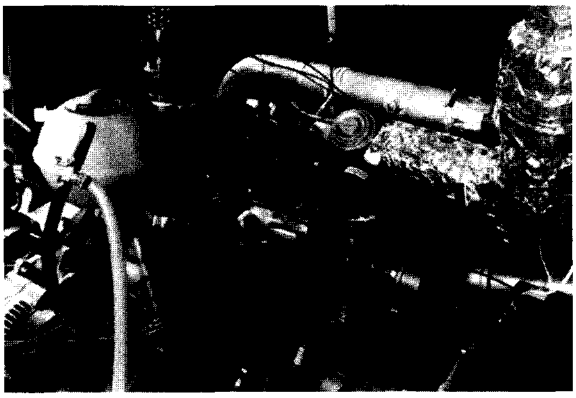

Figure 2. Photo of experimental setup.

run at 1800 RPM.

For data measurements, a crank angle encoder was used to measure the crank angle, and a pressure sensor was used in each cylinder to measure the pressure. Pressure values were taken at 0.1 degrees, and 160 cycles were used to obtain the average value. The acquired pressure values were used to determine burn rate, heat release rate, Indicated Mean Effective Pressure (IMEP), combustion characteristics, etc. Intake air flow was measured using a flow meter with a knife-edge orifice plate, and fuel flow was measured using a mass flow controller. Also, an exhaust gas analyzer was used to measure the amounts of unburned HC, oxygen, Carbon Monoxide (CO), and Nitric Oxide (NOx).

Generally, an EGR device recirculates part of the exhaust gas into the intake pipe in order to reduce NOx emissions. The recirculation amount is controlled by a needle valve located between the exhaust pipe and the intake pipe. EGR % can be calculated with the following equation.

$$\text{EGR rate } (\%)=[(V_1-V_2)/V_1]\times 100$$

Here, V_1 is the amount of intake air without EGR, and V_2 is the amount of intake air when the EGR valve is used to recirculate the exhaust gas.

Propane was used as fuel throughout the experiment.

Intake temperatures of 115°C and 145°C were used, and the fuel flow rate was fixed at 15 Standard Liters Per Minute (SLPM). Based on the experimental engine, explanations of engine performance, combustion process, exhaust gases, etc. are proposed. The HCCI engine is sensitive to temperature variations of the mixture. Also, misfire can occur in the presence of low load or low intake temperature, and knocking can take place when there is high load or high intake temperature.

In this experiment, the EGR rate was varied between 0 and 55% by volume. The residual gas in the exhaust is injected via a conventional EGR line. EGR raises the intake charge temperature; therefore, in order to maintain a constant intake air temperature, the inlet charge preheat was reduced when the EGR rate was increased. The intake temperature was kept constant (115°C or 145°C) regardless of the EGR rate.

3. EXPERIMENTAL RESULTS AND DISCUSSION

Figure 3 shows Start of Combustion (SOC) in crank angles upon varying the EGR rate from 0 to 43% at the constant intake temperature of 115°C. In this paper, the start of combustion is defined as the 50% point of the peak rate of heat release.

Figure 4 shows the SOC upon varying the EGR rate from 0 to 55% at the constant intake temperature of 145°C.

As Figure 3 and Figure 4 show, SOC is sensitive to intake temperature and EGR rate and varies significantly among cylinders. The variation among cylinders is especially large when the intake temperature is low, and

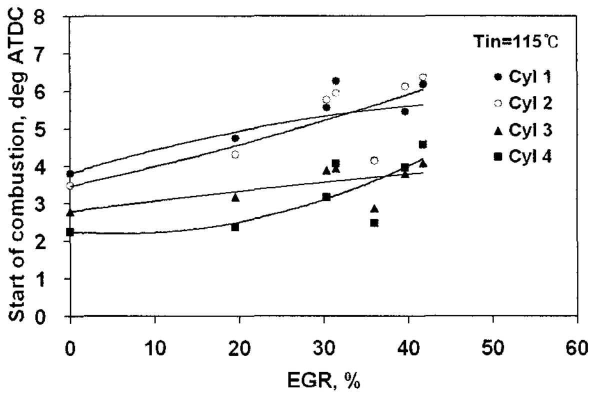

Figure 3. Start of combustion vs. EGR rate for different cylinders at intake temperature of 115°C.

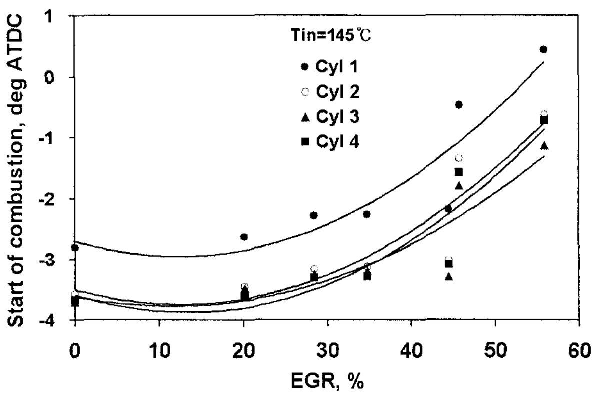

Figure 4. Start of combustion vs. EGR rate for different cylinders at intake temperature of 145°C.

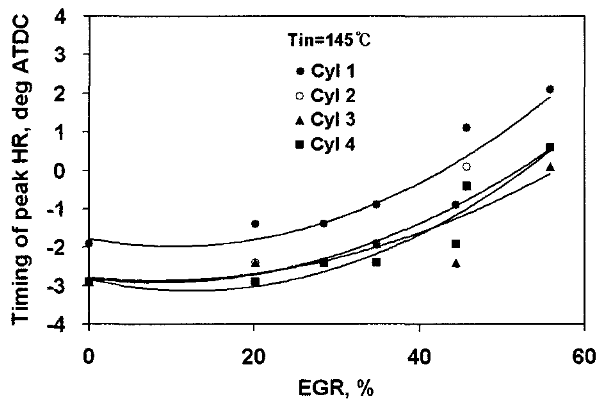

Figure 5. Timing of peak heat release vs. EGR rate for different cylinders at intake temperature of 145°C.

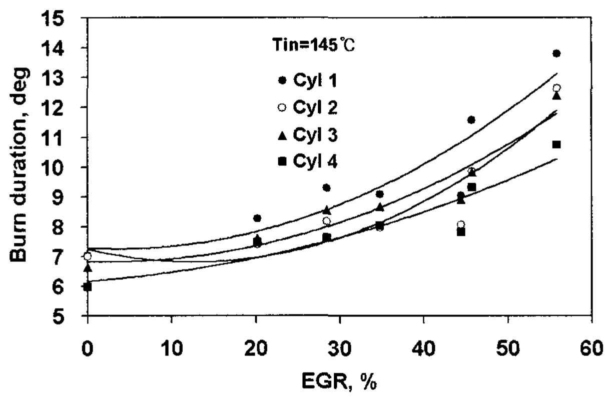

Figure 7. Burn duration vs. EGR rate for different cylinders at intake temperature of 145°C.

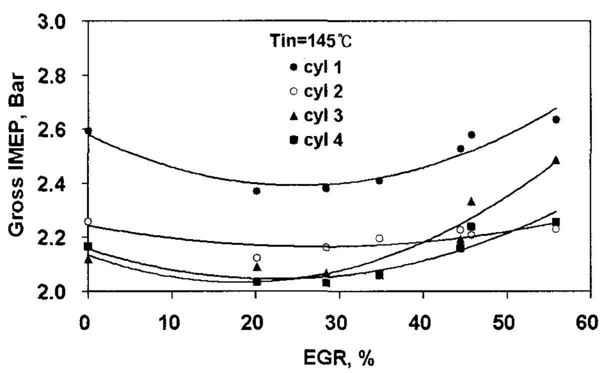

Figure 8. Gross IMEP vs. EGR rate for different cylinders at intake temperature of 145°C.

SOC occurs earlier as intake temperature is raised. For the intake temperature of 115°C, EGR rate of 43% is the maximum rate limits, and for 145°C, 55% is the maximum rate limits. In order for auto-ignition to occur in a HCCI engine, the intake air should be heated. In spark ignition and diesel engines, SOC is determined by ignition timing and injection timing, respectively. However, because ignition in a HCCI engine occurs as result of a spontaneous auto-ignition of homogeneous charge, there is no direct method for controlling SOC.

Figure 5 shows the plot of timing of peak heat release vs. EGR rate for different cylinders at the constant intake temperature of 145°C. In general, maximum heat release is delayed slightly as EGR is increased; the trend is very similar tó that of SOC mentioned previously. Therefore, Figure 3~Figure 5 show that SOC depends on EGR.

Figure 6 and Figure 7 show burn duration vs. EGR rate for different cylinders at the intake temperatures of 115°C and 145°C, respectively. Here, burn duration is defined as the number of crank angles the engine takes to complete 10~90% of heat release. In general, long burn duration indicates slow combustion and vice versa. Burn duration most likely increases with EGR rate in Figures 6 and 7 because chemical reaction is probably retarded due to the increase in EGR rate. Also, Figure 7 shows that when the intake temperature is at 145°C, the burn duration is relatively short. It can be seen that when using propane in a HCCI engine, high intake temperature is needed for a fast burn.

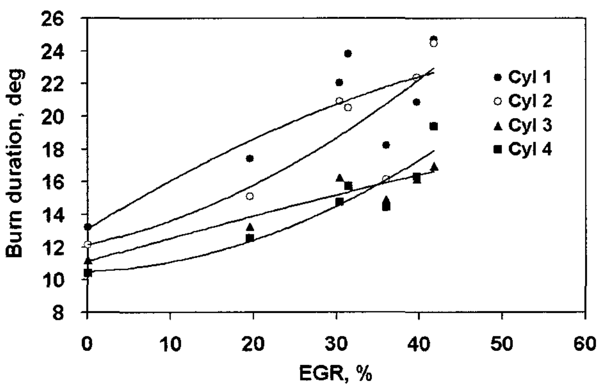

Figure 6. Burn duration vs. EGR rate for different cylinders at intake temperature of 115°C.

Figure 8 shows gross IMEP vs. EGR rate, and it is seen that gross IMEP is about 2.25 bar on average. Pressure values were taken at 0.1 degrees, and 160 cycles were used to obtain the average value.

Figure 9 shows combustion efficiency vs. EGR rate. At 145°C, combustion efficiency is around 95%, and at 115°C, it is around 85%. As for stability, the reduction of combustion efficiency signals instability above the EGR rate of 45% at the intake temperature of 145°C and above 35% at 115°C.

Generally, the most effective way of reducing NOx emissions is to hold combustion chamber temperature down. Although practical, this is a very unfortunate method in that it also reduces the thermal efficiency of

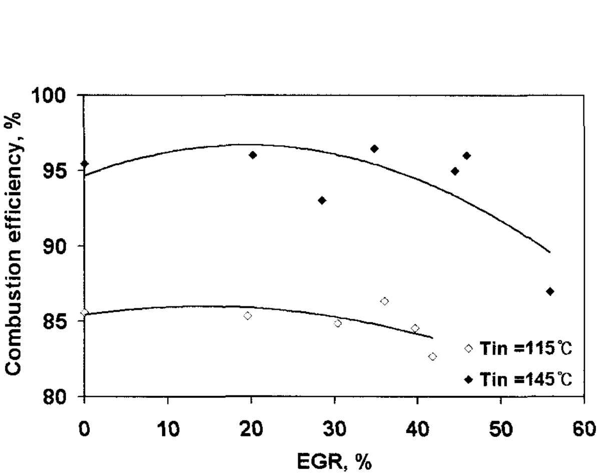

Figure 9. Combustion efficiency vs. EGR rate for different cylinders at intake temperatures of 115 and 145°C.

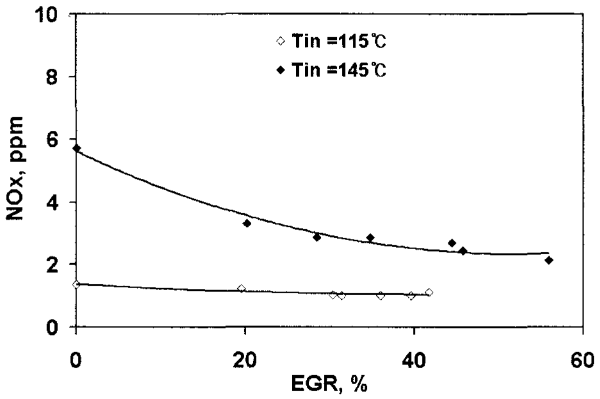

Figure 10. Nitric oxide emissions vs. EGR rate for different cylinders at intake temperatures of 115 and 145°C.

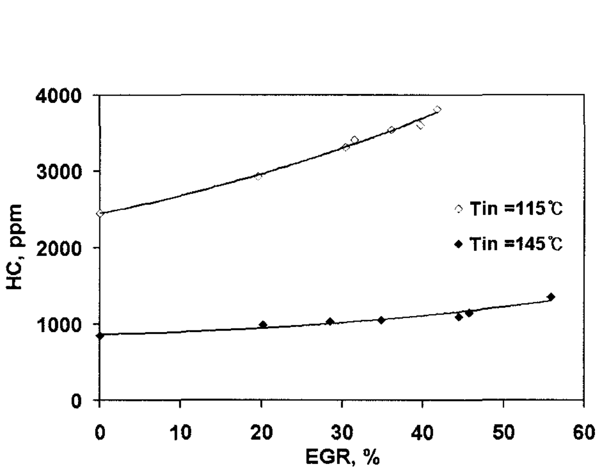

Figure 11. Unburned HC emissions vs. EGR rate at intake temperatures of 115 and 145°C.

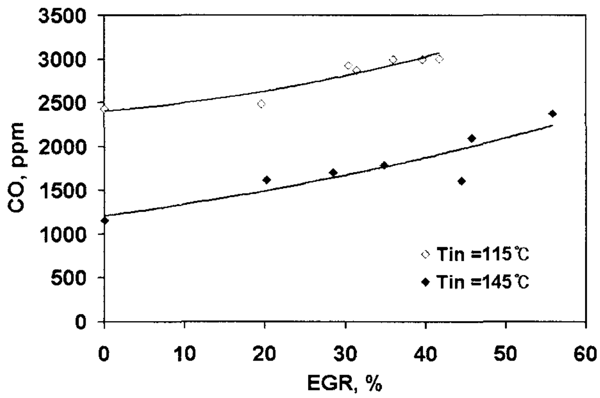

Figure 12. CO emissions vs. EGR rate at intake temperature 115 and 145°C.

the engine. Not only does EGR reduce the maximum temperature in the combustion chamber, but it also lowers the overall combustion efficiency. It could also be that the reduction in NOx is due to the use of EGR resulting in a reduction in oxygen concentration during the charge stage. Figure 10 shows the plots of nitric oxide emissions vs. EGR rate. Also, intake temperature at 115°C shows much less NOx emissions compared to those at 145°C. Even without the use of EGR, HCCI engine alone greatly reduces the quantitative amount of NOx to 1.2~5.8 ppm, a phenomenally low figure.

Figure 11 shows the experimental results of HC emissions vs. EGR rate for varying intake temperatures. The results show that the amount of HC emissions increase slightly with increasing EGR rate. This figure shows that when the intake temperature is at 145°C, the HC emissions are relatively lower than at 115°C.

Figure 12 shows CO emissions vs. EGR rate. The results look similar to those of HC emissions. It is known that a conventional spark ignition engine produces CO emissions of around 1200~2300 ppm. In this test, CO emissions at the intake temperature of 145°C was similar to those of a conventional gasoline engine, and at the low intake temperature of 115°C, the amount was relatively higher.

4. CONCLUSION

A 4-cylinder compression ignition engine has been modified into a HCCI engine. A new heating mechanism was added to increase the intake temperature, and in place of the fuel injector, a pressure transducer was inserted in each cylinder. Propane was used as fuel, and EGR was used. From this experiment, the following conclusions could be made.

The trend of SOC is sensitive to intake temperature, EGR rate, and cylinder-to-cylinder variation. Cylinder-to-cylinder variation was especially large for low intake temperatures, and as intake temperature increased, SOC occurred earlier. For the intake temperature of 115°C, EGR rate of 43% is the maximum, and for 145°C, 55% is the maximum.

By using EGR in the HCCI engine, NOx emissions are decreased even further due to the drop in combustion temperature. The effects of a homogeneous mixture and reduction in combustion temperature simultaneously contribute to a significant reduction in the amount of NOx emissions. Also, the intake temperature of 115°C shows much less NOx emissions compared to 145°C. Even without EGR, the HCCI engine would have significantly reduced the amount of NOx emissions.

However, as for HC and CO emissions, intake temperature must be raised to reduce these values.

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