

A Study of Ignition Characteristics of an HCCI Engine Operating on a Two-component Fuel

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ABSTRACT

The Homogenous Charge Compression Ignition (HCCI) engine is positioned as a next-generation internal combustion engine and has been the focus of extensive research in recent years to develop a practical system. One reason is that this new combustion system achieves lower fuel consumption and simultaneous reductions of nitrogen oxide (NO_x) and particulate matter (PM) emissions, which are major issues of internal combustion engines today. However, the characteristics of HCCI combustion can prevent suitable engine operation owing to the rapid combustion process that occurs accompanied by a steep pressure rise when the amount of fuel injected is increased to obtain higher power output. A major issue of HCCI is to control this rapid combustion so that the quantity of fuel injected can be increased for greater power. Controlling the ignition timing is also an issue because it is substantially influenced by the chemical reactions of the fuel. Various approaches are being researched for expanding the range of stable engine operation, including the application of turbocharging, the use of residual combustion gas, stratification of the fuel concentration, and the use of a blend of two types of fuel having widely different ignition characteristics.

This study focused on the use of a mixture of two fuels with different ignition characteristics. Experiments were conducted using a blend of n-heptane (0 RON) as a readily autoignitable fuel and iso-octane (100 RON) as a fuel non-conductive to autoignition. The results showed that this approach enables the amount of fuel injected to be increased while controlling the ignition timing.

INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) combustion has been widely researched in recent years for application to internal combustion engines in view of its potential for achieving both high efficiency and low exhaust emissions. This combustion process is seen as one way of meeting today's demands for the resolution of air pollution problems and the reduction of engine fuel consumption. The HCCI process facilitates combustion at a high compression ratio of a lean mixture exceeding the limits of flame propagation and is characterized by little inhomogeneity of the local fuel concentration in the cylinder. Because of these characteristics, the HCCI combustion process achieves high efficiency and can reduce nitrogen oxide (NO_x) and particulate matter (PM) emissions simultaneously to provide low levels of exhaust pollutants. However, HCCI engines do not have any physical means of inducing ignition like the spark plug used in spark ignition engines. Furthermore, it is difficult to control the ignition timing and combustion because they are greatly influenced by the chemical reactions of the mixture. Another issue of HCCI engines is the question of how to control the rapid nature of combustion[1,2,3,4,5,6,7].

As one approach to resolving these issues, we have focused on the use of a two-component fuel consisting of n-heptane and iso-octane, which have markedly different ignition characteristics. The results of our experiments conducted to date have shown that retarding the ignition timing is effective in inhibiting the rapidity of HCCI combustion[8]. It has also been confirmed that increasing the quantity of iso-octane injected while keeping the quantity of n-heptane constant tends to result in the same in-cylinder behavior until ignition occurs, even though the total amount of fuel injected is increased[9]. These results show the possibility of increasing

the engine load for greater power while controlling the ignition timing and restraining the rapidity of combustion by injecting a larger quantity of iso-octane and a constant amount of n-heptane. However, the specific details of the combustion process are still unclear. Therefore, to investigate the combustion process in detail, experiments were conducted in which exhaust gas recirculation (EGR) was applied for comparative purposes and an analysis was made of the data obtained by spectroscopic techniques. Chemical kinetic simulations were also performed and the results were compared with the experimental data.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

TEST ENGINE SPECIFICATIONS AND EXPERIMENTAL CONDITIONS

Currently, HCCI combustion is being widely researched using four-stroke test engines. However, the cylinder head of a four-stroke engine has a complex valvetrain system, making it difficult to obtain optical measurements inside the combustion chamber. Compared with a four-stroke engine, a two-stroke engine has a simpler cylinder head construction that allows greater freedom for making optical measurements in the combustion chamber and it is easier to apply EGR continuously. Since the purpose of the present study was to investigate HCCI combustion mechanisms on the basis of optical measurements, it was decided to use a two-stroke test engine. [Table 1](#) gives the specifications of the test engine used in this study. The engine was operated under a wide open throttle (WOT) condition at a constant speed of $N_e = 1000$ rpm. The test fuels used were n-heptane (0 RON) and iso-octane (100 RON). Two fuel injectors were installed in the intake pipe to facilitate independent variation of the injected quantities of n-heptane and iso-octane. [Table 2](#) shows the injected heat values of the fuels in each test case. Recirculated exhaust gas was throttled by an EGR control valve installed at the exhaust port exit.

Table 1. Specifications of test engine

2-stroke Air-cooled Single-cylinder Gasoline Engine	
Bore × Stroke	72 × 60 mm
Displacement	244 cm ³
Type of Scavenging Timing	Schnürle
Exhaust Port Closing Timing	106 deg. BTDC
Effective Compression Ratio	$\varepsilon = 10 : 1$

Table 2. Fuel injected heat values

Case No.	Fuel Injected Heat Value [J/cycle]		
	n-heptane, Bn	iso-octane, Bi	Total, Bt
Case 1	330	0	330
Case 2		50	380
Case 3		120	450
Case 4	280	50	330
Case 5		120	400
Case 6		190	470
Case 7	460	0	460

MEASURED ITEMS AND MEASUREMENT POSITIONS

The configuration of the test equipment is shown schematically in [Fig. 1](#), and the position in the cylinder head where absorbance was measured is shown in [Fig. 2](#). Typical items measured were the cylinder pressure P (MPa), intake air temperature T_{in} (K), scavenging temperature T_{sc} (K), exhaust gas temperature T_{ex} (K) and transmitted light intensity $E_{293.1}$ (V) at an absorption wavelength of 293.1 nm[10] corresponding to that of formaldehyde (HCHO), an intermediate combustion product characteristic of low-temperature oxidation reactions. The cylinder pressure was measured with a crystal pressure transducer fitted to the top of the cylinder head. The intake air temperature, scavenging temperature and exhaust gas temperature were measured with sheathed K-type thermocouples installed in the intake pipe, scavenging port and exhaust port exit, respectively. Two quartz observation windows were provided in the sides of the cylinder head as shown in [Fig. 2](#). Parallel light from a xenon lamp was irradiated through the window labeled “In” and transmitted light was extracted through the window labeled “Out” on the opposite side. The extracted light was introduced into a monochromator via an optical fiber cable having a core diameter of 1 mm and separated at the measurement wavelength. The separated light was converted to an electrical signal by a photomultiplier, the output voltage of which was regarded as the transmitted light intensity.

That value was converted to absorbance $A_{293.1}$ using [Eq. \(1\)](#) below where $E_{293.1(Avg)}$ (V) is the average transmitted light intensity from the onset of compression to just before the low-temperature oxidation reactions and $E_{293.1(\theta)}$ (V) is the transmitted light intensity at any arbitrary crank angle[7, 11]. It will be noted that the monochromator had a wavelength resolution of $\lambda = 4.0$ nm in terms of the half-width value.

$$A_{293.1} [-] = \frac{E_{293.1(Avg)} - E_{293.1(\theta)}}{E_{293.1(Avg)}} \quad (1)$$

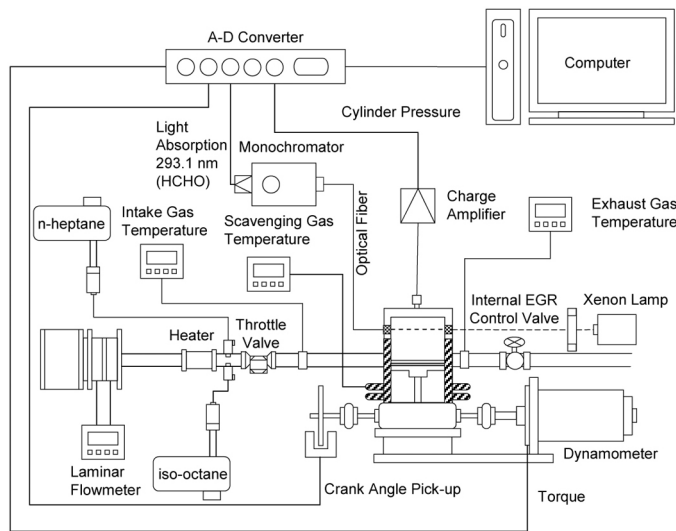


Fig. 1. Configuration of experimental equipment

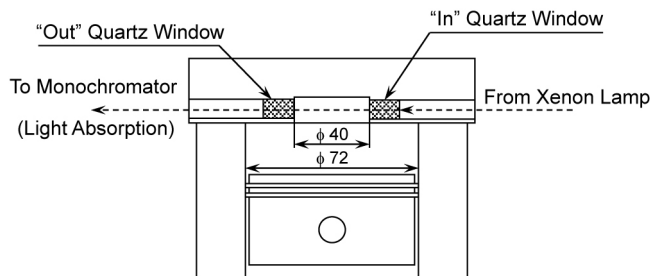


Fig. 2. Absorbance measurement position

DEFINITIONS OF TIME OF COOL FLAME OCCURRENCE, IGNITION TIMING AND IGNITION DELAY

In this study, the ignition timing θ_{IT} (deg.) was defined as the crank angle at which the heat release rate (HRR) reached 10% of the peak HRR (HRRMAX) of the hot flame. The ignition delay τ (deg.) was defined as the interval from the time a cool flame occurred θ_{LTR} (deg.) to the ignition timing. These definitions are diagrammed in Fig. 3

DEFINITION OF EGR RATE

A 2-stroke engine has more residual burnt gases that cannot be removed from the cylinder than a 4-stroke engine. The EGR gas was defined as the burnt gases which could be intentionally varied by opening and closing the EGR control valve provided at the exhaust port exit. Scavenging efficiency η_s (–) was found by assuming a condition of perfect mixing scavenging for the gas exchange process in the cylinder[12]. Accordingly, the EGR rate γ_{EGR} (–) was defined as shown in Eq. (2) below where $\eta_s(EGR)$ and $\eta_s(Limit)$ respectively indicate the scavenging efficiency with and without EGR. The following details have been added to the Appendix. It

should be noted that the scavenging efficiency $\eta_s(Limit)$ of the test engine under the baseline condition was around 70%.

$$\gamma_{EGR} [\%] = (\eta_s(Limit) - \eta_s(EGR)) * 100 \quad (2)$$

CHEMICAL KINETIC SIMULATION METHOD

Chemical kinetic simulations were performed using CHEMKIN-Pro IC Engine software under conditions that simulated those used in the experiments. The detailed reaction mechanism used in the calculations was developed by the Combustion Chemistry Group at the Lawrence Livermore National Laboratory for primary reference fuels and included 1,024 chemical species and 4,236 elementary reactions[13,14,15].

RESULTS AND DISCUSSION

CHARACTERISTICS OF HCCI COMBUSTION USING A TWO-COMPONENT FUEL

The results of previous studies have confirmed that increasing the quantity of iso-octane injected while keeping the quantity of n-heptane constant tends to produce the same in-cylinder behavior in the interval to ignition even though a larger amount of fuel is supplied[10]. Figure 4 shows typical cylinder pressure and HRR waveforms

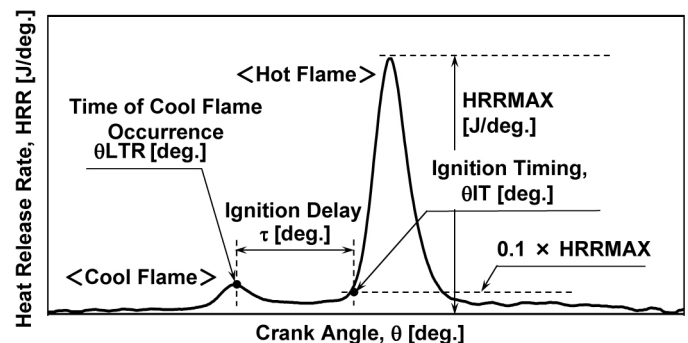


Fig. 3. Definition of ignition timing, time of cool flame occurrence and ignition delay

obtained in Cases 1-3 under a constant scavenging temperature, and Figure 5 shows the corresponding waveforms for Cases 4-6 under the same condition. Figure 6 shows the ignition timing as a function of the temperature at the onset of compression T_i (i.e., the temperature at the completion of the scavenging process) in Cases 1-3.