Homogeneous charge compression ignition (HCCI)

Subjects: Engineering, Mechanical
Contributor: Yew Heng Teoh

Homogeneous charge compression ignition (HCCI) is considered an advanced combustion method for internal combustion engines that offers simultaneous reductions in oxides of nitrogen (NOx) emissions and increased fuel efficiency. The present study examines the influence of intake air temperature (IAT) and premixed diesel fuel on fuel self-ignition characteristics in a light-duty compression ignition engine. Partial HCCI was achieved by port injection of the diesel fuel through air-assisted injection while sustaining direct diesel fuel injection into the cylinder for initiating combustion. The self-ignition of diesel fuel under such a set-up was studied with variations in premixed ratios (0–0.60) and inlet temperatures (40–100 °C) under a constant 1600 rpm engine speed with 20 Nm load.

Keywords: sustainable environment ; HCCI ; self-ignition ; renewable fuels

1. Introduction

The internal combustion engine was invented over a century ago as a replacement for the steam engine. Due to their superior weight to power ratio which grants them higher mobility, they have assumed the lead role in powering transportation. Put simply, two major types of internal combustion engines are spark ignition (SI) and compression ignition (CI) engines. SI engines generally have lower thermal efficiency than CI engines, which are more favourable in heavy duty uses. However, the former emit less nitrogen oxides (NO_x) and particulate matter (PM) into the environment. Over the past few years, global greenhouse gas emissions have continued to grow despite efforts to mitigate climate change. Besides this, among emissions that are commonly found in internal combustion engines are NO_x, carbon monoxide (CO), smoke and PM. Unburnt hydrocarbon (HC) also makes up part of internal combustion engine exhaust due to the use of carbon-rich fossil fuels. The release of those gases due to the combustion of fuel in engines has become a public concern since they threaten not only the environment but are also detrimental to human well-being.

Recently enforced emissions regulations that emphasize a reduction in greenhouse gas emission and improvement in fuel economy have pushed automakers to develop cleaner technologies to meet the stringent requirements. The automotive sector is poised to be a main source of emissions in the year 2030. The Kyoto protocol aims at a more sustainable imminent future by decreasing pollution secreting energy sources [1]. As a result, improvements in SI and CI engine combustion aspects such as optimised fuel injection (FI) timing, altered combustion chamber shape and FI with higher pressure have been introduced over the last decades to make internal combustion engines cleaner and more efficient. Nevertheless, these techniques were not able to substantially resolve SI and CI engines' emission problems. In search of better solutions, attempts to improve contemporary combustion strategies, including low-temperature combustion (LTC), that combine the benefits of both fewer emissions and greater thermal efficiency with a lower combustion temperature have been initiated [2]. Homogeneous charge compression ignition (HCCI) is one of the LTC strategies introduced by Onishi et al. [3] as an attempt to improve combustion stability in gasoline-fuelled engines. It utilises the auto-ignition of well-premixed fuel–air mixture channelled into engine cylinders by piston intake stroke to achieve combustion near the top dead centre (TDC) when the mixture is being compressed and detonated. Since then, many researchers had adapted HCCI combustion in engines operating with various fuels such as alcohols, diesel and biofuels and reported its potential to revolutionise the automobile sector [4][5][6][7].

Generally, HCCI combustion offers reductions in NO_x and smoke emissions, defying the well-known NO_x–smoke trade-off with superior fuel flexibility [8][9][10]. Due to fuel ignitions at multiple spots spontaneously in HCCI combustion, the formation of a localized high-temperature zone that favours thermal NO_x formation through the Zeldovich mechanism can be prevented [8]. Furthermore, the combustion of homogeneous premixed mixture gives rise to the absence of a fuel-rich zone to assist soot formation. Several researchers have also reported comparable or even higher efficiency with the use of HCCI combustion than conventional SI and CI modes [9][11][12]. With its merits, the HCCI strategy has caught the attention of researchers and manufacturers as a promising alternative especially to overcome high NO_x emissions from diesel engines. However, the use of HCCI combustion in engines is also associated with difficulties, particularly in its combustion phase control, cold start, limited operating range and premixed mixture preparation. Engine knocking at high load conditions and misfiring due to late auto-ignition when employing HCCI take a toll on engine performance and may contribute to engine wear and damage [13][14]. Unfortunately, higher emissions of unburnt HC and CO also were found with HCCI combustion [15][16][17]. To overcome the pitfalls of HCCI, combustion modes extended from HCCI such as premixed charge compression ignition (PCCI), homogeneous charge diesel combustion (HCDC) and stratified charge compression ignition (SCCI) were proposed. Besides this, control strategies and systems to achieve designed ignition timing and the start of combustion (SOC) of the fuel mixture have been studied by many researchers [18][19][20][21].

One of the extended combustion strategies is premixed/direct injection HCCI combustion or HCCI-DI that employs the preparation of a homogeneous fuel–air mixture upstream of the cylinder intake manifold and directly injected fuel to trigger combustion near TDC. HCCI-DI has been claimed to provide a wider operating range for the engine with greater thermal efficiency than engines running with pure HCCI combustion [22]. Furthermore, this combustion strategy also exhibits advantages over its predecessor with relatively lower HC and CO emissions [18][23]. The degree of homogeneity of the charge, which is affected by its preparation method, is the key in achieving fuel mixture auto-ignition for smooth applications of the HCCI concept in engines. In fact, fuel–wall interactions when using the internal preparation of the charge in the cylinders may give rise to wall wetting or impingement problems that are highly disadvantageous for HC emissions. Out of the many proposed methods, the most convenient and also effective approach is premixing the fuel and air by port fuel injection (PFI) as applied in a conventional SI engine [24]. This method offers sufficient time for the dispersion of fuel and fuel–air mixture formation in engine cylinders before ignition. According to Ganesh et al. [25], PFI is a strategy that offers economic benefits as it does not require engine modification in the first place, while producing notably few emissions under light load conditions.

2. Experimental Apparatus and Procedure

2.1. Engine

A single-cylinder, four-stroke direct-injection CI engine was employed to carry out tests in the present study. Slight modifications on the test engine were made to add an air-assisted PFI system to it. The specifications of the test engine are presented in **Table 1**. The set-up of the apparatus, including the test engine, hardware and software that were necessary to control engine parameters like IAT and auxiliary fuel injection (FI) rate, is as shown in **Figure 1**.

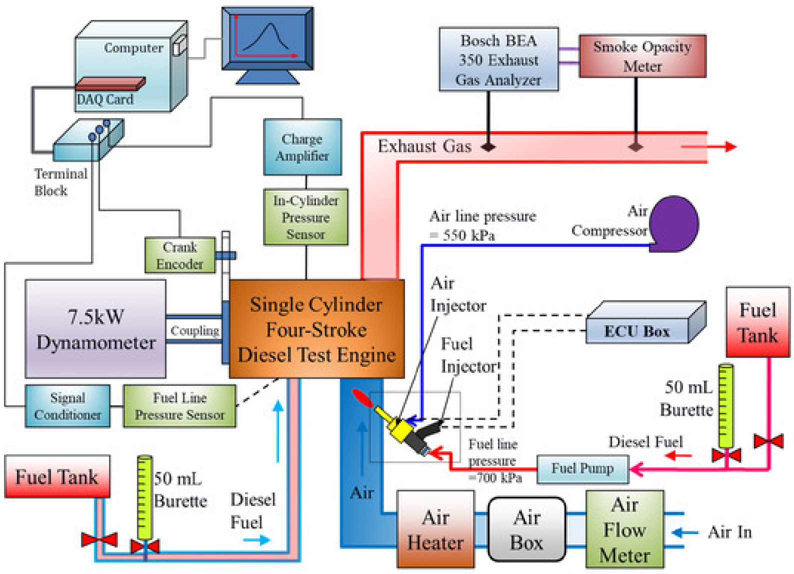

Figure 1. Schematic diagram of the experimental setup.

Table 1. Engine specifications used in the experiments.

Engine Model	Single-Cylinder Water-Cooled 4-Stroke DI Diesel
Bore (mm)	**92**
Stroke (mm)	**96**
Displacement (cm^3)	**638**
Compression ratio	**17.7:1**
Continuous rating output (rpm)	**7.8 kW @ 2400**

Engine Model	Single-Cylinder Water-Cooled 4-Stroke DI Diesel
One-hour rating output (rpm)	8.9 kW @ 2400
Injection timing (°BTDC)	17
Injection pressure (bar or kg/cm^2) Connecting rod length	196 or 200 149.5 mm
Connecting rod length (mm)	149.50

2.2. Fuel Injection System

In this work, an FI system was adapted to provide air-assisted atomisation of the liquid fuels. The air-assisted FI system comprised a 700 kPa fuel metering injector and a 550 kPa air injector. The FI was timed to coincide with the cylinder's intake stroke and was made into the intake port just upstream of the intake manifold. Two separate ECUs, each connected to a Hall Effect inductive proximity sensor, were used to provide precise controls on injection timing and the operating duration of the injectors. The fuel vaporiser plumes produced three series of injections into a 2 L flask under atmospheric conditions as shown in **Figure 2**, where each of the images captured demonstrates a subsequent 2 mL FI. As revealed in the images, this injection strategy produced a finely atomised fuel spray.

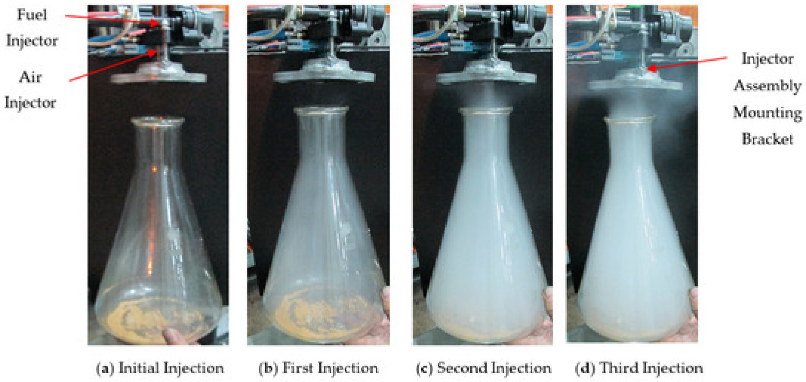

Figure 2. Fuel vaporiser plume in a 2 L flask at room temperature.

2.3. Fuel Consumption Measurement

In this study, fuel consumption for both the diesel direct injection and PFI systems were measured volumetrically. This method employed a glass burette of known volume with volume level markings. The time duration for the consumption of a certain volume of fuel was measured by a data logger system. The volumetric flow rate was calculated by a quotient of the volume measured by the time taken. In this experiment, two separate burettes, each of a 50 mL capacity, were used to measure the volumetric fuel consumption of both the diesel direct injection and PFI systems. To enhance measurement accuracy, a fully automated volumetric-type fuel-flow measuring system was used for each fuelling system. The system included a 50 mL capacity glass burette (A), which had multiple slotted interrupter photosensors (B1 to B5) spaced evenly along its length, each located at a known fuel level to minimize human error in measuring the fuel level against the marks on the burette. The photosensors were arranged perpendicularly to the burette to allow measurement by the infrared light beam transmitted parallel to the fuel-level marks on the burette. Besides this, to ease the fuel refilling process, a magnetic refilling valve (C) was integrated into this system.

It was assumed that the fuel level would initially be at the upper portion (B1 level) of the burette. By keeping the magnetic refilling valve closed, the flow from the fuel tank to the burette was stopped and the fuel level fell at a rate dependent only on the engine consumption. When the fuel level fell below each of the slotted interrupter photosensors, the infrared light beam from the emitter was transmitted to the opposite detector and converted into an electrical signal. These signals together with their time of occurrence were recorded by a data logger system. Thus, the fuel consumption in volumetric units could be obtained by dividing the relative volume (the pre-determined fuel volume between the successive sensors) by the time duration. When the fuel level reached the lower measuring level (B5), the magnetic refilling valve was triggered to open to allow fuel from the tank to flow into the burette. The refilling process was terminated again when the fuel level reached the upper portion (B1 level) of the burette. These automatic refilling processes were repeated throughout the experiment to ensure a continuous supply of fuel into the burette.

2.4. Auxiliary Fuel Injector Calibration Protocol

Prior to the installation of the FI system on the engine, both the fuel and air injectors underwent calibration processes. The bench test was carried out with an ECU with varying injection pulse width to determine the amount of fuel delivered. A commercially available Arduino UNO™ microcontroller board was used to interface with the ECUs by generating a consistent square wave signal that mimics the engine speed signal. In addition, the controller was also connected with the slotted interrupter photosensors and a magnetic refilling valve for the automatic fuel consumption measurement. The number of injection counts and the instances when the fuel level fell below each of the slotted interrupter photosensors were all recorded by using LabVIEW software. Thus, the average fuel volume per injection pulse for a specific injector pulse width was obtained by dividing the relative volume (the pre-determined fuel volume between the successive sensors) by the injection count. Alternatively, the fuel flow rate could be given in terms of fuel mass per injection pulse by multiplying the fuel volume by the density of the fuel. The experimental results revealed that the fuel mass/injection pulse varied proportionally to the injector pulse width.

2.5. Intake Air Systems and Exhaust Emission Measurements

To minimise pressure pulsations of the intake gases and provide higher engine operational stability, a surge tank was installed to enable airflow measurement. The surge tank was installed upstream of a 5.5 kW heater in the engine intake system. The power supplied to the heater was precisely controlled to maintain the intake air at a particular temperature. The flow rate of the intake air to the engine was determined using a hot-film-type air-mass flow meter. Pressurised clean air was supplied to the air injector through an air compressor to enable better fuel atomisation before it entered the intake manifold. On the other hand, exhaust emission measurements were conducted by a Bosch BEA 350 exhaust gas analyser. The concentration of carbon monoxide (CO), carbon dioxide (CO_2), unburned hydrocarbon (HC), oxygen (O_2) and nitric oxide (NO) were measured and analysed. At the same time, an integrated smoke opacity meter was also employed to determine the smoke opacity of the engine exhaust.

2.6. Data Acquisition

A moderate-speed data logging system was established by the implementation of an Advantech USB multifunction module. The LabVIEW program was employed to provide the means of controlling the hardware in this experiment, including the real-time monitoring of the intake air flow rate, fuel flow rate, engine speed and load cell signals. Besides this, the changes in engine coolant and IAT prior to heating along with ambient temperature were monitored using an Advantech USB 8-ch thermocouple. In this experiment, the fuel combustion analysis was carried out by measuring the in-cylinder pressure with a Kistler 6125B-type pressure sensor. The charge signal output from the pressure sensor was processed and conditioned by a PCB model charge-to-voltage converter. The data registered was further processed to determine the heat release rate during the combustion. A high-precision Leine and Linde incremental encoder (model: 632-00685-1) with 720 pulses per revolution was employed to monitor the shaft rotation. Besides this, a computer was used as the data acquisition unit, which was equipped with 14-bit resolution, a 2 MS/s sampling rate and 4 analogue input channels to simultaneously sample and synchronize the in-cylinder pressure and encoder signals. Further processing of the recorded data was then carried out by MATLAB software.

2.7. Dynamometer

A single-phase AC synchronous electrical generator was employed to absorb the engine load from the test engine. The drive end of the dynamometer shaft carried a 30-toothed wheel that was used with a magnetic pick-up to provide the speed measurement. A torque arm was attached to the generator housing on a horizontal centreline and the force exerted by the housing attempting to rotate the torque arm was measured with a strain gauge load cell at a radius of 245 mm. The torque was calculated as the product of the force measured by the load cell and the length of the torque arm. The power output from the generator was dissipated to a load bank consisting of three units of 3 kW screw-in water immersion heaters. A dynamometer controller featuring a digital PID (Proportional Integral Derivative) closed-loop control method was employed to keep the engine speed constant throughout the experiment. The controller pulse width modulated the power transferred to the coils via a field-effect transistor.

2.8. Experimental Procedure

During the start of the experiment, the test engine was initiated in direct injection mode with no load exerted to allow the engine to warm up. When the operating temperature achieved the optimum value of 80 °C, tests involving variations in the premixed ratio, with r_p up to 0.6 and IAT up to 100 °C, were started. The premixed ratio was controlled by adjusting the pulse injection widths of the air-assisted PFI system and also the engine direct injector. The actual premixed ratio was then determined by fuel consumption results. The term r_p represents the ratio of the energy of the premixed fuel Q_p to the total energy Q_t; the value of r_p was calculated by Equation (1).

$$r_p = \frac{Q_p}{Q_t} = \frac{m_p h_p}{m_p h_p + m_d h_d} \quad (1)$$

where m_p is the mass of the premixed fuel, m_d is the mass of the directly injected fuel and h is the lower heating value. The subscripts p and d denote premixed and directly injected fuel respectively. Several physiochemical properties of the diesel fuel employed are listed in **Table 2**. Since the direct-injected fuel and premixed fuel chosen in this study was diesel, the heating values for both fuels were the same as shown here—hence $h_p = h_d$.

Table 2. Fuel properties of diesel fuel.

Properties	Test Method	Specification
Cetane index	ASTM D976	>52
Flash point	ASTM D93	71.5 °C
Density @ 40 °C	ASTM D7042	838.4 kg/m^3
Kinematic viscosity @ 40 °C	ASTM D7042	3.815 mm^2/s
Heating value	ASTM D4809	45.31 MJ/kg

The engine and test conditions used in this experiment are summarised in **Table 3**. The tests conducted can be divided into two main parts. Firstly, the value of r_p was varied from 0 up to 0.6 and the subsequent effects of r_p on HCCI combustion, engine performance and emissions were studied. The engine was run at a constant 1600 rpm speed and load of 20 Nm at all operating points. For each of the operating conditions, the end of injection (EOI) timing for the premixed fuel was held constant, whereas the start of injection (SOI) timing was altered according to the injection pulse width setting. In the second part of this study, the effects of intake temperature were investigated. The premixed fuel ratio was kept around a value of r_p of 0.5 while the IAT was manipulated, ranging from 40 to 100 °C.

Table 3. Experimental test conditions.

Engine Speed		1600 rpm
Engine load		20 Nm
Intake air temperature		40–100 °C
Fuel	Premixed	Diesel
	Directly injected	Diesel
Injection pressure	Premixed	700 kPa
	Directly injected	19,600 kPa
Injection timing	Premixed	47 °ABDC
	Directly injected	17 °BTDC
Premixed ratio, r_p		0–0.6

References

1. Yasin, M.H.; Mamat, R.; Najafi, G.; Ali, O.M.; Yusop, A.F.; Ali, M.H. Potentials of palm oil as new feedstock oil for a global alternative fuel: A review. Renew. Sustain. Energy Rev. 2017, 79, 1034–1049.
2. Agarwal, A.K.; Singh, A.P.; Maurya, R.K. Evolution, challenges and path forward for low temperature combustion engines. Prog. Energy Combust. Sci. 2017, 61, 1–56.
3. Onishi, S.; Jo, S.H.; Shoda, K.; Jo, P.D.; Kato, S. Active Thermo-Atmosphere Combustion (ATAC)—A New Combustion Process for Internal Combustion Engines; SAE International: Warrendale, PA, USA, 1979.

4. Urushihara, T.; Hiraya, K.; Kakuhou, A.; Itoh, T. Expansion of HCCI Operating Region by the Combination of Direct Fuel Injection, Negative Valve Overlap and Internal Fuel Reformation; SAE International: Warrendale, PA, USA, 2003.
5. Gomes Antunes, J.M.; Mikalsen, R.; Roskilly, A.P. An Investigation of Hydrogen-Fuelled HCCI Engine Performance and Operation. Int. J. Hydrogen Energy 2008, 33, 5823–5828.
6. Azad, A.K.; Rasul, M.G.; Khan, M.M.; Sharma, S.C.; Bhuiya, M.M. Recent development of biodiesel combustion strateg ies and modelling for compression ignition engines. Renew. Sustain. Energy Rev. 2016, 56, 1068–1086.
7. Hasegawa, R.; Yanagihara, H. HCCI Combustion in DI Diesel Engine; SAE International: Warrendale, PA, USA, 2003.
8. Juttu, S.; Thipse, S.S.; Marathe, N.V.; Babu, M.G. Homogeneous Charge Compression Ignition (HCCI): A New Concept for Near Zero NOx and Particulate Matter (PM) from Diesel Engine Combustion; The Automotive Research Association of India: Maharashtra, India, 2007.
9. Hasan, M.M.; Rahman, M.M. Homogeneous charge compression ignition combustion: Advantages over compression ig nition combustion, challenges and solutions. Renew. Sustain. Energy Rev. 2016, 57, 282–291.
10. Gowthaman, S.; Sathiyagnanam, A.P. Effects of charge temperature and fuel injection pressure on HCCI engine. Alex. Eng. J. 2016, 55, 119–125.
11. Zhao, F.; Asmus, T.N.; Assanis, D.N.; Dec, J.E.; Eng, J.A.; Najt, P.M. Homogeneous Charge Compression Ignition (HC CI) Engines—Key Research and Development Issues; SAE International: Warrendale, PA, USA, 2003.
12. Szybist, J.P.; Bunting, B.G. Cetane Number and Engine Speed Effects on Diesel HCCI Performance and Emissions; S AE International: Warrendale, PA, USA, 2005.
13. Singh, A.P.; Agarwal, A.K. Combustion characteristics of diesel HCCI engine: An experimental investigation using exter nal mixture formation technique. Appl. Energy 2012, 99, 116–125.
14. Hairuddin, A.A.; Yusaf, T.; Wandel, A.P. A review of hydrogen and natural gas addition in diesel HCCI engines. Renew. Sustain. Energy Rev. 2014, 32, 739–761.
15. Bendu, H.; Murugan, S. Homogeneous charge compression ignition (HCCI) combustion: Mixture preparation and contr ol strategies in diesel engines. Renew. Sustain. Energy Rev. 2014, 38, 732–746.
16. Gray, A.W.; Ryan, T.W. Homogeneous Charge Compression Ignition (HCCI) of Diesel Fuel; SAE International: Warrend ale, PA, USA, 1997.
17. Kim, D.S.; Lee, C.S. Improved emission characteristics of HCCI engine by various premixed fuels and cooled EGR. Fu el 2006, 85, 695–704.
18. Kumar, P.; Rehman, A. Bio-diesel in homogeneous charge compression ignition (HCCI) combustion. Renewable and S ustain. Energy Rev. 2016, 56, 536–550.
19. Pandey, S.; Diwan, P.; Sahoo, P.K.; Thipse, S.S. A review of combustion control strategies in diesel HCCI engines. Biof uels 2018, 9, 61–74.
20. Saravanan, S.; Pitchandi, K.; Suresh, G. An experimental study on premixed charge compression ignition-direct ignition engine fueled with ethanol and gasohol. Alex. Eng. J. 2015, 54, 897–904.
21. Suzuki, H.; Koike, N.; Odaka, M. Combustion Control Method of Homogeneous Charge Diesel Engines; SAE Internatio nal: Warrendale, PA, USA, 1998.
22. Ying, W.; Li, H.; Jie, Z.; Longbao, Z. Study of HCCI-DI combustion and emissions in a DME engine. Fuel 2009, 88, 225 5–2261.
23. Ma, J.; Lü, X.; Ji, L.; Huang, Z. An experimental study of HCCI-DI combustion and emissions in a diesel engine with du al fuel. Int. J. Therm. Sci. 2008, 47, 1235–1242.
24. Yao, M.; Zheng, Z.; Liu, H. Progress and recent trends in homogeneous charge compression ignition (HCCI) engines. Prog. Energy Combust. Sci. 2009, 35, 398–437.
25. Ganesh, D.; Nagarajan, G. Homogeneous charge compression ignition (HCCI) combustion of diesel fuel with external mixture formation. Energy 2010, 35, 148–157.