

25cc HCCI Engine Fuelled with DEE

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ABSTRACT

This paper describes the set-up and testing of a single cylinder 25cc, air cooled, 4-stroke Spark Ignition (SI) engine converted to run in Homogeneous Charge Compression Ignition (HCCI) mode with the aid of various combustion control systems. The combustion control systems were investigated regarding their effects on combustion stability and heat release phasing. Engine operation was compared with unique findings from previous work done on a very small 2-stroke HCCI engine.

HCCI engine operation was possible between 1000 - 4000 rpm when using Diethyl Ether (DEE) as the test fuel. Maximum operational fuel-air equivalence ratio (Φ) was 0.75 when operating without Exhaust Gas Recirculation (EGR). This relatively high equivalence ratio was attainable due to thermal gradients induced by the high surface area to volume ratio of the small engine combustion chamber, resulting in high chamber heat transfer. Combustion phasing peculiarities, due to fuel Negative Temperature Coefficient (NTC) auto-ignition behaviour, engine heat transfer and breathing characteristics, were investigated.

Additionally, this paper provides a useful resource for building a compact, inexpensive HCCI engine test bed and matching HCCI engine model.

INTRODUCTION

Fuel efficiency requirements and ever more stringent emissions legislation require internal combustion engines to improve fuel economy and concurrently reduce exhaust emissions. One possible solution is advancing combustion technology in the form of HCCI engines which could operate either as single or dual combustion mode engines [1, 2].

HCCI combustion is the process in which a homogeneous air/fuel mixture is spontaneously auto-ignited through compression of the piston. Combustion characteristics allow HCCI to combine the best features of SI and Compression Ignition (CI) engines. HCCI engines offer significantly reduced engine out NO_x emissions and fuel economy improvements up to 20% [2] as compared to SI engines, while reducing particulate emissions when compared to CI Engines. However, due to ignition being via the compression of the piston, combustion stability and combustion timing (phasing) over a wide engine load range is a major technical challenge in the development of HCCI combustion. Various combustion control methods are currently being researched and implemented in order to produce stable and reliable combustion while also expanding the HCCI load range. HCCI Electronic Control Unit (ECU) research is ongoing with respect to closed-loop control of HCCI engine dynamics [3].

HCCI has been demonstrated to work under laboratory conditions achieving low emissions and good efficiency under steady state operation, however technical challenges arise when attempting to extend this performance to transient operating conditions.

Primary technical challenges with HCCI development are: [2]

- Combustion phasing control
- Rapid heat release rates and noise level with possible engine damage especially at higher loads
- Transient operation difficulties
- Excess CO and HC emissions, particularly at light loads
- Increased NOx at high loads
- Cold Start difficulty
- A narrow operation range (as result of the above)

This research forms part of an ongoing HCCI study at the Sasol Advanced Fuels Laboratory to investigate and understand engine configuration and fuel chemistry effects on combustion in HCCI engines. This project continues from a previous project [4] where a small high speed model “diesel” aeroplane engine was found to operate in HCCI mode with surprising ease and flexibility. The engine was shown to run from idle through a fixed speed-load curve (propeller) to the maximum speed and load point of the engine at 11000rpm and stoichiometric air-fuel ratio, while maintaining nominally constant combustion phasing without any form of combustion control system. This study continued the investigation by removing lubricant from the fuel and switching to 4-stroke operation while retaining a very small engine size. A 25cc four-stroke, single cylinder engine possessing 2-valves with an overhead cam and sump lubrication system was used in the study.

Insights were gained regarding which engine characteristics play the primary role in the remarkable operational flexibility of the engine used in [4]. A number of additional features of small engine HCCI are also shown.

Additionally this project aimed to produce a low cost flexible HCCI engine/dynamometer rig for research purposes.

PREVIOUS SMALL HCCI ENGINE RESEARCH

At the Sasol Advanced Fuels Laboratory, a standard issue two-stroke model-aero “diesel” engine (Figure 1) was found to operate using HCCI combustion [4]. This type of engine is of particular interest largely because it operated as a self-regulating HCCI engine over a wide range without any undesirable effects such as engine knock or misfire. This engine, unlike the more common glow-plug versions, operates without any form of combustion initiator. Fuel and air are premixed using a simple carburettor and ignited by piston compression only, thus the engine operates in HCCI mode even though it is referred to as a “model diesel” engine.

Engine starting is possible from cold, warm and hot conditions with ability to operate from idle to over 11000rpm. Operation is possible at high load points across the speed range with very conservative pressure rise rates. Furthermore, engine knock (high pressure oscillations) is not experienced during normal engine operation. This posed questions with respect to the possibility of similar full scale HCCI engine operation and the role of engine size and fuel requirements in obtaining HCCI operation over a wide load range.



Figure 1: P.A.W Model Aero-”Diesel” Engine

TEST ENGINE

A 25cc, single cylinder 4-stroke engine was chosen due to it being the smallest commercially available 4-stroke engine operating with no fuel/lubrication mixing. Manufacturer’s engine specifications are shown in Table 1 below.

Table 1: Manufacturer’s Engine Specifications

Engine Designation	GX25
Engine Type	4-stroke single cylinder
Bore X Stroke	35 x 26 mm
Displacement	25cc
Compression Ratio	8:1
Fuel preparation system	Diaphragm carburettor
Net Power Output	0.72 kW at 7000 rpm
Net Torque	1.0 Nm at 5000 rpm
Oil Capacity	80cc
Maximum No-load Speed	11000 rpm
Operational Speed	4000 - 9000 rpm
Inlet Valve Diameter	12.5 mm
Exhaust Valve Diameter	12.0 mm
Valve Lift	2.5 mm
CAD/Tooth on Timing Belt	24 degrees

The GX25 possesses two valves, inlet and exhaust with a single overhead cam and timing belt. The single overhead cam results in fixed valve lift, fixed relative valve timings - durations between inlet and exhaust valves. However the camshaft timing belt can be offset on the cam by shifting the belt a desired number of teeth, allowing for flexibility in altering of the relative valve timing. The standard valve lift profiles are given in Figure 2 below.

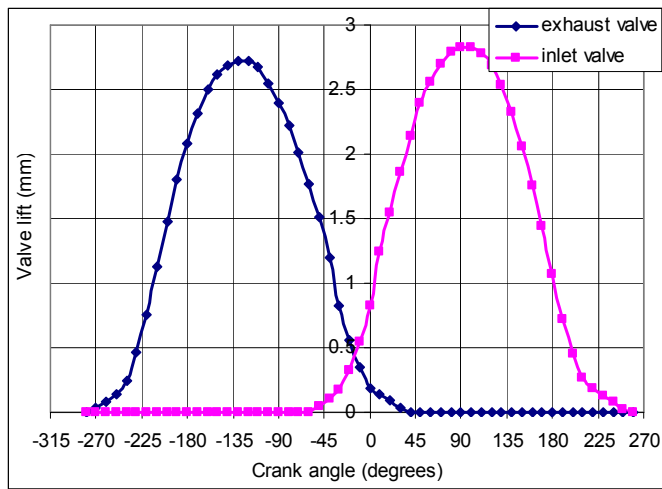


Figure 2: Standard exhaust and inlet valve lift profiles

An oil sump separates the lubricant from the air/fuel mixture (see Figure 3 below), thus allowing for comparison with the model “diesel” engine, which utilised castor oil as a lubricant in its fuel.

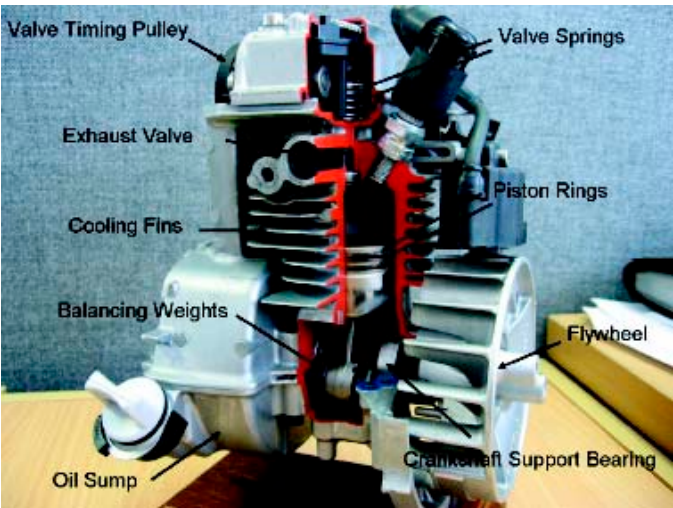


Figure 3: Cut-away Model of the GX25 Engine

Setup for the study therefore involved conversion of the engine to operate in HCCI mode. This required the engine to be operated on a test bench with a reduction drive, regenerative DC dynamometer and electronic control system. Various combustion control systems were added to the engine to obtain HCCI combustion

and investigate combustion control mechanism effects on small engine HCCI combustion stability, heat release phasing and exhaust emissions. These controls included, inlet mixture heating, internal and cooled external EGR, inlet pressure boosting, valve timing and external cylinder cooling.

TEST FUELS

The engine’s low compression ratio of 8:1, small displacement and associated high combustion chamber heat loss, necessitated the need for a volatile, high ignition quality fuel. Primary testing was conducted using Diethyl Ether as it’s high ignition quality and high cetane number (CN > 125 [5]) allows for operation without excessive inlet heating requirements. Properties of this test fuel are given in Table 2.

Table 2: Diethyl Ether Properties ([5], [6])

Property	Value
Molar Mass	74.12 g/mol
Density	0.7134 g/cm ³
Boiling Point	308 Kelvin
Autoignition Temperature	433 Kelvin
Cetane Number	> 125
Stoichiometric Air/Fuel Ratio	11.1 (mass basis)
Lower Heating Value	33892 kJ/kg

DEE is a two-stage fuel [7, 8], exhibiting cool-flame and peroxide autoignition reactions with a corresponding Negative Temperature Coefficient (NTC) region. NTC auto-ignition behaviour is shown by Figure 4 below. The expected trends are such that as the initial temperature increases, the pre-coolflame (period “a” in Figure 5) delay reduces and the associated cool-flame temperature rise also becomes smaller. The NTC region is a region of the ignition delay curve where ignition delay increase as temperature increases. This results from the diminishing magnitude of the cool flame temperature rise as the initial temperature increases [9].

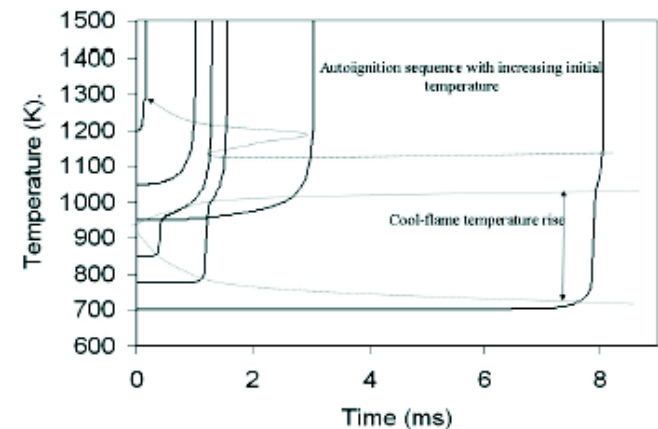


Figure 4: Profiles Showing the Effect of Initial Temperature on Ignition Delay of a 2-Stage Ignition Fuel in a Constant Volume Combustion Chamber [9]

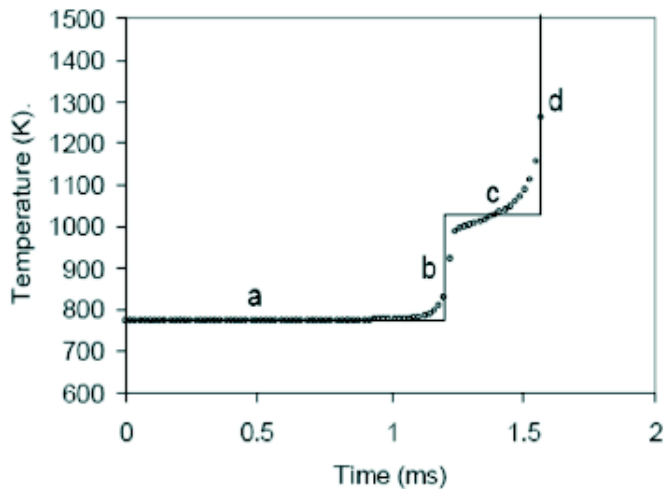


Figure 5: Ignition Delay Curve with Cool Flame Ignition Delay [9]

Engine testing using n-Heptane was completed in order to model observed engine operational characteristics, such as combustion phasing, heat release rates and combustion chamber heat transfer characteristics. n-Heptane was used due to the availability of thermodynamic and auto-ignition data, which was not available for DEE.

Table 3: n-Heptane Properties Table

Properties	Value
Mole Weight	100.20 g/mol
Density	0.682 g/cm ³
Boiling Point	371.3 Kelvin
Lower Heating Value	44566 kJ/kg

EXPERIMENTAL SETUP

A secondary goal for this research was to design an inexpensive and compact HCCI rig. Specifications of the equipment used and a table of equipment costs can be found in Appendix C.

Figure 6 illustrates the engine mounting, which consists of a stand with adjustable feet and slotted plates allowing for component mounting flexibility and shaft alignment.

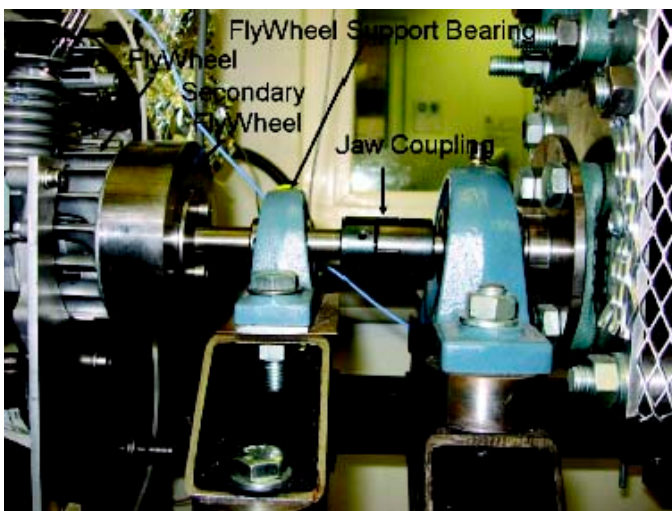


Figure 6: Power Transmission Mechanism

The crankshaft assembly consists of two separate shafts that are press-fitted with counterweights on either side and supported by bearings, creating an inherent misalignment along the crankshaft. A secondary flywheel, supported by a self aligning bearing, was necessary due to the low rotational inertia of the original flywheel. The additional flywheel support minimises the effects of crankshaft misalignment as well as reducing static loading on the crankshaft.

Power from the engine is transmitted to a regenerative DC motor, able to absorb 1.1 kW at a speed of 3000rpm. Operational speed differences between the engine and dynamometer resulted in a speed reduction requirement. A toothed belt reduction drive, see Figure 7, with an overall reduction ratio of 3:1 was installed.

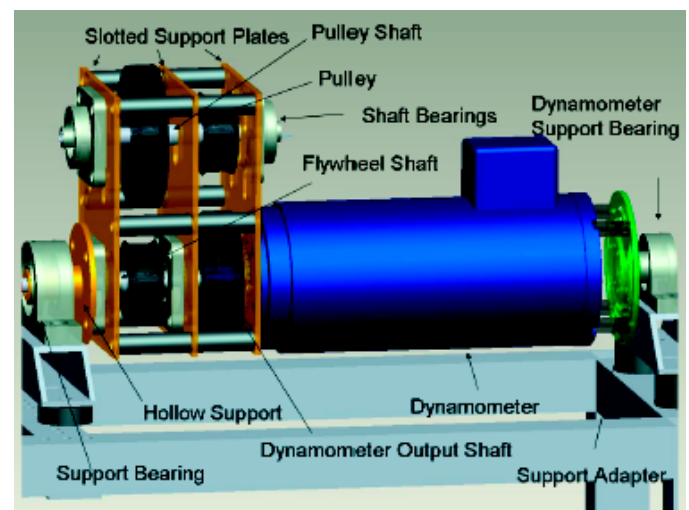


Figure 7: Reduction Drive and Dynamometer Assembly

The reduction drive and dynamometer are assembled as a 'freely' rotational system, supported by two non-rotating trunion bearings, to facilitate the installation of a simple strain gauge to monitor engine load.

MEASUREMENT EQUIPMENT

Various measuring devices were installed for data capturing and engine monitoring during operation. Below is a list of all installed instrumentation and their location:

Table 4: Location of Measurement Equipment

Signal	Description
P_1	Cylinder Pressure
T_{AR}	Air Reservoir Temperature
$T_{mixture}$	Post Injector Mixture Temperature
T_{exh}	Exhaust Gas Temperature
T_{EGR}	EGR Line Pre-injector Temperature
T_{HE}	EGR Heat Exchanger Temperature
T_{cyl1}	Cylinder Wall Temperature 1
T_{cyl2}	Cylinder Wall Temp 2
λ_1	Exhaust lambda value
λ_2	Inlet EGR lambda value
P_2	Air Reservoir Pressure
P_3	Exhaust Back-Pressure
Torque	Reduction/Dyno Assembly
Crank Angle	Engine Rear - Speed
FTIR Analyzer	Exhaust

Cylinder pressure was measured with a water-cooled piezo-electric pressure transducer mounted into the existing spark-plug hole, without the need for any modification. The pressure transducer used was calibrated for a pressure range up to 250 bar.

T_{cyl1} and T_{cyl2} measure temperature in the cylinder wall. These temperature measurements are used to create a linear temperature extrapolation through the cylinder wall, based on their relative placements in the cylinder wall. The extrapolated temperature measurement is then utilised in heat loss modelling of the engine. (See Appendix A for positioning of the thermocouples)

NO_x, CO₂ and CO exhaust emissions sampling measurements were measured using a Fourier Transform Infra- Red (FTIR) analyzer connected to the exhaust manifold.

DATA ACQUISITION

Engine control is achieved via an in-house developed Electronic Control System (ECS) using National Instruments LabView_{TM} and a National Instruments FPGA DAQ card, allowing for constant monitoring of engine operation as well as basic control.

The ECS is able to directly control the following operating parameters:

- Engine Speed
- Injection Timing
- Injection Duration

A realtime data acquisition system is used to monitor cylinder pressure, manifold pressures, fuel-air equivalence ratio, temperatures, engine torque, injection control and engine speed. The ECS calculates 50% heat release position as well graphing relative cylinder pressure, heat release rates and cumulative heat release rates. Analysis of the pressure trace is used to obtain pressure rise rates, peak pressures and locations of peak pressure. (See Appendix B for schematic layout of the engine and data capturing setup)

COMBUSTION CONTROL METHODS UTILISED

FUEL INJECTION

Obtaining a homogeneous mixture requires extended mixing time between the fresh charge and fuel. Use of injection timing strategies have shown that when Start of Injection (SOI) is retarded, the combustion timing is also retarded, whereas advancing fuel injection timing would lead to an increase in NO_x emissions [10].

A port fuel injection (PFI) system was installed by utilising a fuel injector (Figure 8) from a commercially available 50cc scooter. The injector is housed downstream of a throttle approximately 200mm from the inlet valve, allowing for evaporation and mixing time of the air/fuel mixture. Injection duration is set to a desired value on the ECS and an injector calibration graph is used to obtain mass flow rates at the set injection pressure of 5 bar. Injection timing is determined using a crank angle encoder on the engine, with injection timing being set at a desired crank angle BIVO (Before Inlet Valve Opening).

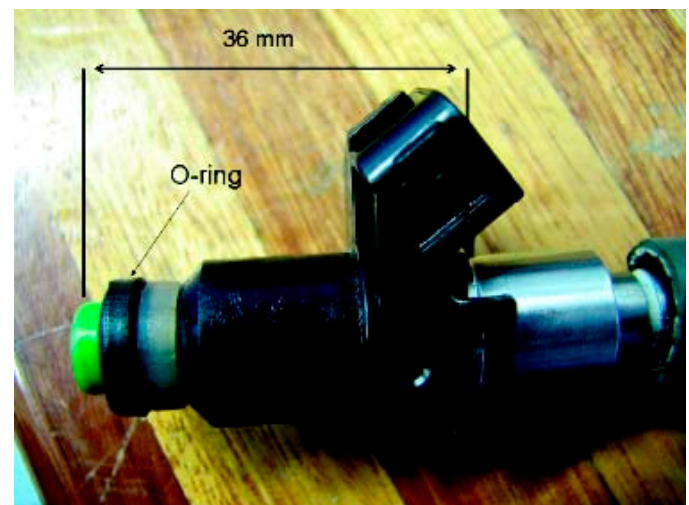


Figure 8: Fuel Injector

INLET MANIFOLD TEMPERATURE CONTROL

Inlet manifold temperature control allows for control of mixture temperatures pre-compression and is 16.5 °C