

Gasoline Fuelled Partially Premixed Compression Ignition in a Light Duty Multi Cylinder Engine: A Study of Low Load and Low Speed Operation

Adam Weall and Nick Collings
University of Cambridge

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ABSTRACT

The objective of this study was to examine the operating characteristics of a light duty multi cylinder compression ignition engine with regular gasoline fuel at low engine speed and load. The effects of fuel stratification by means of multiple injections as well as the sensitivity of auto-ignition and burn rate to intake pressure and temperature are presented. The measurements used in this study included gaseous emissions, filter smoke opacity and in-cylinder indicated information.

It was found that stable, low emission operation was possible with raised intake manifold pressure and temperature, and that fuel stratification can lead to an increase in stability and a reduced reliance on increased temperature and pressure. It was also found that the auto-ignition delay sensitivity of gasoline to intake temperature and pressure was low within the operating window considered in this study.

Nevertheless, the requirement for an increase of pressure, temperature and stratification in order to achieve auto-ignition time scales small enough for combustion in the engine was clear, using pump gasoline.

INTRODUCTION

At the present time a number of advanced combustion concepts for diesel engines are being researched with the aim of reducing engine out emissions while attempting to maintain high levels of fuel conversion efficiency.

The methods predominantly concentrate around achieving HCCI-like premixing using direct fuel injection in the compression stroke of a diesel engine. Approaches include using very high injection pressures with late injection timing, [1], [2], early injection timing [3], and very low temperature combustion [4,5]. In reality a combination of these methods is often used depending on the operating region of the engine. More recent studies into these advanced diesel combustion approaches include [6, 7, 8].

However, advanced diesel combustion approaches tend to differ from studies into gasoline HCCI. The former operates using a low temperature partially premixed charge while the latter in general operates using a homogeneous fully premixed charge.

Diesel fuelled partially premixed compression ignition (PPCI) has by and large suffered from a) difficulty in achieving fuel spray vaporization at low temperature and pressure b) limited premixing time resulting from the short ignition delay. This is directly attributed to the high boiling point and high Cetane number characteristics of diesel fuel. Although fuels with lower boiling points do allow a greater level of premixing to be achieved in a given time period, the effect of fuel ignition delay tends to dominate in this regime.

Diesel partially premixed compression ignition possesses both advantages and disadvantages when compared to gasoline HCCI; NO_x and soot emissions of the former tend to be higher than the latter although are still considered low when compared with conventional diesel combustion. One of the most important advantages of diesel PPCI is the ability to control the combustion phasing using fuel stratification achieved

during direct injection late in the compression stroke. The transition from diesel PPCI to conventional diesel combustion is also more straightforward than that from gasoline HCCI to gasoline SI mode.

Recently, the suitability of diesel as a fuel for future advanced diesel combustion engines has been discussed, for example [9,10]. One of the points of discussion is that the auto-ignition properties of diesel fuel are not suited to the increased ignition delay that is so crucial for achieving adequate levels of fuel-air premixing with relatively late direct injection fuelling events.

In order to overcome this inherent short ignition delay of diesel fuel and achieve enough premixing in a short duration, high levels of technological investment are required, such as very high pressure fuel injection (currently exceeding 1800bar common rail pressure), as well as complex fuel injector geometries.

The extent to which fuels with characteristics that differ from diesel fuel will benefit these advanced combustion methods is now being researched extensively and the range of candidate fuels is growing with time. The majority of the candidate fuels possess auto-ignition characteristics that are extended in duration compared to diesel. They include kerosene, [11], blends of diesel and gasoline during advanced diesel combustion [11,12] (as well as blends of diesel and gasoline in the case of fully premixed HCCI, [13,14]). Investigations using regular gasoline during advanced diesel combustion have also been carried out [10,15].

The findings of these studies show that benefits are apparent when using fuels with extended auto-ignition delays, firstly by reducing the requirement for advanced engine technology, and secondly by an increase in the operating range of low emission combustion into higher loads and engine speeds. These alternative fuels also possess several limiting factors such as cold start problems with very low Cetane number fuels, flashpoint issues with some fuel blends and ultimately the fuel distribution and engine compatibility issues that arise with a new type of fuel.

The use of regular gasoline fuel in a partially premixed operating regime in high load operation has been demonstrated, [10], and showed that the operating envelope far exceeds the range possible using diesel fuel. It was also shown that gasoline PPCI was possible using relatively low intake temperatures with a 14:1 compression ratio by using a combination of fuel stratification and higher intake boost pressure. This was achieved in a single cylinder engine using externally controlled intake air conditioning.

Successful operation of a gasoline PPCI engine would need to be extended to include low speed and low load regions as well as multi-cylinder engine operation. The study in this paper is concerned with the limits of

operation of regular gasoline in a multi-cylinder diesel engine in PPCI mode.

EXPERIMENTAL SETUP

The test engine was a Ford 2.0L, 4 cylinder Duratorq diesel engine with variable geometry turbocharger, external EGR cooling and common rail direct injection system. The valve timing was not modified and remained as production configuration.

The engine was connected to a motored dynamometer and operated with all four cylinders firing. The engine was started in a warm condition in all tests performed in this study.

In-cylinder pressure was measured with Kistler pressure transducers mounted in glowplug adaptors and recorded using an AVL Indimaster with 0.2degCA resolution. Combustion analysis data was calculated separately for each cylinder in each engine cycle and then all results were averaged. Cylinder to cylinder variation is also shown when appropriate. The coefficient of variance of IMEP was calculated using the relation shown in equation 1.

$$COV_{IMEP} = 100 \cdot \frac{\sigma_{IMEP}}{IMEP}$$

Equ 1 : Definition of COV [16]

Smoke measurements were taken using an AVL 415s heated sample smoke meter which uses a filter paper sampling method. The exhaust gas measurements were taken using a Horiba EXSA-1500 gas analyzer which included CO, CO₂, NO_x and THC measurement. Exhaust lambda values were measured using a Horiba MEXA-700 lambda sensor. Fuel consumption was measured using an AVL 733s gravimetric fuel meter. A diagram of the setup is shown in fig 1.

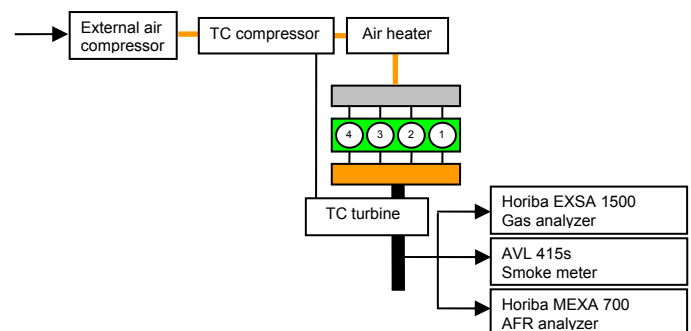


Fig 1 Engine test layout

The general engine specifications can be seen in table 1.

Engine type	4 stroke, 4 cylinder in-line, 16 valve, turbo intercooled, direct injection diesel
Displacement	1.998 liters
Bore x Stroke	86mm x 86mm
Nominal Comp. Ratio	14.4:1
Fuel injection equipment	Common rail solenoid (max. 1600bar)
Injection nozzle	6 hole VCO, $\phi=154\mu\text{m}$, 154° cone angle
Nozzle flowrate	380cc/30 sec@100bar

Table 1 Test engine specifications

TEST FUEL

Regular gasoline was used in this study. In order to be able to use gasoline fuel in a standard common rail injection system a lubricity additive was used to avoid damage to the fuel injection system components. The test fuel boiling point range can be seen in fig 2.

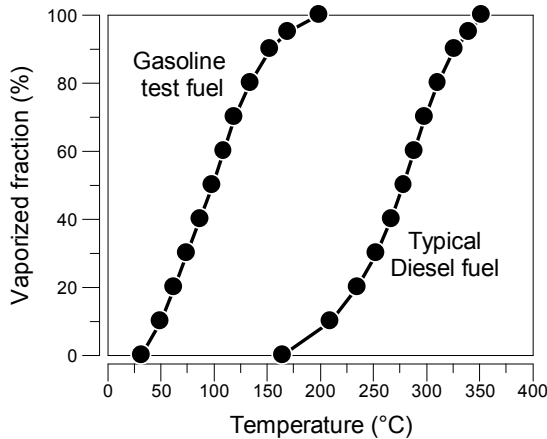


Fig 2 Boiling point characteristics of test fuel

The fuel specifications can be seen in table 2.

Test Fuel	European Gasoline
RON	95.2
MON	85.2
Density@15°C/(kg/m ³)	745.9
H/C ratio/-	1.788
Paraffins/%v	11.70
Isoparaffins/%v	34.78
Olefins/%v	13.27
Dienes/%v	0.14
Naphthenes/%v	4.07
Aromatics/%v	35.28
Oxygenates/%v	0.00
AFR stoich/-	14.45
Enthalpy of Combustion/(MJ/kg)	-43.3
Cetane nr/-	~15*

Table 2 Fuel specifications (*=estimated)

COMPRESSION RATIO

The test engine used in this study used a 14.4:1 compression ratio (CR) calculated with the inclusion of a nominal piston-firing deck clearance and valve recess volumes. This reduction of compression ratio from the respective production value of 18.4:1 was carried out in order to study low temperature PPCI using diesel-like fuels, [17,12].

A suitable CR for gasoline PPCI remains an open question due to the limited knowledge base in this area. This is discussed further in combination with the experimental results. In any case, the CR used in this study was approximately midway between those found in typical gasoline and diesel engines.

INTAKE CHARGE HEATING AND BOOST

The use of a high charge temperature to enable gasoline to auto-ignite in HCCI engines is well known, either using intake air heating or residual gas through negative valve overlap (NVO). It was also shown in [10] that gasoline would auto-ignite with a lower charge temperature when fuel stratification was increased and intake charge pressure was increased.

In the study of [10], a 14:1 compression ratio engine was operated using a 40°C intake air temperature with 1 bar gauge air pressure at high operating loads. The engine used in the study had a large displacement single cylinder, which, although not using NVO, would have operated with a residual gas fraction of approximately 5%, effectively raising the mixture temperature to some extent. This could well have aided the gasoline auto-ignition combustion process. The study carried out in this paper was concerned with low speed and low load operation where engine operating temperatures are reduced and the air/fuel mixture is leaner.

In addition, a recent study of fuel effects in a diesel PPCI combustion regime comparable to that studied here, found that regular gasoline would not auto-ignite in a 15:1 CR engine without additional intake air charge conditioning, [11]. The requirement for intake air conditioning in order to operate gasoline PPCI despite fuel stratification could therefore be expected to be necessary in this study. The modifications made included a 2kW intake air heater with variable power control and boost pressure controlled up to a maximum of 1.05bar gauge using an external air compressor in combination with the turbocharger fitted to the engine.

CHARGE INTERCOOLING AND EGR

The requirement for increased intake air temperature meant that charge intercooling was not beneficial. All tests in this study were therefore carried out without an intercooler. It was possible to operate the engine without EGR while still maintaining relatively low emissions during low load operation in this study as EGR was not used.

FUEL INJECTION SPECIFICATIONS

A conventional diesel high pressure common rail injection system was used with conventional, 6 hole VCO, 154° included spray angle nozzles with a flowrate of 380cc/30sec at 100bar. In contrast to diesel PPCI, the use of high injection pressure to aid premixing was unnecessary with gasoline fuel and therefore a low rail pressure (compared with conventional diesel common rail systems) of 275bar was used throughout this study (similar to the pressure suggested for some GDI concepts).

EXPERIMENTAL RESULTS

TEST POINTS

The objective of studying low speed and low load operation was approached by studying two operating points, 1000rpm at 1bar BMEP and 1200rpm at 2bar BMEP. A limited excursion into a 900rpm, 1bar BMEP operating point was also carried out in order to approach near idle speed operation. The study of medium and high load operation was not considered beyond a discussion of the hardware implications.

ESTABLISHMENT OF GASOLINE AUTO-IGNITION

The engine was initially motored at an engine speed of 1000rpm to enable the effect of intake pressure, temperature and fuel stratification to be evaluated. The latter was achieved by multiple injections during the compression stroke. A maximum of two injections per cycle were used, with variation of the respective injection duration split as well the separation of the start of injection (SOI) of the two injections.

It was found that gasoline auto-ignition did not occur when HCCI combustion was attempted with a very advanced injection earlier than 50degCA BfTDC despite using an intake pressure and temperature of 0.95bar gauge and 95°C. The charge temperature and/or pressure would need to be raised significantly before a fully premixed gasoline charge would ignite in this manner, at which point the combustion process could be described as HCCI.

Single Injection

The limits of single injection event gasoline auto-ignition at 1000rpm, 1bar BMEP, with intake pressure of 0.95bar gauge and 95°C were found to be confined to combustion instability caused mainly by late combustion, leading to the onset of fuel consumption increases.

NOx formation was reduced during single injection event tests with early injection, reaching the detection limits of the gas analyzer as shown in fig 3. The level of HC and CO emissions increased with increased levels of premixing and the level of indicated specific fuel consumption increased with start of injection advance as the fuel conversion efficiency reduced. Fuel consumption

also increased with retarded SOI due to incomplete combustion later in the expansion stroke. A minimum level of fuel consumption combined with the lowest COV of IMEP was found with a SOI of 15degCA in this case.

An increase in indicated specific fuel consumption and small reduction in stability (indicated by COV) was shown by advance of SOI from 15 to 20degCA BfTDC and NOx emissions were reduced to less than 20ppm.

The variation in COV between cylinders did increase for both late and early SOI settings. A given level of partial premixing was therefore desirable, maintaining NOx emissions at very low levels rather than at a zero level whilst maintaining stable combustion and minimizing fuel consumption.

The use of an intake manifold temperature of 95°C and 0.95 bar gauge in this study can be compared with [10], where an intake temperature of 40°C and pressure of 1bar gauge were used. This shows that the limitations of gasoline auto-ignition are increased at low loads as a result of lower gas temperature, pressure and the inherently leaner air/fuel mixture.

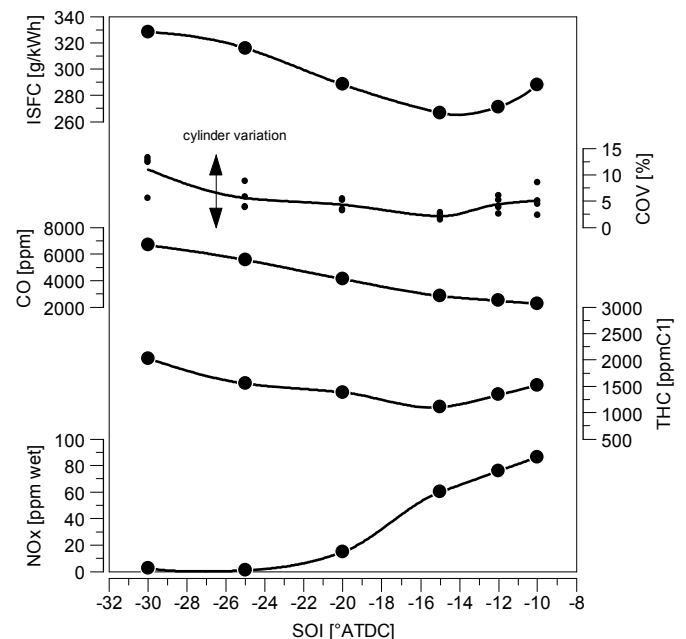


Fig 3 Single injection SOI, 1000rpm, 1bar BMEP

CHARGE TEMPERATURE AND PRESSURE

The limits of operation at lower temperatures and pressures were investigated by varying the power supplied to the intake air heater and reducing the boost pressure supplied by the external compressor.

The effect of reducing intake temperature whilst maintaining 0.95bar gauge intake manifold pressure was investigated first. A single injection event at 20degCA BfTDC with an engine speed of 1000rpm was used. The fuelling was set to give 1 bar BMEP with an intake temperature of 95°C. The fuelling was not readjusted