

Insights into Cold-Start DISI Combustion in an Optical Engine Operating at -7°C

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ABSTRACT

Particulate Matter (PM) emissions reduction is an imminent challenge for Direct Injection Spark Ignition (DISI) engine designers due to the introduction of Particulate Number (PN) standards in the proposed Euro 6 emissions legislation aimed at delivering the next phase of air quality improvements.

An understanding of how the formation of combustion derived nanoparticulates in engines is affected by the engine operating temperature is important for air quality improvement and will influence future engine design and control strategies. This investigation has examined the effect on combustion and PM formation when reducing the engine operating temperature to -7°C .

A DISI single-cylinder optical research engine was modified to simulate a range of operating temperatures down to the proposed -7°C . A high-speed 9 kHz optical investigation of the in-cylinder combustion and fuel spray along with in-cylinder pressure measurements was completed with the engine motored and fired at 1500 rpm during homogeneous and stoichiometric combustion conditions.

Results show striking differences between the flame growth structures at various operating conditions with the notable presence of significant fuel-rich regions which are understood to be prominent areas of PM formation. Measured engine performance parameters such as Indicated Mean Effective Pressure (IMEP) and Mass Fraction Burned (MFB) times correlated with the observed differences in combustion characteristics and flame growth speed. Flash boiling of the fuel spray was present in the fully heated engine case and significantly reduced the penetration of the spray plume and the likelihood of piston crown and cylinder liner impingement.

A clear link was shown between operating temperature, engine performance and in-cylinder combustion parameters which contribute to the formation of PM.

CITATION: Efthymiou, P., Davy, M., Garner, C., Hargrave, G. et al., "Insights into Cold-Start DISI Combustion in an Optical Engine Operating at -7°C ," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-1309.

INTRODUCTION

Today's gasoline combustion systems are able to fulfill the requirements of all current global emissions standards due to successful engine optimization and the effective utilization of the three-way catalyst.

The subject of cold-start performance in DISI engines has become an increasing focus in research to date due to engine particulate emissions performance when considered against future legislation [1,2,3].

It is understood that the highest contribution of PN emissions in DISI engines occurs within the first 100 seconds of the Euro 5 New European Drive Cycle (NEDC) [1] when an engine is still at cool operating temperature and during the

first transient events as shown in [Figure 1](#). This is further compounded by the fact that the three-way catalyst system has not yet achieved light-off temperatures.

The fundamental design of DISI engines, in particular their injection of fuel in-cylinder, reduces the time for homogeneous air-fuel mixture preparation which is one of the most important processes in ensuring a successful combustion system [4]. High levels of turbulence, created from the dissipation of bulk air flow [5,6] are essential in enhancing the air-fuel mixing process and improving the evaporation of fuel which has impinged on to cold surfaces [7]. Low-speed and transient engine operating conditions during startup can inherently affect these levels of controlled

turbulence and contribute to poorer charge mixture preparation.

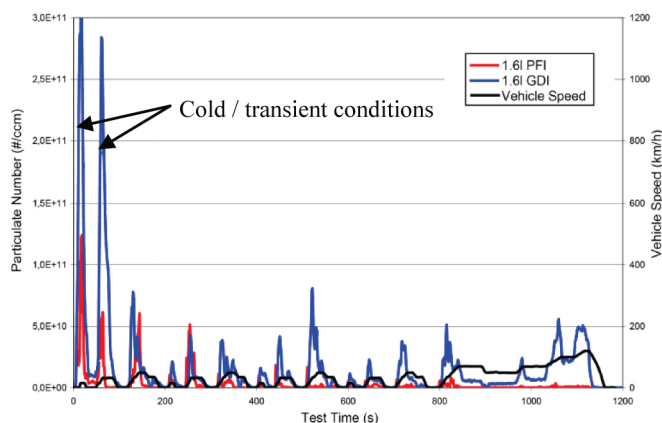


Figure 1. PN emissions plot during the Euro 5 NEDC comparing a DISI engine and Port Fuel Injected (PFI) engine (adapted from [1])

PM emissions in DISI engines are trimodal, with particle size modes being classified as nuclei, accumulation and coarse. Particles in the nuclei mode consist of particles with diameters < 50 nm which are primarily formed during combustion and dilution, usually by means of both homogeneous and heterogeneous nucleation mechanisms [8]. The nuclei mode form the majority of particles created in-cylinder with their formation occurring in fuel-rich region, illustrated by regions of high luminosity soot incandescence.

Accumulation mode particles are generally formed through the agglomeration of nuclei mode particles. Both the peak magnitude and size distribution change significantly with engine operating condition [8].

A number of studies [9,10] have cooled an engine to below 0°C in order to understand the effect on exhaust-out PM emissions. There is, however, a lack of literature that analyses the effect of in-cylinder combustion and PM formation regions.

This paper reports for the first time detailed in-cylinder flame structures and fuel spray injection of an optical DISI engine cooled down to a range of temperatures including -7°C .

By understanding the modes of nanoparticulate generation at cold engine operating temperatures and the effect on engine performance, strategies can be utilised by engine designers and calibrators to minimise PN and PM emissions as well as maintaining efficient engine cold-start combustion performance.

EXPERIMENTAL SETUP

A single cylinder optical research engine with a similar combustion system to the Jaguar AJ133 V8 [11] was used for all experimentation (Figure 2). A number of the core engine components such as the cylinder liner and piston crown window were interchangeable between optical and

thermodynamic configurations depending on the degree of optical access required. Table 1 gives the basic engine configuration details and valve timings during the intake stroke.

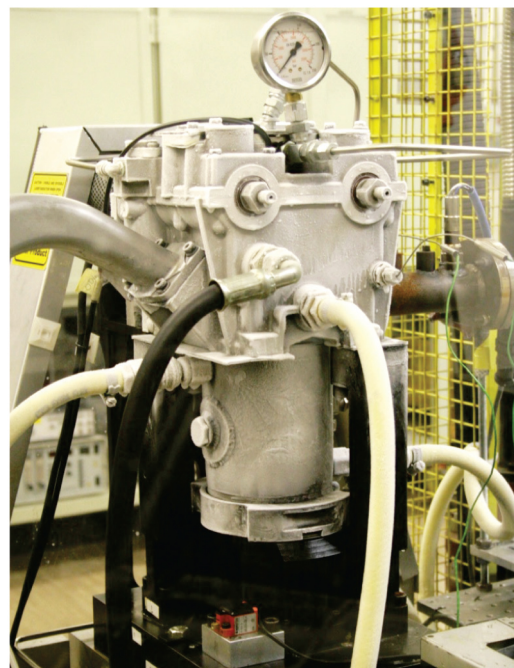


Figure 2. Jaguar optical engine cooled down to an operating temperature of -7°C

Table 1. Single cylinder optical engine configuration details

Engine Speed	1500 rpm
Manifold Pressure	500 mbar
Bore	89.0 mm
Stroke	90.3 mm
Compression Ratio	10.5:1
Number of Valves	2 Intake, 2 Exhaust
Fuel Delivery	6-hole direct injection
Fuel Pressure	150 bar
Valve Timings (ATDC)	IVO 24° , IVC 274° EVO 475° , EVC 6°
Injection Timing (ATDC)	80°
Ignition Timing (ATDC)	325°
λ Ratio	1.0

Optical Engine Configuration

Speed control

A motoring dynamometer was used to maintain the engine at a constant desired speed with a tolerance of ± 2 rpm. The timings of the fuel injection, ignition and data acquisition events were controlled manually with an AVL engine timing

unit and the engine crank angle and speed was calculated from a rotary encoder with a precision of 0.1 °CA

Fuel injection and ignition

Fuel injection was provided by a 6-hole prototype injector mounted vertically and located centrally within the cylinder head, in close proximity to the spark plug. The injector nozzle consisted of two groups of three nozzle holes, all have a 0.5 mm outlet diameter. The six holes each have a different injection angle, producing the distinctive spray pattern, illustrated in (Figure 3). The individual injection angles for each fuel plume was designed to achieve maximum spray coverage over the cylinder bore necessary for mixing with the intake air prior to ignition timing. Plumes 1 and 6 are orientated such that they passed around the spark plug in-order to minimise wetting of the electrode during fuel injection.

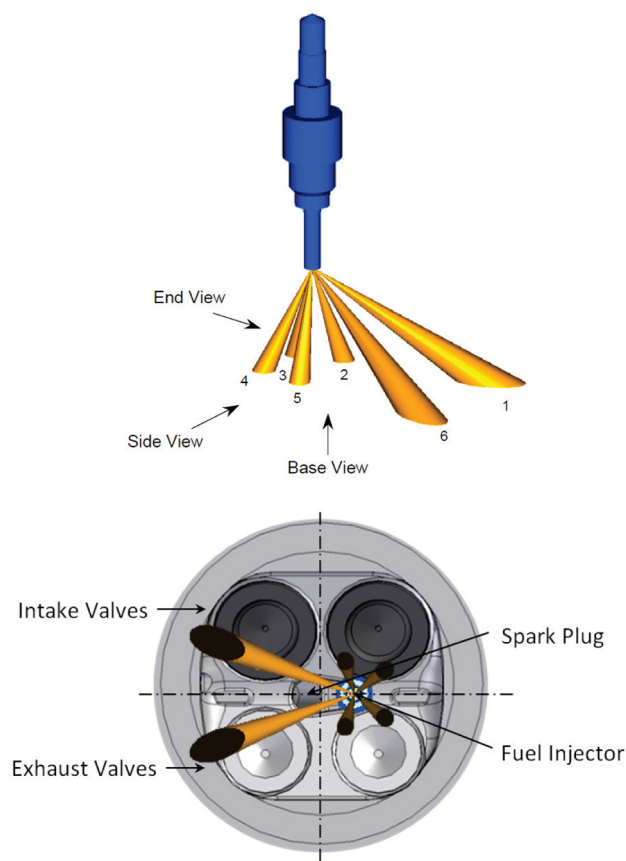


Figure 3. Schematic of six-hole fuel injector showing orientation of spray plumes around cylinder head (adapted from [12])

A coil-on-plug type ignition system was used with a 12 V, 9A supply. The orientation of the spark plug electrode was maintained perpendicular to the pent roof window as to not obscure the initial flame kernel growth when view through the pent roof window and to maintain the default orientation used in previous research on the engine [5,7].

The fuel injection and spark ignition timing events were kept constant throughout the experimentation at 80 °CA and 325 °CA ATDC respectively. This represented a Jaguar Land Rover standard test point which is optimised for homogeneous stoichiometric combustion.

Cylinder head and liner temperature

A 6 kW vapour chiller was required to cool an antifreeze-water mixture to -20°C and achieve an operating condition of -7°C . This was connected to the coolant circuit which flowed through the cylinder head and metallic liner. When an operating temperature above ambient conditions was required, the coolant circuit was connected to a tank that heated the fluid using a 3 kW thermostat-controlled immersion heater. The operating condition temperature was defined by a thermocouple embedded into the cylinder head on the exhaust-side of the engine away from the coolant circuit; this ensured the recorded temperature was representative of all of the cylinder head components. The thermocouple was coated with a silicone-based thermal compound to achieve good thermal contact with the cylinder head surface.

Intake air temperature

A vortex tube device was used to cool the intake air temperature of the engine and simulate a cold-soak test at -7°C (Figure 4). The device was connected to a 6.0 bar compressed air line which supplied a cool air stream of -7°C , this was adjustable using an exhaust valve. The cooled air was fed into the intake plenum of the engine to allow stable measurement of the air temperature using a thermocouple located inside the plenum. The device operated using a tangential compressed air intake to create a high-speed vortex across the tube. Energy from the centre of the vortex was transferred to the high-pressure outer vortex in the form of heat, which escaped through the centrally-restricted exhaust geometry whilst the cooled vortex stream is escaped from the unrestricted orifice at the opposite end of the tube.

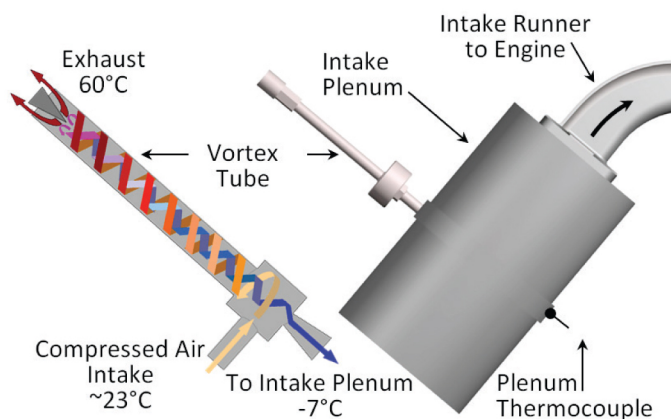


Figure 4. Outline of vortex tube arrangement intake plenum of engine

Data Acquisition

In-cylinder pressure measurement

The in-cylinder pressure across the entire engine cycle was measured using a Kistler piezo-electric transducer mounted within the pent-roof combustion chamber. The transducer measured relative pressure change so a reference pressure was also required, this was recorded using a piezo-resistive transducer installed in the lower barrel of the cylinder liner and measured an absolute pressure at Bottom Dead Centre (BDC). Pressures throughout the entire engine cycle were recorded at a temporal resolution of 0.1 °CA.

Image Acquisition

Combustion imaging

In-cylinder combustion imaging was obtained via the quartz optical piston crown window using a Photron FASTCAM APX-RS high-speed camera viewing across the 45° mirror located inside the extended Bowditch piston as shown in Figure 5. The camera recorded an imaging area of 640 by 480 pixels at a rate of 9 kHz producing a spatial resolution of 0.142 by 0.142 mm/pixel and allowing a high temporal resolution of 1 °CA at the tested engine speed of 1500 rpm.

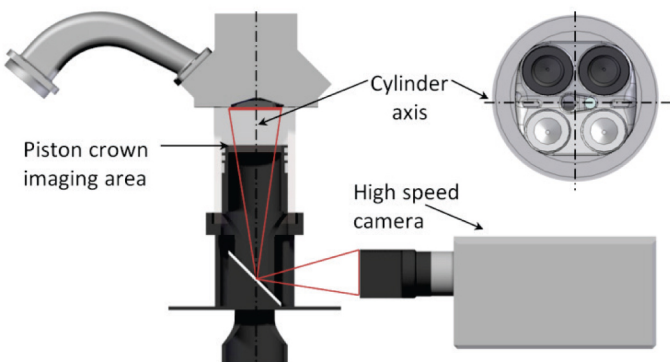


Figure 5. Experimental setup of combustion imaging through optical piston crown

The camera memory was partitioned to allow 80 frames per trigger to be captured, with the camera trigger occurring 5 °CA before spark ignition. Each test run recorded 85 engine cycle events which was limited by the 2GB on-board memory of the camera, this produced a total of 6800 images through the optical piston crown. Figure 7 illustrates the location of the camera trigger and image capture time relative to the pressure curve during the engine's cycle ATDC relative to the intake stroke.

Fuel-spray visualisation

The fuel spray was imaged through the quartz optical cylinder liner with the camera positioned perpendicularly to the cylinder liner. Illumination of the combustion chamber was provided by a Copper Vapour Laser, capable of

producing a pulsed repetition rate of 50 kHz with pulse duration in the region of 10-30 ns. The camera and laser were synchronised at 9 kHz to allow for a high temporal resolution of 1 °CA per frame with illumination energy in the order of 2 mJ per pulse. The copper vapour laser used an optical fibre system in order to allow for greater position flexibility, the beam was reflected on the 45° mirror below the extended Bowditch piston and through the piston crown. By using an illumination source through the piston crown window, reflection was minimised from the optical cylinder liner which reduced the presence of optical artefacts in the recorded images and improved the accuracy of post-processed groups of fuel spray images.

A system using a traversable spherical lens was used to reduce the divergence of the conical beam as it exited the fibre and maximise the energy density of the laser by restricting the beam's diameter to that of the optical piston crown. The 'flood-fill' illumination setup of the beam allowed all injector plumes to be detected, the camera was then focused on the central axis of the cylinder in line with the injector. Figure 6 outlines the optical arrangement of the illumination system and high-speed camera on the optical engine.

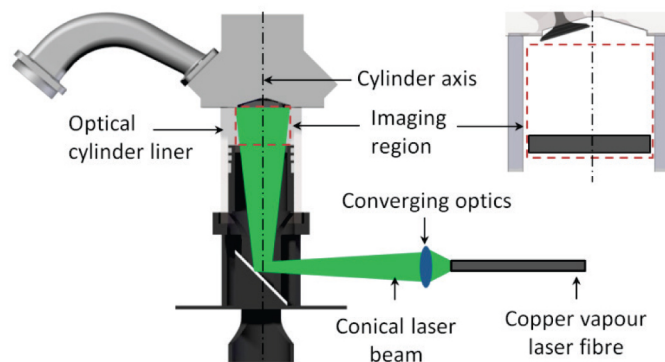


Figure 6. Experimental setup of fuel-spray visualisation through optical cylinder liner

The camera recorded an imaging area of 640 by 480 pixels at a rate of 9 kHz fps producing a spatial resolution of 0.124 pixels/mm and again allowing a high temporal resolution of 1 °CA at the tested engine speed of 1500 rpm. A Nikon lens with a focal length of 60 mm was used with a maximum aperture setting of f2.8 to maximise the detection of scattered light from the fuel spray without the requirement for any post image enhancement. The camera memory was partitioned to allow 70 frames per trigger to be captured, with the camera trigger occurring 10 °CA before fuel spray injection (see Figure 7). The increased delay time of the image trigger before the injection event allowed any liquid fuel deposits present in the cylinder from the intake ports to be detected during intake.

The engine was motored continuously with fuel injection and without spark ignition. This allowed the conditions over a single test run of 97 cycles to be kept as consistent as possible

by avoiding the high temperature build-up from combustion, which would ultimately have an impact of the fuel spray. Compressive heating and friction were still considered, but their effect would be far less than the energy released during combustion and this heating was considered to be offset by the evaporative cooling effect of the evaporating fuel spray.

A fuel-spray duration of 0.85 ms was used consistently throughout testing, which represented an Air-Fuel λ ratio of 1.0. Using consistent fuel spray durations across all tested conditions also ensured the same fuel quantity was delivered in every case allowing consistency when comparisons were made between conditions.

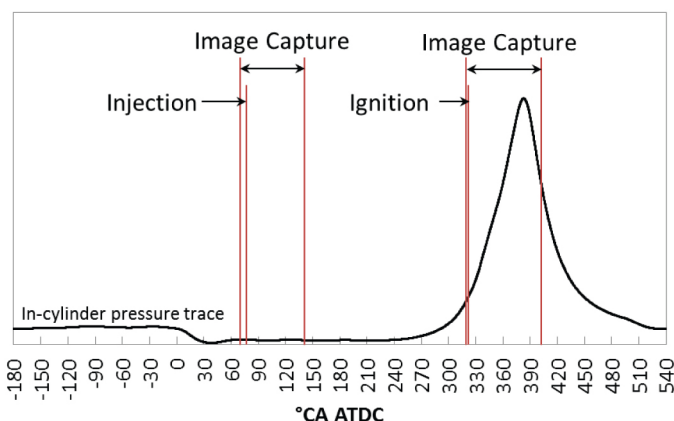


Figure 7. Timing graph outlining camera trigger events during injection and combustion

Fuel-spray image processing

The fuel spray structure over complete engine runs was analysed using image processing for up to 300 cycles. This produced a contour plot highlighting the statistical average of the location of the fuel spray in-cylinder. This allowed the average fuel-spray penetration and atomisation structures to be directly compared across different engine operating conditions. The first 25 °CA (80° to 105 °CA ATDC) after the fuel injection event were used as this captured the time up to piston-crown impingement.

Experimental test points

To observe the fundamental effect that engine operating temperature has on injection, combustion and in particular, flame propagation, a range of operating temperatures were tested as shown in Table 2. These ranged from 80°C, a typical fully heated engine test condition, to -7°C, the soak temperature for one the European emissions tests. An ambient laboratory temperature of 23°C was tested as well as an intermediate temperature of 10°C to further understand the effects of the transition to a negative operating temperature.

Table 2. Summary of test points outlining cylinder head, liner and intake temperatures used

Test Point	Head and Liner Temperature (°C)	Intake Air Temperature (°C)
1	80	23
2	80	-7
3	23	23
4	23	-7
5	10	23
6	10	-7
7	-7	23
8	-7	-7

RESULTS AND DISCUSSION

Quantitative data will first be presented from the recorded and processed in-cylinder pressure trace to show the effect of operating temperature on engine performance. An optical analysis of the in-cylinder combustion and fuel spray will then be presented to link key findings in engine performance to the observations made.

In Cylinder Pressure Measurements

The upper graph in Figure 8 outlines the in-cylinder pressure relative to after Top Dead Centre (ATDC) of the compression stroke as well as the Mass Fraction Burned (MFB) profile for all head and liner temperatures tested with an intake temperature of 23°C. There is clear and strong correlation evident between the engine operating temperature and the peak cylinder pressure achieved with a range from 13.16 bar at 80°C, to 8.79 bar at -7°C. The location of the peak pressure in the engine cycle remained relatively consistent at 23 °CA with a maximum deviation of 2 °CA. Notably, the difference in peak pressure increased as operating temperature deviated from 80°C, with a less significant difference in peak pressure evident between the 10°C and -7°C peak pressures.

The MFB profiles presented show the change in pressure with crank angle ($dp/d\theta$) and can be used to represent the speed of the flame from spark ignition to the complete consumption of the unburned charge. Although similar ignition delays were observed between operating conditions, the MFB rates were lower and show longer times for the charge to be burned at colder conditions. These key differences can be used to further explain the differences observed in peak pressure. The increased rate of pressure rise and therefore energy release at warmer temperatures allowed a greater proportion of the charge to be burned by the time the piston reached TDC. After this point, the remaining charge was burned into an increasing cylinder volume and at a lower flame growth speed during the expansion stroke, resulting in a decrease in peak pressure.