

Effect of Injection Strategy on Cold Start Performance in an Optical Light-Duty DI Diesel Engine

Clément Chartier, Ulf Aronsson, Öivind Andersson, Rolf Egnell

Division of Combustion Engines, Lund University

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ABSTRACT

The present study investigates cold start at very low temperatures, down to -29 deg C. The experiments were conducted in an optical light duty diesel engine using a Swedish class 1 environmental diesel fuel. In-cylinder imaging of the natural luminescence using a high speed video camera was performed to get a better understanding of the combustion at very low temperature conditions. Combustion in cold starting conditions was found to be asymmetrically distributed in the combustion chamber. Combustion was initiated close to the glow plug first and then transported in the swirl direction to the adjacent jets. A full factorial study was performed on low temperature sensitivity for cold start. The effects of cooling down the engine by parts on stability and noise were studied. Furthermore, different injection strategies were investigated in order to overcome the limited fuel evaporation process at very low temperatures. A strategy with 3 pilot injections and increasing fuel quantity for each injection gave a more homogeneous combustion. The natural luminescence was symmetrically distributed in the piston bowl and all the jets contributed to the combustion. Significant improvements in combustion stability, load level and potential for load increase were observed as well.

INTRODUCTION

Cold start at very low ambient temperatures is an important aspect of diesel engine development. At ambient temperatures down to -25 °C it is difficult to auto-ignite the charge, and even more so to ignite it in every cycle. The latter problem is referred to as intermittent combustion [1]. Cold starting represents only a minor part of the engine utilization. Demands on cold starting performance may still compromise the development of other aspects of the engine.

There is a general trend towards lower compression ratios. This is due to ever higher specific powers [2] and more stringent emission legislations, especially for NO_x. This poses even greater challenges for cold start performance.

Lower compression ratios mean lower pressure and temperature levels in the cylinder at start of injection (SOI). This leads to fuel evaporation issues. It has been demonstrated that as much as 80% of the injected fuel mass does not burn during a cycle [3]. Most of it adheres to the cold surfaces and is difficult to vaporize.

A typical cold start for diesel engines is presented in Figure 1 and can be divided into five phases. First, the glowplug is glowing during a given period, here 10 seconds. After this, the starter motor is activated and the engine is cranked up to 200 rpm until the combustion takes over. Cranking time is longer at lower temperatures. At engine start an engine speed overshoot is typically observed. It is due to a steep load increase during the first combustion cycles. Load can be in the range of 8-11 bar IMEP in this third phase.

Friction losses are very high due to the low temperature of the lubricant in the engine and gearbox. The engine speed is then unstable for a limited period of time where load and combustion phasing variations are large. During this after-start phase loads typically average 4-5 bar IMEP. It is this specific phase that is investigated in the present study. The last phase commences when engine speed stabilizes at the nominal idle speed.

The glowplug plays a central role in cold starting. It locally heats up the air which speeds up the fuel evaporation and starts the fuel oxidation process. Today, ceramic glowplugs are becoming increasingly common. They provide higher temperatures (>1200°C) than metal glowplugs (~950°C). Built-in controllers make them reach a stable working temperature [4].

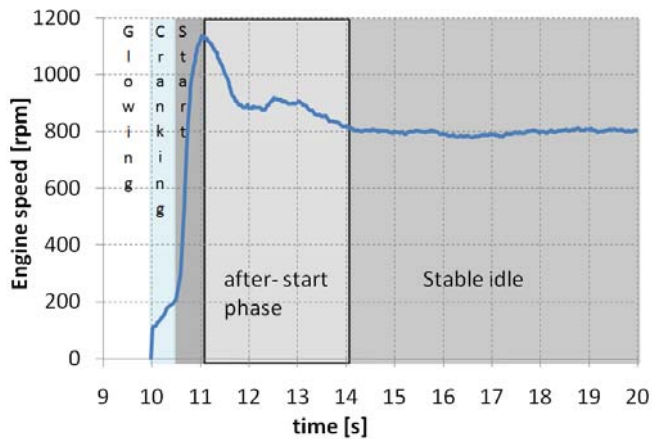


Figure 1 A typical cold start sequence.

Most Diesel engines are equipped with sophisticated injection systems. Especially common rail systems provide a large flexibility in the choice of injection strategy for a wide variety of conditions. All phenomena caused by advanced injection strategies are not fully understood. A number of investigations have demonstrated that adjusting the injection strategy can compensate for lower compression ratios during cold start [3,5,6]. Split-main injection strategies may improve startability (i.e. reduce cranking time) and at the same time reduce the injected fuel mass and unburned hydrocarbon emissions by almost 50% [7]. Although cold startability is improved, poor after start stability may still be a problem [6].

A certain fuel mass is required to start the engine. However, increasing the injected fuel mass above a certain level may deteriorate the starting performance. Contrary to intuition, more fuel mass may lead to lower loads and unstable combustion [8]. This problem is due to fuel evaporation issues and must be solved in order to quickly reach idle speed.

Overfuelling leads to a successively increasing effective compression ratio, since some fuel is left in the piston bowl from cycle to cycle. Eventually, the highly fuel rich mixture ignites and consumes most of the fuel leading to violent intermittent combustion [9]. The following cycle again has too low compression ratio leading to misfire or low load during a few cycles.

No study dedicated to combustion behavior and stability after start has been found in the literature. Apart from reduced cranking time, the stability of the engine speed after start and the potential load increase are important aspects of a successful cold start. This study consists of three parts:

- The first part investigates the differences between starts at normal and low temperature conditions for a baseline case. This is done to extract the main characteristics and patterns from each case.

- In the second part a designed experiment is used to separate the effects of three factors on after start stability. The factors are the engine block temperature, the fuel temperature, and the inlet air temperature.
- In the third part the effects of the injection strategy on after-start stability is investigated. Six different strategies are investigated and a best case is proposed based on a thermodynamic analysis.

EXPERIMENTAL SETUP

OPTICAL ENGINE SETUP - The study was conducted in a on optical HSDI diesel engine based on the Volvo D5. It featured a Fused Silica piston with realistic bowl geometry as seen in Figure 2. The piston rings are made of Fluorlon 412 for dry lubrication usage. The engine was equipped with a piezo-electric common rail injection system from Bosch. The rail pressure was kept constant at 460 bar for all cases studied. This pressure level is typical for start conditions. Engine operating conditions are presented in Table 1. The glowplug used in this study was a ceramic, 7V type from Bosch. Swedish environmental Class 1 Diesel fuel, MK1, was used. The distillation curve is presented in Figure 3.

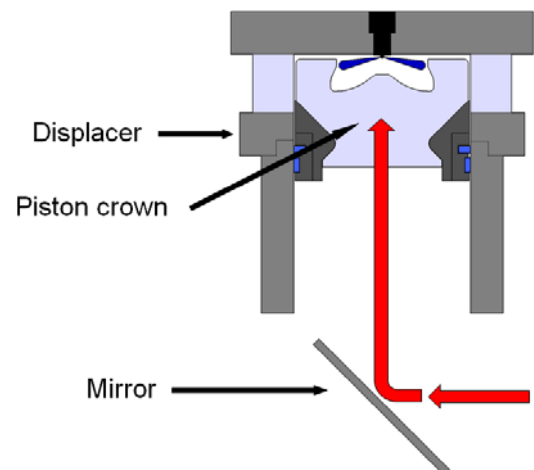


Figure 2 Schematic of the optical access on the D5.

Images of natural luminescence were recorded using a high speed video Phantom V7.1. The natural luminescence mainly originates from soot radiation. The camera was equipped with a UV-Nikkor lens of focal length 105mm and f/2.5. The frame rate was 7200 frames per second, equivalent to one image per crank angle degree (CAD) at 1200 rpm. The camera resolution was 512x512 pixels and an 80μs exposure time was used. This exposure time made it possible to detect the first combustion luminosity. Furthermore, with appropriate lightning, the liquid sprays could be observed as well.

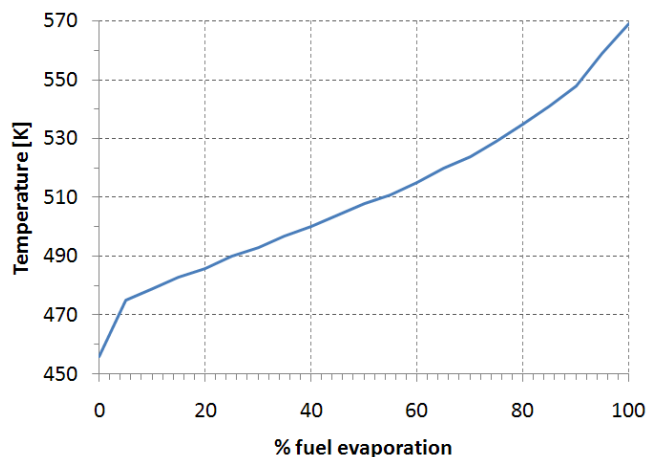


Figure 3 Distillation curve for MK1 fuel.

Figure 4 shows the positions of the glowplug and pressure transducer between the inlet valves. In this engine, the pressure transducer is normally placed in the glowplug hole. The new location of the pressure transducer is not expected to have an impact on the pressure measurement.

Table 1 Engine and fuel injection system specifications.

Engine Specifications	
Number of cylinders	1
Number of intake valves	2
Number of exhaust valves	2
Displacement vol. [l]	0.48
Bore [mm]	81
Stroke [mm]	93.2
Connecting rod [mm]	147
Effective Compression ratio	16.8
Swirl ratio	2.0
Fuel injection system	
Type	Common-rail
Injector type	Piezoelectric
Number of holes	7
Included angle [°]	150
Orifice diameter [mm]	0.140
Rail pressure [bar]	460

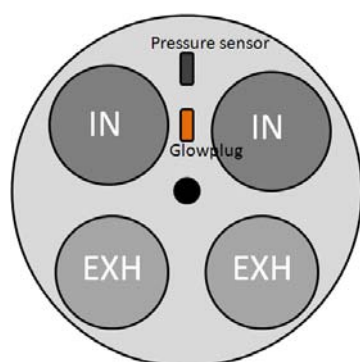


Figure 4 Schematic of the cylinder head with glowplug and pressure transducer positions.

COOLING CIRCUIT - To perform investigations at low temperatures, the engine cooling circuit was modified. A domestic freezer was filled with 160 liters of coolant liquid, consisting of 50% glycol and 50% water. This setup could provide a coolant temperature of -28°C.

To provide a fully flexible system, the cooling circuit was divided into three independent paths:

- The cylinder head and liner
- The fuel
- The intake air

To reproduce cold ambient conditions, the coolant was circulated in the cylinder head and liner by means of a pump submerged in the freezer. The fuel was cooled down between the fuel tank and the high pressure pump using a copper coil submerged in the coolant bath. The low pressure fuel pump assured constant flow through the cooling device. Finally, the intake air was taken from the lab's supply of dehumidified shop air. Dehumidified air prevents ice clogs from forming in the hoses or heat exchangers. The air was cooled using two EGR-coolers mounted in series, submerged in the coolant bath. Coolant was circulating through the EGR-coolers by means of the submerged pump.

The fuel, inlet air or engine coolant flows could independently be set to bypass the cooling device at any time to run at normal conditions. All hoses going to or from the freezer were isolated using a thick rubber insulation hose. This increased the cooling capacity of the system and extended the time of use at optimal conditions.

To prevent frost from forming on the mirror surface and optical piston bottom (see Figure 2) dehumidified shop air was constantly blown towards the mirror.

TEST PROCEDURE – The investigated phase of cold start is after-start stability. Therefore, the engine was not cranked as shown in Figure 1, but motored at 1200 rpm prior to first injection. To obtain a stable rail pressure at first injection, the engine was run for a short period before cooling down the system. During this time the rail pressure was adjusted to 460 bar, and it remained at this level during cool-down.

Figure 5 shows the temperature history during the cooling down period. The optimal cooling down duration turned out to be 25 minutes. After this the temperature of the displacer started to increase due to the engine working as a heat sink for the laboratory air.

Shop air was blown through the cooling system and escaped by a valve just before entering the inlet manifold. By doing so, the hoses were cooled down and low temperatures could be obtained from the start. After 20 minutes of cooling down, the mirror and piston elongation were cleaned from ice.

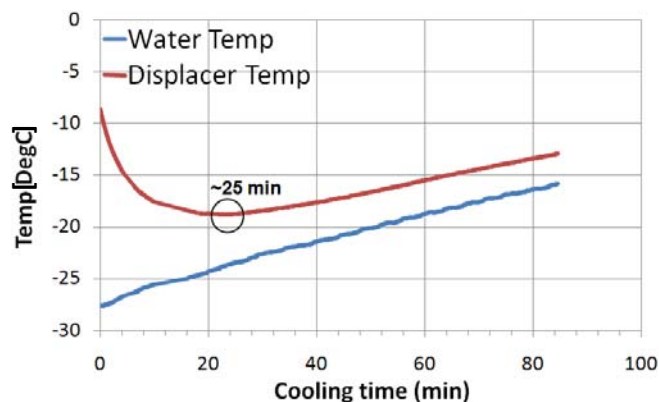


Figure 5 Temperature of the coolant and of the displacer during the cooling down process.

START PROCEDURE – After the cooling down procedure described above, the engine was motored for approximately 45 seconds in order to adjust the common rail pressure to the desired level. Then, fuel injection was switched on and pressure traces were acquired. No skip fire strategy was employed, i.e. every cycle was fired. The first ten fired cycles are presented here in the results. The coefficient of variation for the indicated mean effective pressure is calculated from these ten cycles.

RESULTS AND DISCUSSION

NORMAL VS. LOW TEMPERATURE START – Figure 8 shows images extracted from a high speed video movie of an engine start at 20°C. The case presented is a baseline with two pilot injections and a main injection. The total injected fuel mass of 12mg is divided between the injections as follows: 30% in pilot 1, 30% in pilot 2, and 40% in the main injection. The injection timing corresponds to the fourth from the top in Figure 17. One of the sprays was aimed directly in the direction of the glowplug during this comparison. The number indicated in the upper left corner of each frame is the crank angle position at which the image was acquired. The glowplug position is indicated in the first frame. The glowplug seems to have little influence on the onset of combustion. At -2 CAD, multiple ignition kernels can be seen throughout the combustion chamber. The seven burning jets are then developing equally fast.

Figure 9 shows an engine start with the same injection strategy as in Figure 8, but at -20°C. There are large differences between the two cases. Visible reactions occur 2 CAD later in the cold case at a position next to the glowplug, downstream in the swirl direction. They remain close to the glowplug and later spread in the swirl direction. The combustion does not spread to other jets and involves only a small portion of the piston bowl. Six jets are not involved in the combustion process, probably due to poor evaporation in these regions.

Swirl seems to be a factor of interest when looking at the transport of the reacting area and mixture passing the glowplug during the combustion. To investigate the effect of swirl, the swirl ratio was switched between 2.0 and 2.6 by means of a control valve in one of the inlet ports.

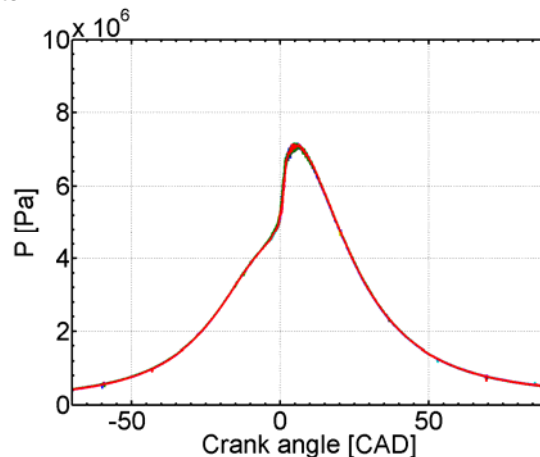


Figure 6 Pressure trace at normal temperature for the baseline case.

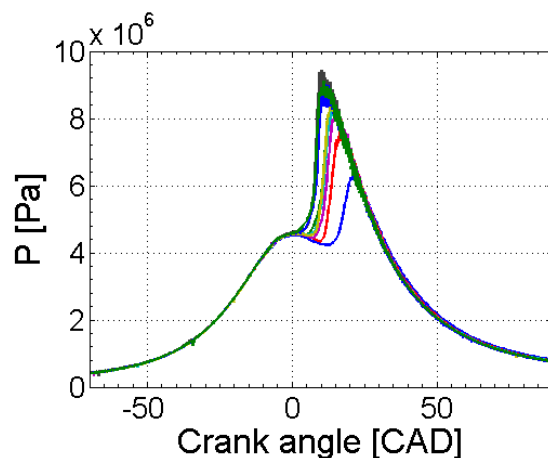


Figure 7 Pressure trace at low temperature for the baseline case.

The sequence presented in Figure 10 represents the same case as in Figure 9 but at the higher swirl ratio. It shows no major difference to the prior sequence. Aiming one spray directly towards the glowplug is, however, not typical of a production engine. A slight rotation of the injector against the swirl direction might have produced different results.

Figure 6 shows pressure traces for 10 cycles acquired at the same conditions as in Figure 8. Apparently, the combustion is very stable from cycle to cycle. Figure 7 shows on the other hand pressure traces for the low temperature case. Cold conditions lead to a longer ignition delay and more late-cycle combustion compared to ambient temperature. The peak pressure is higher for cold conditions as a result of the longer ignition delay.