

An Experimental Investigation on Low Speed Pre-Ignition in a Highly Boosted Gasoline Direct Injection Engine

Xuwei Luo, Ho Teng, Tingjun Hu, Ruigang Miao, and Liming Cao
JiangLing Motors Co., Ltd.

ABSTRACT

The biggest challenge in developing Turbocharged Gasoline Direct Injection (TGDI) engines may be the abnormal combustion phenomenon occurring at low speeds and high loads, known as low-speed pre-ignition (LSPI). LSPI can trigger severe engine knocks with intensities much greater than those of spark knocks and thus characterized as super knocks. In this study, behavior and patterns of LSPI were investigated experimentally with a highly-boosted 1.5L TGDI engine. It was found that LSPI could occur as an isolated event, a couple of events in sequence, or a trail of events. Although occurring randomly among the engine cylinders, LSPI took place frequently when the engine was operated at low speeds and high loads in the zone where scavenging was employed for boosting engine torques at low speeds, typically < 2500 rpm. Frequencies and patterns for LSPI were found to be influenced considerably by engine operation parameters, such as excess air coefficient, engine coolant temperature, valve timing, etc. Intensities of super knocks triggered by LSPI varied with the timing for the pre-ignition. Based on findings in this study, it was proposed that the oil particles as oil loading in the blowby recirculation could be the root cause for LSPI as they cause dirtiness of the combustion chamber roof where the oil particles could be attached and then initiate LSPI in the late compression stroke. It was demonstrated that although LSPI may not be eliminated due to the nature of the triggers, TGDI engines can be operated super-knock free through controlling conditions for combustion and engine cooling. As long as the values of the maximum peak firing pressure in LSPI events and their frequencies are under the design criteria, the engine can tolerate the LSPI events with acceptable knock intensities.

CITATION: Luo, X., Teng, H., Hu, T., Miao, R. et al., "An Experimental Investigation on Low Speed Pre-Ignition in a Highly Boosted Gasoline Direct Injection Engine," *SAE Int. J. Engines* 8(2):2015, doi:10.4271/2015-01-0758.

INTRODUCTION

Turbocharged Gasoline Direct Injection (TGDI) is an effective method for improving fuel economy of gasoline engines because the engine can achieve high torques at low engine speeds and thus can be downsized. Stoichiometric mixture combustion mode is widely employed in TGDI engines because it allows gasoline engines using a high-efficiency three-way catalyst converter (TWC) for after-treatment. For highly-boosted TGDI engines with homogeneous mixture combustion, pre-ignition (PI) is frequently encountered, typically at speeds < 2500 rpm and specific torque > 120 Nm/L or brake mean effective pressure (BMEP) > 15 bar. This abnormal combustion is termed as low-speed pre-ignition (LSPI) in the literature [1, 2, 3, 4, 5, 6, 7, 8, 9]. Because LSPI often leads to a severe knock combustion with cylinder pressure fluctuations on the same order of magnitude of the peak firing pressure (PFP) in a normal combustion, it is also termed as super knock or mega knock [3,4].

Figure 1 compares cylinder pressure traces for normal combustion and abnormal combustion with LSPI triggered super knock at 1500 rpm and full load for a highly-boosted 1.5L TGDI engine tested by

the authors of this paper. In Figure 1, 0 deg-CA indicates the crank angle is at the top dead center (TDC) for the compression stroke. It is seen that the magnitude of pressure fluctuations in the super-knock combustion is over 100 bar, exceeding PFP for the normal combustion. Figure 2 illustrates relative crank-angle windows for the fuel injection, PI, spark timing and conventional or spark knocks. Because the PI event occurs much earlier than the spark timing for normal combustion, super knocks cannot be mitigated by changing the spark timing as in controlling spark knocks. To depress super knocks, it is needed to identify the PI event in an appropriate crank-angle window through the signals of the engine's knock sensor. Differently from the window for spark knocks, which is typically in the range 10~50 deg-CA after TDC, an appropriate window for identifying the PI event may be in the range -30~-10 deg-CA, i.e., before TDC as shown in Figure 2. If the PI takes place at -30 deg-CA, the super knock triggered may occur near TDC, with the strongest knock taking place before the knock window for normal combustions.

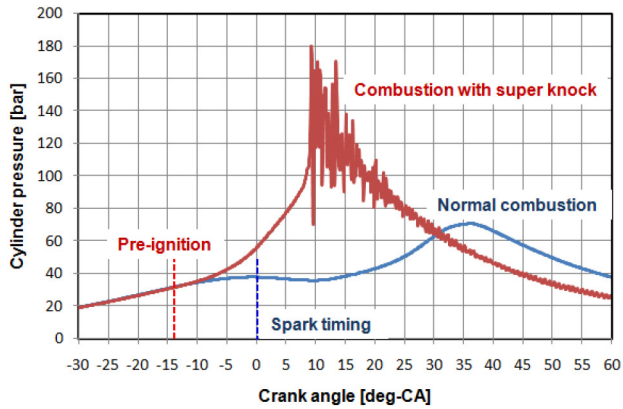


Figure 1. Cylinder pressure traces for normal and super-knock combustion for a highly-boosted 1.5L TGD engine at 1500 rpm.

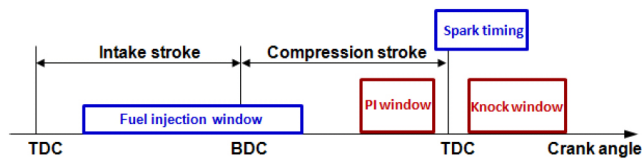


Figure 2. Illustration of windows for pre-ignition, spark timing and knock.

The cause for LSPI is widely believed to be the oil particles entering the engine cylinder [1, 2, 3, 4, 5, 6, 7, 8, 9]. However, the origins of the PI initiators or triggers and locations where PI is initially started are still not clearly understood. As was appointed out by Willand et al. [1], LSPI has become a limit on downsizing the spark-ignited gasoline engines. LSPI is a challenge to be faced in development of the highly-boosted TGD engines. It is neither possible nor practical to eliminate the oil particles completely from the engine cylinder; thus, a reasonable approach for mitigating super knocks is to weaken the conditions favoring LSPI. To that end, this study investigates the conditions that potentially lead to LSPI and methods to suppress super knocks triggered by LSPI.

CHARACTERISTICS OF TGD ENGINE OPERATED AT LOW SPEEDS AND HIGH LOADS

Achievement of Low-Speed Torque for TGD Engine with Scavenging

The engine under study is an inline 4-cylinder 1.5L TGD engine. The basic engine parameters are presented in Table 1. The engine is equipped with a high-efficiency, waste-gated turbo charger and dual cam phasers allowing both intake and exhaust cams to shift maximum of 60 deg-CA with respect to their lock positions independently. The engine is highly boosted and has a flat torque curve with low-end speed = 1500 rpm and high-end speed = 4600 rpm. Figure 3 shows the engine torque and power curves normalized with their peak values.

Table 1. Basic parameters for the TGD engine under study.

Engine configuration	I4
Engine displacement	1.5L
Bore	79 mm
Stroke	76 mm
Compression ratio	9.5:1
Number of valves per cylinder	4
Fuel injection equipment	Direct injection
Max injection pressure	150 bar
Injector mounting	Side mounted
Fuel	RON93 gasoline
Turbo flow control	Waste gate

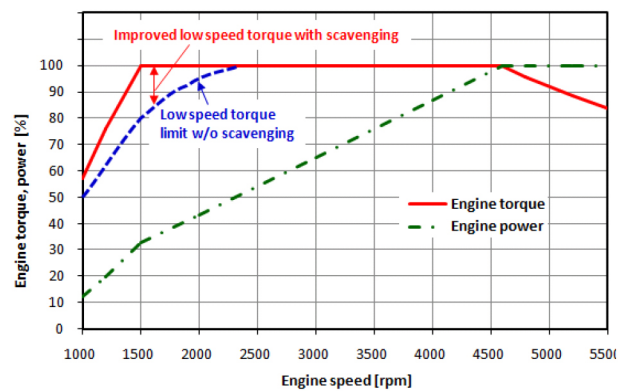


Figure 3. Torque/power curves for TGD engine under study.

As all turbo-charged engines, the engine's low-speed torque capability is limited by the surge line of the turbo compressor. In order to improve the low-speed-end torques, scavenging is employed for engine speeds ≤ 2500 rpm at high loads with help of variable valve timing (VVT). Scavenging enhances the engine torque at low speeds and extends the maximum torque down to 1500 rpm. The low-speed-end torques without and with scavenging are illustrated in Figure 3. At 1500 rpm, the torque boost by scavenging is up to 20%, similar to that at the same engine speed reported by Millo and Rolando [11]. Figure 4 shows the zone of scavenging in the load-speed map for the TGD engine under study. Measured intake and exhaust manifold pressures at full load are also presented. Within the scavenging zone on the top-left corner at speeds ≤ 2500 rpm, the intake port pressures are greater than the exhaust port pressures and thus a portion of the intake flow short-circuits the engine cylinder and flows to the exhaust ports during the valve overlap. Outside the scavenging zone, since the exhaust port pressures are always greater than those of the intake ports, some high-temperature exhaust gas may potentially flow reversely into the intake ports during the valve overlap. The zone for speeds and loads where LSPI was observed is also plotted in Figure 4. It is seen that the LSPI zone falls completely in the scavenging zone. The speeds and loads covered by the LSPI zone agree well with those reported by Haenel et al. [2]. The fact that LSPI occurs within the scavenging zone suggests that LSPI may be related to the scavenging flow.

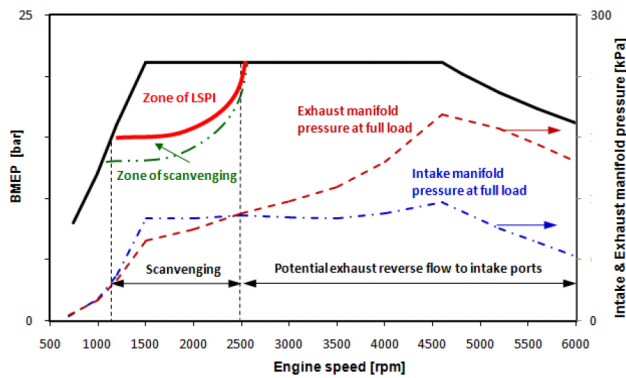


Figure 4. Zones of scavenging and LSPI for TGD engine under study.

Challenge to Fuel Delivery at Low Speeds for Highly-Boosted TGD Engine

The fuel injector for the engine under study is a 6-hole injector, side mounted and with maximum injection pressure being 150 bar. The window for the fuel injection is illustrated in Figure 2. The fuel spray targets on the piston top are illustrated in Figure 5. Figure 6 shows the normalized fuel delivery at full load. The fuel delivery at 1500 rpm is 80% that at 4600 rpm. However, as shown in Figure 7, the maximum piston speed (at about 45% of the piston stroke) at 1500 rpm is less than 1/3 that at 4600 rpm.

Visualization of patterns for the fuel sprays from the injector at 100-bar injection pressure in the test facility is presented in Figure 8. Figure 9 shows the measured spray penetrations vs. time under three different injection pressures, according to which, if the injection duration is longer than 2 ms, the tips of the fuel sprays would reach the piston top with an equivalent spray tip speed ≈ 40 m/s at injection pressures ≥ 100 bar. Although at low engine speeds the required fuel delivery can be split up to three injections, there is still a high tendency for the fuel to wet the cylinder wall under the influence of a strong intake tumble. Cylinder wall wetting would result in dilution of the crankcase oil by the fuel entered the oil.

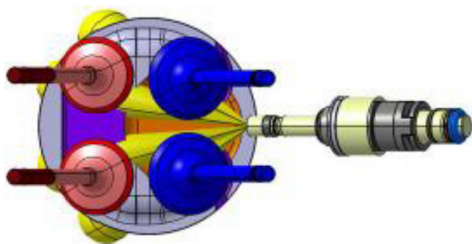


Figure 5. Side mounted fuel injector and the spray targets.

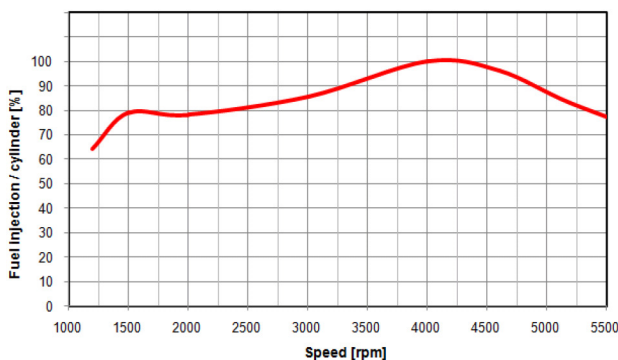


Figure 6. Normalized fuel injection per cylinder at full load.

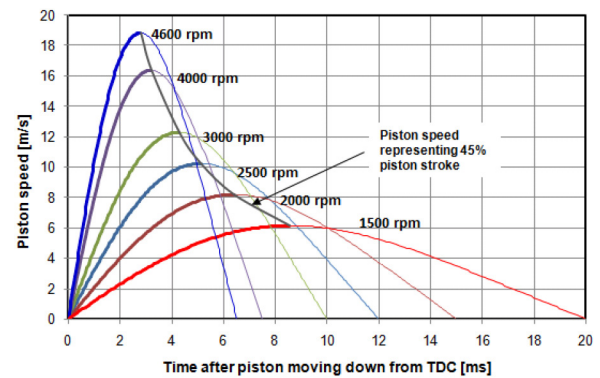


Figure 7. Piston speeds at engine speeds = 1500-4600 rpm.

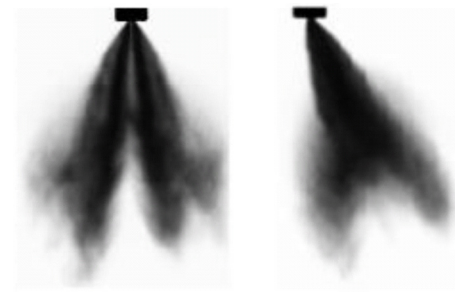


Figure 8. Patterns of fuel sprays at injection pressure = 100 bar viewed from two different directions of the test facility.

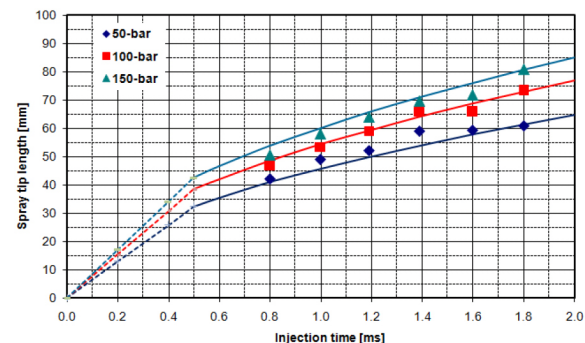


Figure 9. Measured spray lengths at different injection pressures.

Impact of Direct Injection on Fuel Dilution of Crankcase Oil

The crankcase oil for the TGD engine under the study is SAE 5W30 synthetic oil. Figure 10 shows the dilution of the crankcase oil in a cyclic load test. The profile for one cycle is also presented in Figure 10. In the test, both the oil and coolant temperatures were maintained at 95 °C with external conditioning systems. The oil dilution curve represents the period of the first 40 test cycles, during which the oil was sampled in an interval of 20 hours and no new oil was added into the crankcase. In this particular test, fuel dilution increased with time continuously: fuel content increased almost linearly in the early stage and it slowed down after 40% of the test duration. In the rest of the test (total of 1000 hrs), thanks to the new oil being added into the crankcase to compensate the oil taken out and oil consumption, the maximum level of fuel dilution was $\leq 9.5\%$, similarly to that shown in Figure 10.