

Impact of Fuel Injection on Dilution of Engine Crankcase Oil for Turbocharged Gasoline Direct-Injection Engines

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ABSTRACT

Turbocharged gasoline direct injection (TGDI) engines often have a flat torque curve with the maximum torque covering a wide range of engine speeds. Increasing the high-speed-end torque for a TGDI engine provides better acceleration performance to the vehicle powered by the engine. However, it also requires more fuel deliveries and thus longer injection durations at high engine speeds, for which the multiple fuel injections per cycle may not be possible. In this study, results are reported of an experimental investigation of impact of fuel injection on dilution of the crankcase oil for a highly-boosted TGDI engine. It was found in the tests that the high-speed-end torque for the TGDI engine had a significant influence on fuel dilution: longer injection durations resulted in impingement of large liquid fuel drops on the piston top, leading to a considerable level of fuel dilution. Test results indicated that the higher the torque at the rated-power, the greater the level of fuel dilution. In a cyclic-load engine test simulating the customer drives of a target vehicle powered by the engine, the maximum level for fuel dilution was found to be up to 9%, causing significant drop in the oil viscosity. The causes for fuel dilution and impacts of it on the oil consumption and formation of carbon deposits on the piston ring area, and methods for mitigating impacts of fuel dilution are discussed in the paper.

CITATION: Hu, T., Teng, H., Luo, X., and Chen, B., "Impact of Fuel Injection on Dilution of Engine Crankcase Oil for Turbocharged Gasoline Direct-Injection Engines," *SAE Int. J. Engines* 8(3):2015, doi:10.4271/2015-01-0967.

INTRODUCTION

Turbocharged Gasoline Direct Injection (TGDI) has been recognized as a promising technology for improving fuel economy for gasoline engines [1, 2, 3, 4, 5]. One of the issues that have to be considered in developing TGDI engines is fuel dilution of the engine crankcase oil (for short, fuel dilution) [6, 7, 8, 9]. Fuel dilution is encountered in all direct-injection diesel and gasoline engines [6, 7, 8, 9, 10, 11, 12]. It is due commonly to cylinder wall wetting caused by one of following two mechanisms: condensation of the fuel vapor on the cylinder wall or dispersed liquid fuel drops in contact with the cylinder wall. In either case, some amount of liquid fuel on the cylinder wall passes down through the clearance between the piston and the cylinder and enters the engine crankcase, causing fuel dilution of the engine crankcase oil (for short, oil).

Fuel condensation typically occurs during the engine warmup phase when the engine is cold and thus some fuel vapor in the cylinder could condense on the cold cylinder wall before the combustion could start. Wall wetting can also be caused by impingement of some liquid fuel drops in the sprays onto the cylinder wall, either directly or indirectly due to interaction of the fuel spray with a strong in-cylinder charge motion, more often for side mounted fuel injectors as upward moving tumble flow turns spray towards the cylinder wall [1,3], or due to splashed fuel vapor and / or liquid fuel drops when the sprays impinging the piston top, which could happen to both side- and central-mounted injectors for TGDI engines, as shown in Figure 1.

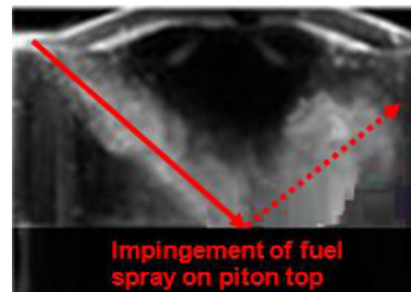


Figure 1. Impingement of fuel spray from a side-mounted gasoline injector on the piston top causing cylinder wall wetting.

In order to meet ever tightening emission regulations, most modern TGDI engines employ homogeneous mixture combustion [3, 4, 5]. For engines under this combustion mode, fuel is injected into the cylinder in the intake stroke or early of the compression stroke, the latter of which is due to long injection duration at high loads or last of the multiple injections. In reality, considerable wall wetting resulted from the piston impingement can also be encountered when the piston is at the bottom dead center (BDC), at which the cylinder wall temperature is close to the coolant temperature [13]. Since the oil film on the cylinder wall near BDC is thick [10], the fuel on the wall can be directly absorbed into the oil film.

The unburned fuel entering the crankcase oil causes following impacts: (1) it dilutes concentrations of the oil additives such as wear, corrosion and oxidation inhibitors, dispersant, detergent, etc. (2) it

may react with some oil additives and reduce their functionalities; (3) it makes the oil become thinner and more volatile, causing degradation in engine lubrication due to dropping in oil viscosity, and deterioration in oil consumption due to more oil loading in the blowby as a result of increased volatility of the oil; (4) it shortens the oil change interval. For highly boosted TGDI engines, increased engine oil loading in the blowby recirculation enhances low speed pre-ignition (LSPi) or super knock [14], which is encountered typically at high engine loads with speeds ≤ 2500 rpm [15,16]. Table 1 shows major oil analysis items and their levels calling for an oil service for gasoline engines. Among the items listed in Table 1, deteriorations in viscosity, total base number (TBN) and / or total acid number (TAN) are commonly related to fuel dilution. The method for measuring gasoline dilution has been specified in ASMT standard D3525-04 [17].

Table 1. Major oil analysis items and levels requiring oil change.

Oil analysis items	Levels requesting oil change
Viscosity @ 100 °C	One grade lower from new oil
Viscosity @ 40 °C (reference only)	25% change from new oil
Gasoline dilution	5%
Water	1.5%
Insolubles	1.5% max
Total base number (TBN)	50% change from new oil
Total acid number (TAN)	TBN – TAN < 0.5
Fe	70 ppm
Cu	40 ppm
Pb	40 ppm
Al	30 ppm
Cr	40 ppm
Si	30 ppm

This study reports the test results for fuel dilution in a highly-boosted TGDI engine and its impacts on the properties of the engine crankcase oil. The causes for fuel dilution and methods for reducing fuel dilution also will be discussed.

TGDI ENGINE AND CHARACTERISTICS OF ITS FUEL INJECTOR

Requested Fuel Delivery by the Engine

The engine under study is an inline 4-cylinder 1.5L TGDI engine with bore $B = 79$ mm and stroke $S = 76$ mm. The engine is highly boosted and fueled with RON93 gasoline. Figure 2 shows the engine torque and power normalized with the peak values. The engine has a flat maximum torque curve with the low-end speed = 1500 rpm and the high-end speed = 4600 rpm. The required fuel flows at 50, 75 and 100% loads are plotted in Figure 3, with the maximum fuel delivery occurring at the maximum torque at the high-speed end.

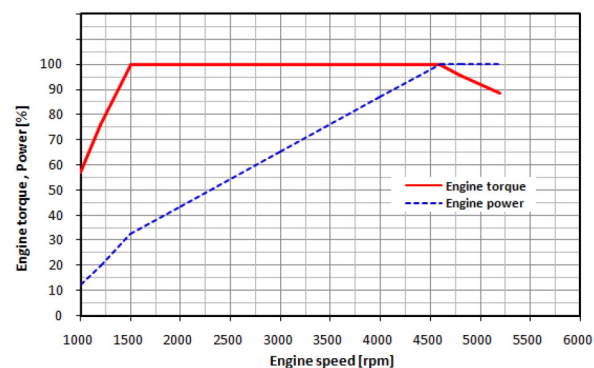


Figure 2. Torque curve for the TGDI engine under study.

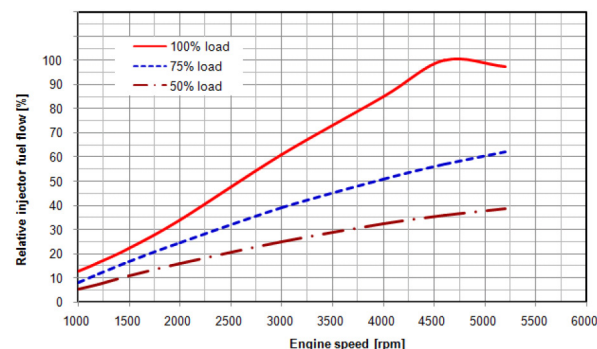


Figure 3. Relative injector flows at different loads and speeds.

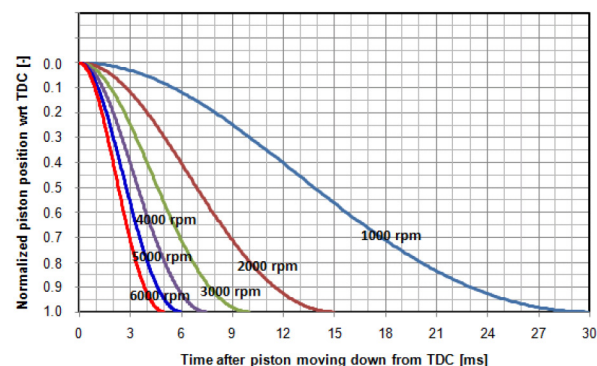


Figure 4. Normalized piston trajectory for different engine speeds..

Maximum allowed fuel injection duration varies with the engine speed and the selected injection window. Figure 4 shows the time for the piston to travel from TDC (top dead center) to BDC at different engine speeds. It is seen that the higher the engine speed, the less the time for the fuel injection. For a given fuel injection pressure, the delivery of the fuel injector is proportional to the injection duration. The higher the high-end speed for the maximum torque, the greater the challenge to the fuel injection system. Increasing the high-speed torque requests a more advanced fuel injection system for the engine, and it may also potentially lead to higher combustion smoke due to difficulty in forming homogeneous air-fuel mixture. The longer the crank angle for the injection duration, the shorter the crank angle left for the post-injection fuel evaporation and air-fuel mixing, and the greater the tendency for diffusion-flame combustion on the piston top surface to be encountered.

Influence of Engine Speed on Fuel Delivery

Start of fuel injection (SOI) is bounded with the earliest allowed injection timing in the intake stroke. To avoid the fuel impingement on the piston top, the fuel injection must start after the piston leaves TDC for a certain distance and gains a certain speed. End of fuel injection (EOI) is bounded with the time requested for fuel evaporation and mixing with the in-cylinder air in the compression stroke. Under given cylinder temperature and pressure, the injected fuel must fully vaporize and form a homogeneous combustible mixture before the ignition. The corresponding crank angle between these two bounds is a limit for the injection window. This limit for duration of fuel injection (DOI) also is a limit on the maximum high-end-speed torque.

Figure 5 illustrates relative timings for SOI, EOI, DOI, and injection window. Note that in cases of multiple injections in each engine cycle, SOI for the last fuel injection could be in the compression stroke. Figure 6 shows the piston speeds for 10, 20 and 45% of the intake stroke for different engine speeds. It is seen that for 5000 rpm, the piston speed is about 13 m/s at 10% of the intake stroke, greater than the maximum piston speeds (at about 45% piston stroke in this study) for engine speeds ≤ 3000 rpm. This indicates that the earliest allowed SOI increases with increasing the engine speed. For example, to avoid piston impingement, SOI at 10% piston stroke may be sufficient for 5000 rpm, but it may have to be at 45% piston stroke for 1000 rpm. As seen in Figures 4 and 6, at low engine speeds, although SOI may be around half of the piston stroke, it still has sufficient time for the fuel injection, even multiple injections, before the piston reaches BDC.

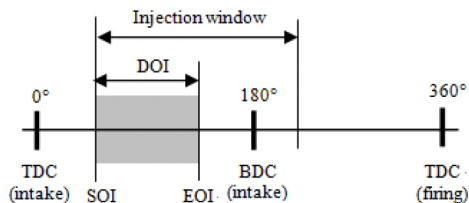


Figure 5. Illustration of relative timing for fuel injection.

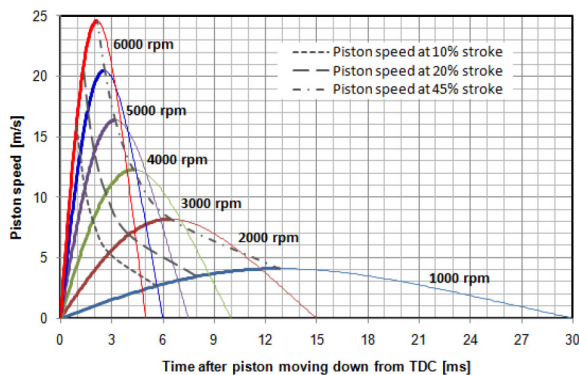


Figure 6. Piston speeds at 10, 20 and 45% intake stroke.

The engine angular speed in degrees of crank angle can be expressed as $\omega = 6 \times n$, where n is the engine speed in rpm and ω is in degCA/s. If $\Delta\phi_{\text{inj-window}}$ and $\Delta t_{\text{inj-duration,max}}$ denote the injection window and maximum allowed injection duration at given engine speed, then following relationships hold: $\omega = \Delta\phi_{\text{inj-window}} / \Delta t_{\text{inj-duration,max}}$.

$\Delta t_{\text{inj-duration,max}} = \Delta\phi_{\text{inj-window}} / (6 \times n)$. Figure 7 plots the available time for injection durations as a function of the engine speed for three arbitrarily selected injection windows. For a selected injection window, the maximum allowed injection duration is inversely proportional to the engine speed; thus, the challenge to the fuel injection is at the high-end-torque speed for the full load, where the maximum fuel demand per cycle is encountered.

In practice, the maximum injection duration for a given fuel system is commonly set at the maximum fuel demand; thus, in Figure 7, the allowed maximum fuel injection duration should be determined at the high-end-torque speed. The maximum fuel demand for this study is at 4600 rpm (Figure 3). If SOI is at 20 degCA (about 4% stroke) after intake TDC with corresponding piston speed (7.7 m/s) close to the maximum at 2000 rpm (8.2 m/s), then the allowed latest EOI is, respectively, at BDC for injection window = 160 degCA, at 10 degCA after BDC for injection window = 170 degCA, and at 20 degCA after BDC for injection window = 180 degCA. At the maximum fuel demand, the injection window = duration of injection. The corresponding injection duration in time is presented in Figure 8. Note that for a selected injection window for the fuel system, the maximum value for DOI applies to the entire speed range for the engine operations.

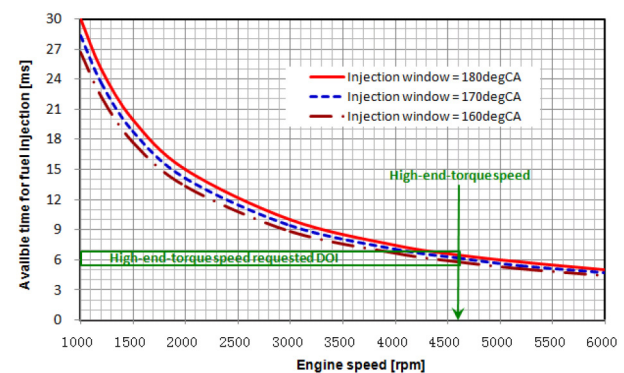


Figure 7. Available time for DOI at different engine speeds.

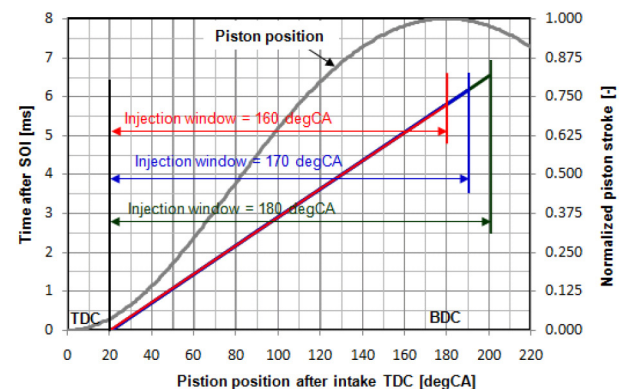


Figure 8. Illustration of influence of injection window on injection duration for 4600 rpm under full load condition.

Hydrodynamic Characteristics of Fuel Injector

The fuel injector for the TGDI engine under study is a 6-hole injector, as shown in Figure 9. The injector parameters are presented in Table 2. For the injection rate specified as the maximum injection