2015-01-0967
Published 04/14/2015

doi:10.4271/2015-01-0967
saeeng.saejournals.org

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

Tingjun Hu, Ho Teng, Xuwei Luo, and Bin Chen
Jiangling Motors Co., Ltd.

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 deports 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.

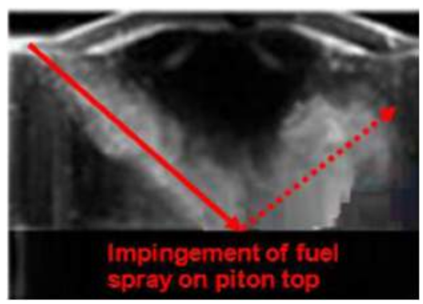

Figure 1. Impingement of fuel spray from a side-mounted gasoline injector on the piston top causing cylinder wall wetting.

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.

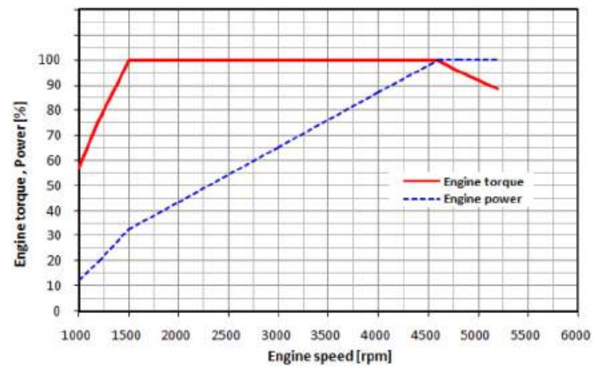

Figure 2. Torque curve for the TGDI engine under study.

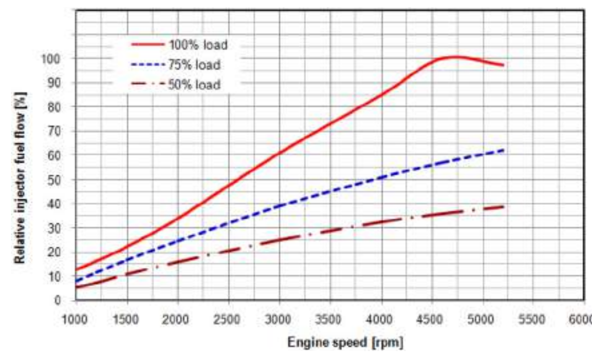

Figure 3. Relative injector flows at different loads and speeds.

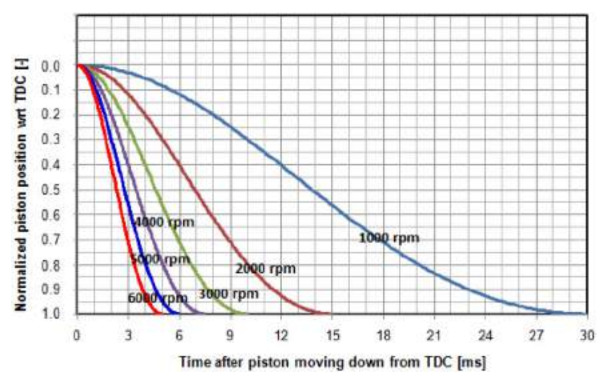

Figure 4. Normalized piston trajectory for different engine speeds..

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.

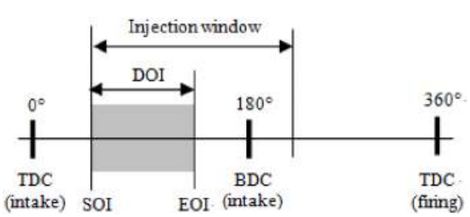

Figure 5. Illustration of relative timing for fuel injection.

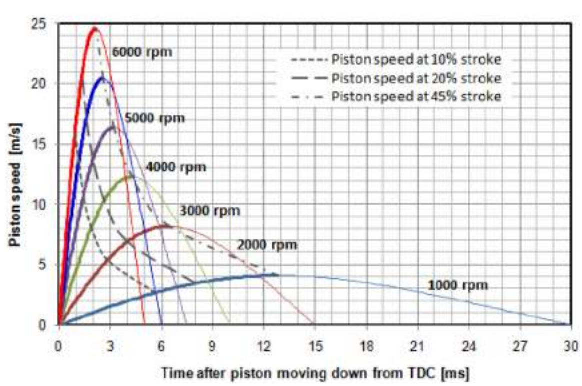

Figure 6. Piston speeds at 10, 20 and 45% intake stroke.

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 $\Delta\varphi_{inj\text{-}winidow}$ and $\Delta t_{inj\text{-}duration,max}$ denote the injection window and maximum allowed injection duration at given engine speed, then following relationships hold: $\omega = \Delta\varphi_{inj\text{-}winidow} / \Delta t_{inj\text{-}duration,max}$; $\Delta t_{inj\text{-}duration,max} = \Delta_{\varphi inj\text{-}winidow} / (6\times n)$. 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.

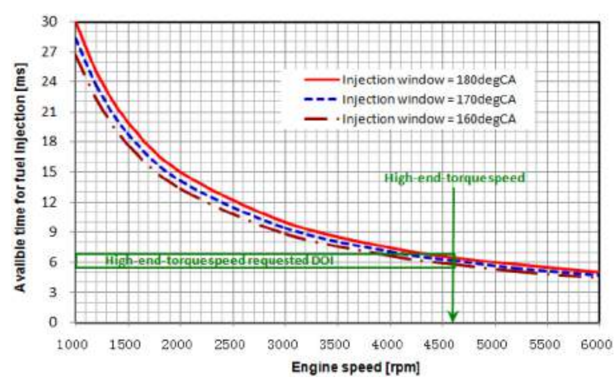

Figure 7. Available time for DOI at different engine speeds.

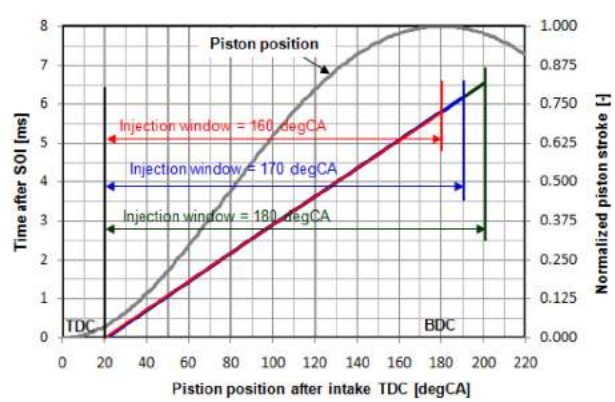

Figure 8. Illustration of influence of injection window on injection duration for 4600 rpm under full load condition.

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, if the maximum injection window is limited to 180 degCA, then the maximum fuel delivery will be 12 g/s for 4600 rpm, which is sufficient to meet the fuel demand for the engine under study. The injector is side mounted and its spray targets on the piston top are illustrated in Figure 10. Visualization of patterns for the fuel sprays from the nozzle at 100-bar injection pressure in the test facility is presented in Figure 11.

Figure 9. TGDI injector with a 6-hole nozzle.

Table 2. Injector parameters.

Fuel injector	6-hole gasoline injector
Injector mounting	Side mounted
Max injection pressure	150 bar
Injection rate at 150 bar	12 g/s
Fuel	RON93 gasoline

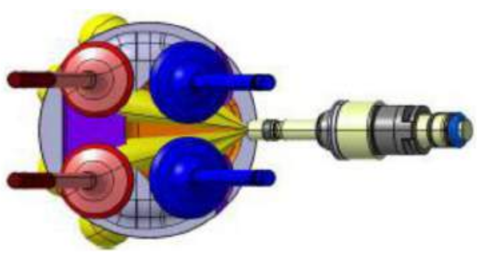

Figure 10. Side mounted fuel injector and the spray targets.

Hydraulic characteristics of the fuel injector were tested on an injection bench with test conditions and procedures following the SAE standard J2715 for the GDI spray characterization [18]. Figure 12 presents the measured spray tip penetrations at 50, 100 and 150 bar injection pressures. Corresponding Sauter mean diameters (SMD) for the fuel spray and drop diameters corresponding to 90% of the fuel mass distribution DV90 [18] are shown in Figure 13. The drop sizes were measured at 50mm for 50 bar and 75 mm for 100 and 150 bar from the injector. Predicted spray lengths and fuel drop diameters for injection pressures up to 200 bar employing the models reported by Teng [19] are also presented as a comparison in Figures 12 and 13. Since measurements did not cover the early stage of the spray development, where the spray length is a linear function of the injection time [20,21], the linear section for the spray lengths shown in Figure 12 was estimated without test data supports. It is seen that the predictions reasonably represent the measured spray lengths and fuel drop sizes.

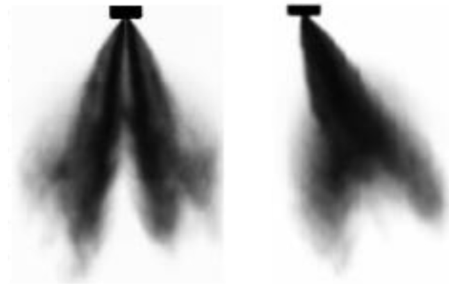

Figure 11. Patterns of fuel sprays at injection pressure = 100 bar viewed from two different directions of the test facility.

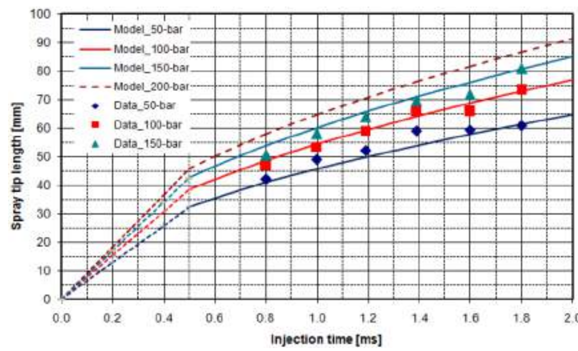

Figure 12. Measured and predicted spray lengths.

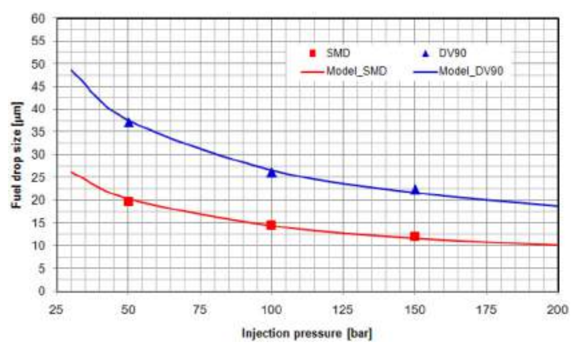

Figure 13. Measured and predicted SMD and DV90.

According to Figure 12, if the injection duration is longer than 2 ms, the tips of the fuel sprays would reach the piston top. The measured sizes for fuel drops were at the room temperature, at which the fuel drop evaporation rate is lower than that in the engine cylinder where the cylinder charge temperature is much higher due to being subject to the strong wall heating as well as mixing with the hot residual gas. Vaporization of the fuel sprays in the engine cylinder is faster than that in the test facility because fuel drops at the same spray length are smaller. As long as the distance between the fuel injector and piston is greater than 75 mm, the mean sizes for fuel drops in the sprays should be smaller than those shown in Figure 13. Figure 14 shows the predictions to the accumulated fuel mass function F and the drop size distribution expressed in the dF/dD-D relationship (the model for which can be found in reference [19]) for injection pressure = 150 bar, where D is the drop size. It is seen that last 10% fuel mass with drop sizes in a range from 22 to 40 µm. It needs to be pointed out that this group of large drops could contribute considerably to fuel dilution if they hit the piston top and are splashed onto the cylinder wall, because their temperatures are lower and thus are more difficult to vaporize than the rest fuel drops [22].

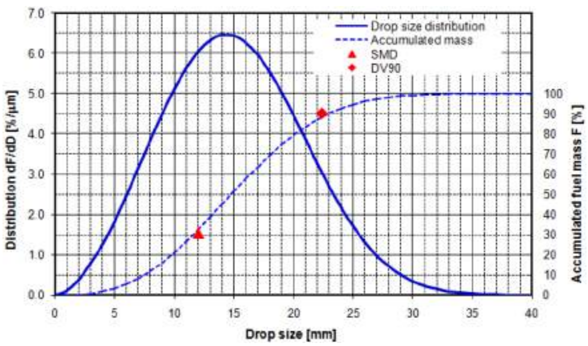

Figure 14. Predicted drop size distribution and accumulated fuel mass for 150-bar injection pressure and measured SMD and DV90.

PHENOMENA OF FUEL DILUTION

The crankcase oil for the TGDI engine is SAE 5W30 synthetic oil. Selected physical properties given in the oil specification are presented in Table 3. Characteristics of Chinese RON93 gasoline used in the tests are presented Table 4.

Table 3. Selected properties of engine crankcase oil.

Oil grade	5W30
Density @ 15 °C [kg/L]	0.852
Viscosity @ 100 °C [cSt]	11.7
Viscosity @ 40 °C [cSt]	67.5
Total base number TBN [mg-KOH/g]	10
Flash point [°C]	229
Noack volatility @ 250 °C [wt%]	< 13

Table 4. Characteritics of RON93 gasoline used in the tests.

Items	Values
Research Octane number (RON)	93.4
Motor Octane number (MON)	84.7
Density @ 20C [g/cc]	0.751
10% distillation temperature [°C]	58.6
50% distillation temperature [°C]	103.1
90% distillation temperature [°C]	167.5
FBP [C]	203
Residual [vol%]	1.2
Sulfur [wt%]	0.0031
Benzene [vol%]	0.4
Aromatics [vol%]	33.9
Olefins [vol%]	16.7
Pb [g/L]	<0.0025
Mn [g/L]	0.0051
Fe [g/L]	<0.0020

Temperatures of the crankcase oil at full load under different engine cooling conditions are shown in Figure 15, where the speed range from 5200 to 6000 rpm is only for a short duration of engine operations. The highest oil temperature is 135 °C, reached at 6000 rpm under the maximum engine coolant temperature 110 °C. Viscosities for the crankcase oil in the temperature range from 20 to 140 °C are shown in Figure 16. Viscosities for gasoline in the same temperature range are also plotted in Figure 16 as a comparison.

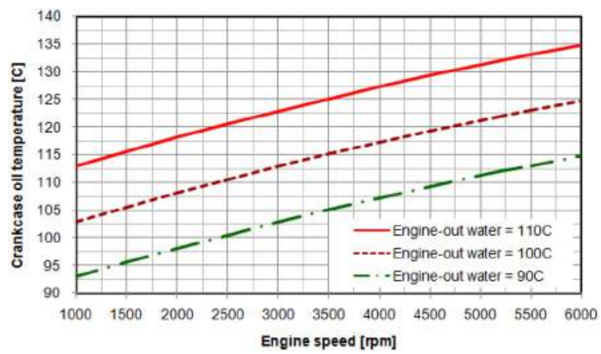

Figure 15. Engine crankcase oil temperatures at different speeds.

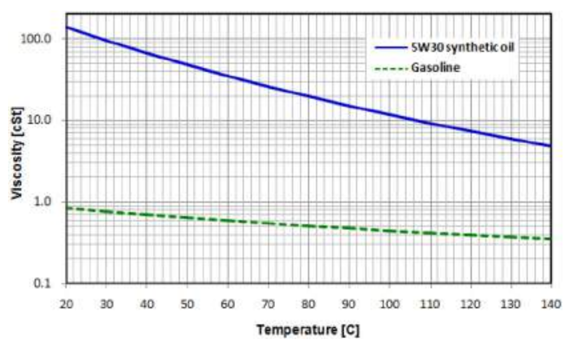

Figure 16. Viscosities of 5W30 synthetic oil and gasoline at different temperatures.

Fuel dilution was noticed in various engine durability tests. Figure 17 shows the dilution of the crankcase oil in a cyclic load test. The cycle profile was composed by a cold start phase (0.75 hr), an engine cooling phase with engine stopped (1.5 hr, not expressed in scale in Figure 17), and a cyclic-load test phase (1 hr). In the test, both the oil and coolant temperatures were maintained at 95 °C with external conditioning systems. Figure 17 covers the result for the first 40 cycles, during which the oil was sampled in about every 20 hours and no new oil was added into the crankcase. In this particular test, fuel dilution increased with time continuously: the content of fuel in the oil increased almost linearly in the early stage and it slowed down after 40% of the test duration, with the maximum fuel dilution being slightly greater than 9%. In the rest of the test (total of 1000 hrs), because of the new oil added into the crankcase to compensate the oil taken out for analysis, the maximum level of fuel dilution was similar to that shown in Figure 17. The distillation curve for Chinese RON93 gasoline used in the test is presented in Figure 18. The distillation curve suggests that for oil temperatures < 95 °C, 50% of the fuel representing the high-end or heavy components entered the crankcase oil may stay in the oil, as shown in the shaded area in Figure 18.

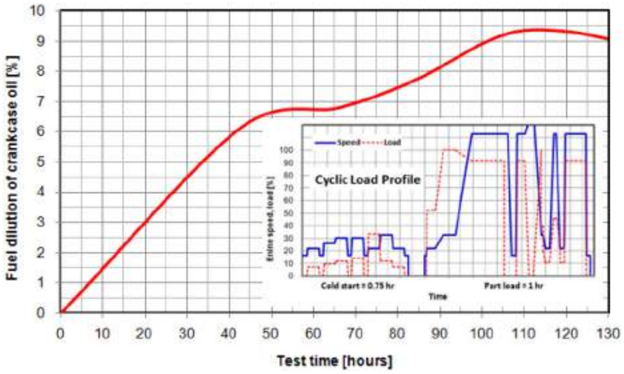

Figure 17. Fuel dilution of the crankcase oil in a cyclic load test with Crankcase oil temperature = 95 C.

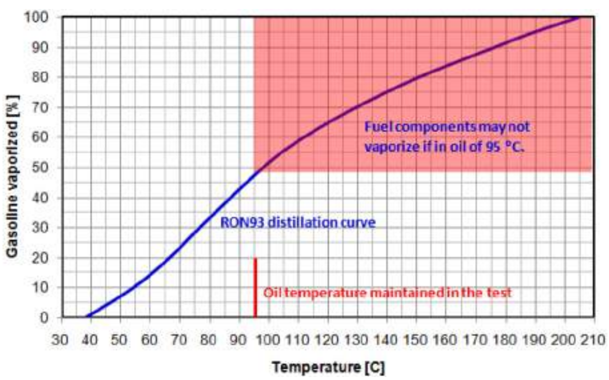

Figure 18. Engine crankcase oil temperatures at different speeds.

The load profile (shown in Figure 17) for the cyclic-load test has a large portion of high-speed / high-load engine operations, under which the impingement of fuel sprays on the piston top is unavoidable according to the analysis in the previous section. The piston inspection after the engine test indicated that the impingement by liquid fuel drops on the piston top was indeed encountered, with the spray impingement marks on the piston top as the evidence (shown in Figure 19). The impingement marks were caused by large fuel drops from only certain fuel sprays under high-speed / high-load operations. Considerable carbon deposits were observed on the top land and second land as well as on the oil ring of the piston, as shown in Figure 20. The carbon deposits were soft and different from those of the oil coking type and they could be removed easily. This may suggest that the oil temperatures in the ring area might not reach that for oil coking. Because the clearances bounded by the cylinder wall and the ring lands of the piston is the path for the fuel to flow into the crankcase, the level of fuel dilution for the oil in this space should be much greater than that in the crankcase. Thus, deterioration in the properties of the local oil due to heavy fuel dilution may be responsible for the soft carbon deposits in the ring area because, apparently, the deposit-control additives in the oil (e.g., detergent and dispersant) did not function well [23]. No abnormal wear was noticed for any of the engine bearings in the engine inspection after completing the durability test. In this particular test, the oil consumption was < 0.06% of the fuel consumption and the soot level was low with the maximum filter smoke number FSN_{max} < 0.1. This may suggest that the spray impingement on the piston top and fuel dilution of the oil may contribute considerably to the carbon deposits observed on the piston top and at least on the top land of the piston.

Figure 19. Spray impingement marks on the piston top.

Figure 20. Soft carbon deposit buildup on lands of the ring grooves and on the oil ring due to deterioration in oil properties.

ANALYSIS ON FUEL DILUTION OF CRANKCASE OIL

Experimental Evaluation of Fuel Dilution

Figure 21 shows three representative load points selected for characterizing fuel dilution of the crankcase oil: 1500 and 5200 rpm at full load with normal engine cooling, and 2500 rpm / 60% load with a low coolant temperature simulating the warmup phase for the engine operation. For the low speed (1500 and 2500 rpm) load points, as indicated in Figure 3, the fuel demands are much less and the available times for fuel injection are much longer than those at the high-end peak torque; thus, triple short fuel injections were applied to these load points to minimize the spray impingement on the piston top. Hence, for these low speed load points, potential fuel dilution was due largely to cylinder wetting by the fuel sprays. For the 5200 rpm load point, because of the high fuel demand and short available time for fuel injection, only a single injection is possible. In this case, the cylinder wall wetting may be due primarily to the contribution of the splashed fuel drops as a result of the impingement of liquid fuel drops on the piston top. In the tests for the full load points, the engine coolant temperature was set at 90 °C. In the part-load engine test, the engine coolant temperature was controlled at 55 °C simulating the potential cylinder wall wetting due to fuel vapor condensation during the engine warmup. Corresponding oil temperatures for the three load points are shown in Figure 22.

To estimate the evaporation of the fuel entered oil, the fuel distillation curve is also plotted in Figure 22.

For each of the full load points selected, the test was lasted two hours after the engine was thermally stabilized. For the part load point the test duration was only one hour. In the tests, the engine was operated with a stoichiometric mixture only for the 1500 rpm load point and the other two load points with enriched mixture (excess air-fuel ratio $\lambda = 0.8$). In each test, the oil was sampled at the beginning of the test after the engine was thermally stabilized and at the end of the test after the test duration was reached.

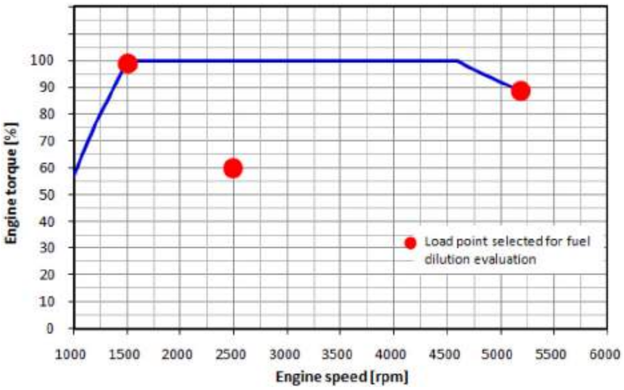

Figure 21. Load points selected for fuel dilution evaluation.

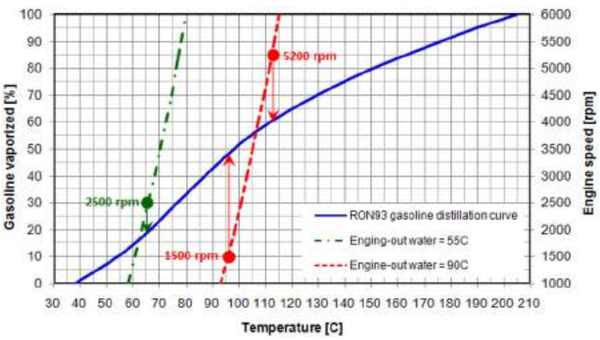

Figure 22. Load points selected for fuel dilution evaluation.

The fuel content in the oil was conducted following the test conditions and procedures specified in ASTM Standard D3525-04 [17], and oil viscosities were evaluated at 100 °C. The difference between the fuel contents in the oil samples taken at the beginning and end of the tests was taken as the fuel dilution. The results of the tests are shown in Figure 23. The levels of fuel dilution are reasonably close to those detected in the cyclic-load engine test shown in Figure 17. It is seen in Figure 23 that fuel dilution due to cylinder wall wetting caused by condensation of the fuel vapor at 2500 rpm / part-load when the engine was cold (corresponding oil temperature = 65 °C) and that caused by spray impingement on the piston top at 5200 rpm / full load when the engine was warm (corresponding oil temperature = 112 °C) are on the same order. However, because the oil was more viscous when the engine was cold (as shown in Figure 16), the impact of fuel dilution on the oil viscosity was much less than when the engine was warm: the oil viscosity at 65 °C is two times greater than that at 112 °C.

Items	Fuel Dilution Test Results						Acceptable level
	1500rpm / WOT		2500rpm / 12bar		5200rpm / WOT		
	0 hr	2 hr	0 hr	1 hr	0 hr	2 hr	
Viscosity @ 100 °C	10.37	9.87	10.57	8.21	10.73	8.63	≥ 8.2
TBN [mg-KOH/g]	9.3	9.1	9.3	8.9	9.4	8.8	> 5.0
TAN [mg-KOH/g]	2.9	2.3	2.6	2.2	2.3	2.2	< 4.0
Water [ppm]	174	59	110	179	282	56	< 2000
Propane insoluable[%]	< 0.1	< 0.1	< 0.1	< 0.1	< 0.1	< 0.1	< 1.5%
Gasoline [%]	2.41	5.15	2.22	8.78	1.62	8.59	
Net fuel dilution [%]	2.74		6.56		6.97		
Fe [ppm]	2	2	2	2	3	3	70
Pb [ppm]	<1	<1	<1	<1	<1	<1	40
Cu [ppm]	<1	<1	<1	<1	<1	<1	40
Cr [ppm]	<1	<1	<1	<1	<1	<1	40
Mo [ppm]	<1	<1	<1	<1	<1	<1	40
Al [ppm]	5	4	4	5	5	4	30
Si [ppm]	5	2	4	3	5	4	30

Figure 23. Load points for fuel dilution evaluation and test results.

The results suggest that, for the TGDI engine under study, the control of fuel dilution should be focused on the engine operations with high demands for fuel injections.

Impact of Fuel Dilution on Properties of Crankcase Oil

Figures 24 plots measured viscosities at 100 °C as a function of the gasoline content in oil. Oil viscosity degradation with the new oil as a reference is also presented. It is seen that 9% gasoline in the oil causes a decrease in oil viscosity by 30%. Figure 25 compares viscosities for 5W30 and 5W20 synthetic oils based on the oil specifications from the same oil supplier. In the temperature range from 20 to 140 °C, the ratio of viscosity for 5W20 to that for 5W30 is 0.61 at 20 °C and increasing to 0.71 at 140 °C. At 100 °C, viscosity for 5W20 is 68% that of 5W30, indicating that 9% fuel dilution is at a critical condition that may call for an oil service according to the criteria presented in Table 1. Figure 26 plots the total base number TBN and the total acid number TAN as a function of fuel dilution for the data presented in Figure 23. Values for TBN and TAN are less than the critical values given in Table 1.

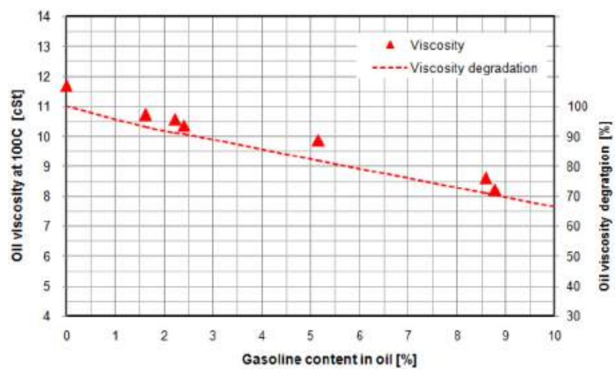

Figure 24. Degradation of oil viscosity with fuel dilution.

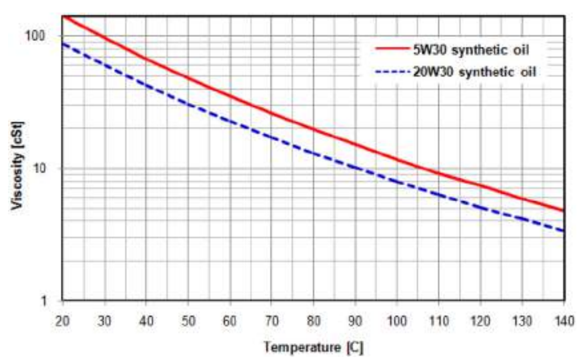

Figure 25. Viscosities of SAE 5W30 and 5W20 synthetic oils.

Figure 27 presents measured viscosities 100 °C and flash points of the diluted oil for other high-speed / full load points under test conditions same as that for the 5200-rpm load point. It is seen that fuel dilution also causes a significant drop in the flash point, which is more sensitive to fuel dilution than viscosity. Flash point is the temperature at which oil gives off vapors that can be ignited with a flame held over the oil. Flash point may be used as a measure for the oil volatility. Decrease in flash point by fuel dilution increases the oil vaporization loss at high temperatures as well as the oil burning-off on the hot piston surfaces in contact with the oil. This may give some explanation to the phenomenon observed on the piston in the ring area shown in Figure 20. With this link, it may be concluded that fuel dilution might be a major contributor to formation of the soft carbon deposits on the piston surfaces in the ring area.

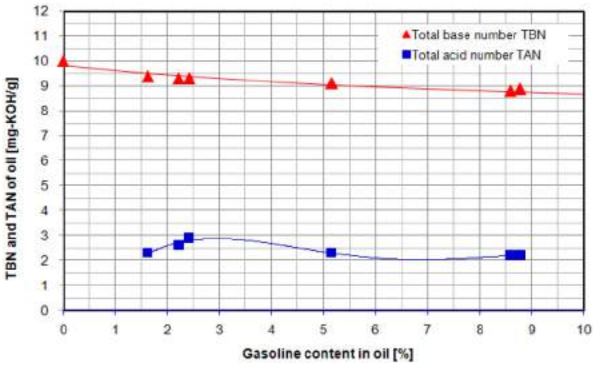

Figure 26. Degradations of TBN and TAN of oil with fuel dilution.

Besides viscosity, flash point of the oil is another indicator for oil deterioration [24]. These two physical properties of oil are inter-related. Figure 28 plots the flash point against the viscosity for the data presented in Figure 27. The relationship shows that oil thinning increases volatility of the oil. Oil with a low flash point would lead to high oil consumption as it enhances oil loading in the blowby recirculation and vaporization of oil on the cylinder wall. Thus, in evaluating the impact of fuel dilution, a criterion should be set for the flash point of the diluted oil to limit the maximum level for fuel dilution on the ground of oil consumption and carbon deposit formation on the piston surface. Note that the flash point of the diluted oil also has a strong influence on the ignitability of the oil particles entering the engine cylinder, and thus it affects the low-speed pre-ignition (LSPI) when the engine is operated at low speeds and high loads, which is another important issue that has to be faced in developing TGDI engines.

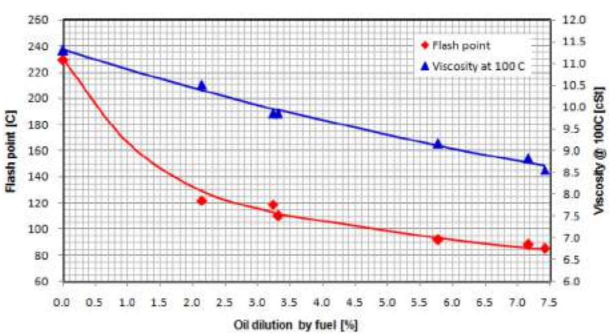

Figure 27. Measured viscosities and flash points for diluted oil.

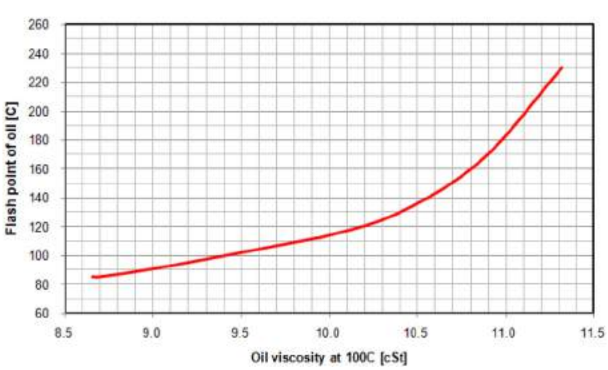

Figure 28. Relationship of flash point and viscosity for diluted oil.

INFLUENCE OF FUEL INJECTION ON DILUTION OF CRANKCASE OIL

Influence of Fuel Evaporation on Fuel Dilution

As aforementioned, the control of fuel dilution should be focused on control of the cylinder wall wetting resulted from the spray impingement on the piston top. The impact of the spray impingement on the cylinder wall wetting can be reduced by increasing the evaporation rates for the fuel drops in the sprays. According to the dependence of SMD on the injection pressure shown in Figure 13, increasing the fuel injection pressure will lead to smaller sizes for fuel drops in the sprays. Employing a method reported in an SAE paper of Teng [19], evaporation times were analyzed for fuel drops with different initial sizes in the engine cylinder with the evaporation time counting from SOI. It is seen in Figure 29 that the evaporation time for the 10 μm fuel drop is only half that for the 14 μm. This suggests that increasing the injection pressure can reduce fuel dilution for the same amount of the fuel delivered into the engine because reducing sizes of the drops in the fuel spray leads to a lower tendency for fuel drops to wet the cylinder wall and impinge the piston top.

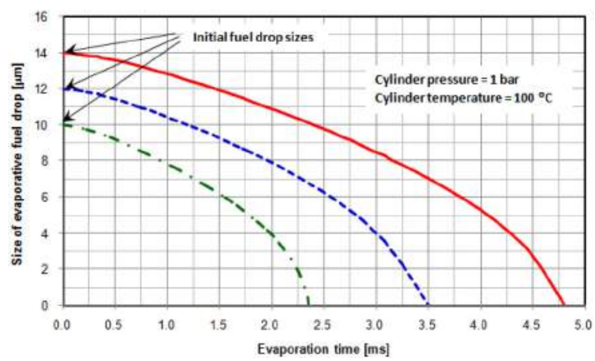

Figure 29. Evaporation of fuel drops with different sizes.

Fuel properties also have an influence on fuel dilution. Figure 30 compares the distillation curves for RON93 gasoline with that for RON97 gasoline. Overall, RON97 gasoline is more evaporative than RON93 gasoline. RON97 gasoline has higher anti-knock index and thus its optimized spark timings are more close to the maximum brake torque spark timing (MBT) under the same engine load condition, resulting in better fuel economy for RON97 gasoline and hence a less fuel demand in each cycle. Figure 31 compares the fuel demands at full load for the engine fueled with RON93 gasoline and RON97 gasoline. As a result of less fuel injection and thus shorter injection duration for RON97, fuel dilution at 5200 rpm / full load was 1.7%, only about 1/4 that for RON93 gasoline. Figure 32 shows the piston top and the ring sealing area for the engine fueled with RON97 gasoline after 100-hr operations at the rated engine power. No spray impingement marks on the piston top were noticed and limited soft carbon deposits could be observed below the top land of the piston, indicating that fuel properties indeed have a significant influence on fuel dilution and related phenomena. This supports the argument that the impact of spray impingement on the piston top varies with sizes of fuel drops in the spray; the larger the sizes of fuel drops hitting the piston top, the greater the level of fuel dilution.

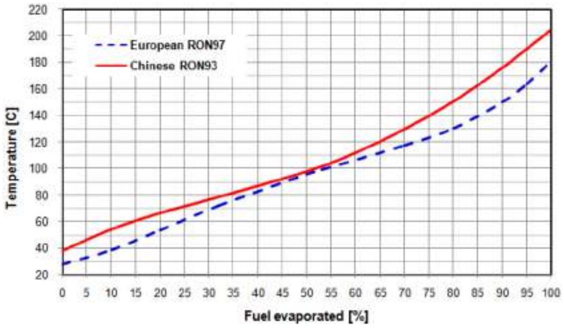

Figure 30. Distillation curves for RON93 and RON97 gasoline.

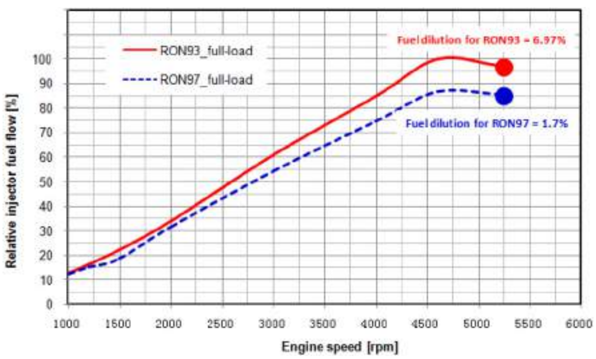

Figure 31. Comparison of fuel dilution at the rated engine power for the engine fueled with RON93 and RON97 gasoline.

Figure 32. Piston top and ring area for the engine fueled with RON97 gasoline.

Influence of Engine Torque Curve on Fuel Dilution

For a given fuel and an injection pressure, the fuel delivery is governed by the injection duration. Increasing the high-speed-end torque for a TGDI engine provides to better acceleration performance to the vehicle powered by the engine. However, increasing the high-speed-end torque also leads to longer injection durations to deliver the fuel requested and higher levels of fuel dilution, resulting in a shorter interval for oil change. Unless with significant down-speeding at the rated power, the high-end speed for the maximum engine torque may not cover that for the rated power for a highly-boosted TGDI engine in order to avoid a high-level fuel dilution. Figure 33 compares measured values of fuel dilution at the maximum engine power for the engine with three different torque curves. The test conditions were same as stated previously. These three torque curves have a same maximum engine power but achieved at different engine speeds. It is seen that the higher the torque at the rated-power, the higher the level of fuel dilution: measured the fuel dilution levels are, respectively, 5.0% for 5200 rpm / full load, 4.2% for 5500 rpm/full load and 2.5% for 6000 rpm / full load. In this comparative test for fuel dilution at the maximum engine power, fuel dilution for 5200 rpm / full load was lower than that in the previous test, which was due largely to differences in the initial engine conditions, the warmup process controls, and the uncertainties in the analyses on the fuel content in the diluted oil. As stated in ASTM Standard D3525-04, the results for the fuel content measurement are not highly repeatable [17].

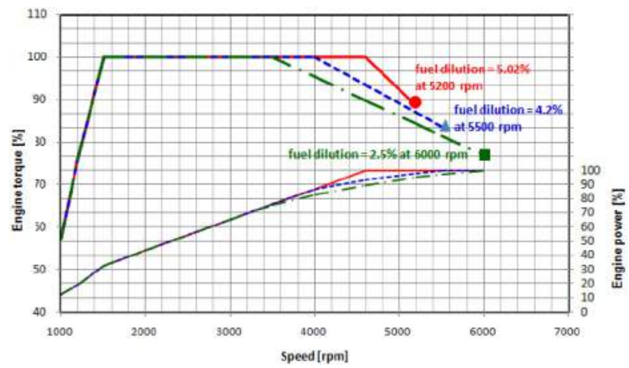

Figure 33. Comparison of fuel dilution for the same rated engine power but different engine torques.

SUMMARY

Impact of fuel injection was investigated experimentally on dilution of the crankcase oil for a highly-boosted TGDI engine lubricated with SAE 5W30 synthetic engine oil. It was found that the high-speed-end torque for the TGDI engine had a significant influence on fuel dilution: increasing the high-speed-end torque led to longer injection durations to deliver the fuel requested, which could result in an impingement of large fuel drops on the piston top and high levels of fuel dilution. Influence of the engine torque at the rated-power speed was investigated; the test results showed that the levels of fuel dilution were, respectively, 5.0% for 5200 rpm / full load, 4.2% for 5500 rpm / full load and 2.5% for 6000 rpm / full load, which indicates that the higher the torque at the rated-power, the higher the level of fuel dilution. Fuel dilution was also investigated in a cyclic-load engine test, for which the maximum level for fuel dilution reached 9%, causing 30% drop in the oil viscosity. Flash point of oil was found to be more sensitive to fuel dilution than oil viscosity. Decrease in flash point by fuel dilution increased the vaporization loss of oil at high temperatures, leading to higher oil consumption and resulting in formation of soft carbon deposits on the piston surfaces in the ring sealing area. For the same fuel delivery, improving the fuel evaporation, e.g., using a more evaporative fuel or decreasing the sizes of fuel drops in sprays by increasing injection pressures, can mitigate the impact of fuel injection on the crankcase oil dilution.

REFERENCES

1. Zhao, F.-Q. and Lai, M.-C., "Automotive spark-ignited direct-injection gasoline engines," *Progress in Energy and Combustion Science*, 25: 437-562, 1999.
2. Lang, O., Geiger, J., Habermann, K., and Wittler, M., "Boosting and Direct Injection -Synergies for Future Gasoline Engines," SAE Technical Paper 2005-01-1144, 2005, doi:10.4271/2005-01-1144.

3. Basshuysen, R.V., Gasoline Engine with Direct Injection: Processes, Systems, Development, Potential, Vieweg +Teubner, 1st ed., 2009.
4. Wurms, R., Jung, M., Adam, S., Dengler, S., et al., "Innovative Technologies in Current and Future TFSI Engines from Audi," 20th Aachen Colloquium Automobile and Engine Technology, 2011.
5. Königstedt, J., Assmann, M., Brinkmann, C., Eiser, A., et al., "The New 4L V8 TFSI Engines from Audi - the Logical Combination of Efficiency and Performance," 33th International Wiener Motor Symposium, 2012.
6. Sagawa, T., Fujimoto, H., and Nakamura, K., "Study of Fuel Dilution in Direct-Injection and Multipoint Injection Gasoline Engines," SAE Technical Paper 2002-01-1647, 2002, doi:10.4271/2002-01-1647.
7. Kume, T., Iwamoto, Y., Iida, K., Murakami, M. et al., "Combustion Control Technologies for Direct Injection SI Engine," SAE Technical Paper 960600, 1996, doi:10.4271/960600.
8. Shayler, P., Winborn, L., and Scarisbrick, A., "The Build-Up of Oil Dilution by Gasoline and the Influence of Vehicle Usage Pattern," SAE Technical Paper 2000-01-2838, 2000, doi:10.4271/2000-01-2838.
9. Alger, T., Huang, Y., Hall, M., and Matthews, R., "Liquid Film Evaporation Off the Piston of a Direct Injection Gasoline Engine," SAE Technical Paper 2001-01-1204, 2001, doi:10.4271/2001-01-1204.
10. Song, B.-H. and Yun-Ho Choi, Y.-H., "Investigation of Variations of Lubricating Oil Diluted by Post-Injected Fuel for the Regeneration of CDPF and Its Effects on Engine Wear," *Journal of Mechanical Science and Technology*, 22, 2526-2533, 2008.
11. Nanjundaswamy, H., Tatur, M., Tomazic, D., Wiartalla, A., et al., "A Novel Approach in Determining Oil Dilution Levelon a DPF Equipped Vehicle as a Result of Regeneration," DEER Conference, 2009.
12. Wattrus, M., "Fuel Property Effects on Oil Dilution in Diesel Engines," *SAE Int. J. Fuels Lubr.* 6(3):794-806, 2013, doi:10.4271/2013-01-2680.
13. Heywood, J.B., Internal Combustion Engine Fundamentals, McGraw-Hill Book Company, 1988.
14. Hirano, S., Yamashita, M., Fujimoto, K., and Kato, K., "Investigation of Engine Oil Effect on Abnormal Combustion in Turbocharged Direct Injection - Spark Ignition Engines (Part 2)," SAE Technical Paper 2013-01-2569, 2013, doi:10.4271/2013-01-2569.
15. Dahnz, C. and Spicher, U., "Irregular Combustion in Supercharged Spark Ignition Engines - Pre-Ignition and Other Phenomena," *Int. J. Engine Res.*, 11: 485-498, 2010, DOI:10.1243/14680874JER609.
16. Kalghatgi G.T. and Bradley, D., "Pre-ignition and 'Super-Knock' in Turbo-Charged Spark-Ignition Engine," *Int. J. Engine Res.*, 13(4): 399-414, 2012, DOI:10.1177/1468087411431890.
17. ASTM, "Standard Test Method for Gasoline Diluent in Used Gasoline Engine Oils by Gas Chromatography," ASTM Standard D 3525 - 04, November 2004.
18. SAE International Surface Recommended Practice, "Gasoline Fuel Injector Spray Measurement and Characterization," SAE Standard J2715, Iss. Mar. 2007.
19. Teng, H., "A Model-Based Analysis on Size Distribution and Rate of Evaporation for Fuel Drops in a Gasoline Spray in the Engine," SAE Technical Paper 2012-01-1264, 2012, doi:10.4271/2012-01-1264.
20. Hiroyasu, H., Kadota, T. and Aria, M., "Supplementary Comments: Fuel Spray Characterization in Diesel Engines," Combustion Modeling in Reciprocating Engines (Mattavi et al. ed), 369-408, Plenum Press, 1980.
21. Arai, M., Tabata, M., Hiroyasu, H., and Shimizu, M., "Disintegrating Process and Spray Characterization of Fuel Jet Injected by a Diesel Nozzle," SAE Technical Paper 840275, 1984, doi:10.4271/840275.
22. Lefevre, A.H., Atomization and Sprays, Hemisphere Publishing Corporation, 1989.
23. Diabya, M., Sabliera, M., Le Negrateb, A., Fassib, M., E., et al., "Understanding Carbonaceous Deposit Formation Resulting from Engine Oil Degradation," *Carbon*, 47, 355-366, 2009.
24. Ljuubas, D., Krpan, H. and Matanovic, I., "Influence of Engine Oils Dilution by Fuels on Their Viscosity, Flash Point and Fire Point," *NAFTA* 61(2): 73-79, 2006.

CONTACT INFORMATION

TingJun Hu, Deputy Director
Engine Research & Development Department
Jiangling Motors Co., Ltd
Phone: +86 791 8526-7387
thu9@ford.com

Ho Teng, Chief Engineer in Gasoline Engines
Engine Research & Development Department
Jiangling Motors Co., Ltd
Phone: +86 791 8526-7012
hteng1@jmc.com.cn

ACKNOWLEDGMENTS

The authors thank colleagues of the engine lubrication team at JMC for collecting the gasoline samples and Shanghai Oil Gas & Chemicals Testing Center of SGS for conducting the crankcase oil property measurements.