

Fuel Effects on Knock in a Highly Boosted Direct Injection Spark Ignition Engine

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ABSTRACT

Extensive tests have been carried out in a single-cylinder Direct Injection Spark Ignition (DISI) engine using up to fifteen different fuels at inlet pressure of up to 3.4 bar abs. to study fuel effects as well as inlet pressure effects on knock. In addition fuel effects on particulate emissions at part-throttle were measured. Fuel anti-knock quality does not correlate with MON and is best described by the Octane Index, $OI = RON - KS$ where $S = RON - MON$ is the sensitivity of the fuel and K is a constant depending on the engine pressure/temperature regime. The RON of the fuels considered was in the range between 95 and 105 and the sensitivity between 8 and 13. K is negative at all the conditions tested i.e. for a given RON, a higher sensitivity fuel has better anti-knock quality. K decreases with increasing intake pressure and more generally, decreases as T_{comp15} , the temperature of the unburned gas at a pressure of 15 bar decreases. If the temperature of the unburned gas at a given pressure is lower than that in the RON test conditions, K is negative. Knock becomes more likely with increasing intake pressure and increasing compression ratio with all else being equal. Higher anti-knock quality of the fuel enables higher load to be reached. However the gain in load with increasing OI decreases as OI increases - there are fast diminishing returns on increasing RON above 100 as long as the sensitivity is 10 or greater. Reducing the compression ratio enables greater boosting levels and hence a higher potential for down-sizing. In addition, compared to the pump gasoline (96.4 RON), a higher OI fuel can be run at a higher intake pressure, if such boosting levels can be provided by the hardware, so that the potential for down-sizing can be increased e.g. a 3.2 liter engine could be down-sized to 1.5 liters using a 2-stage turbo-charger. A vehicle simulation over an urban, road and highway conditions suggests that, with the pump fuel, this would lead to around 16% reduction in fuel consumption. This reduction could be around 19% with a fuel of RON of 99.6 and MON of 86.8 compared to a 95 RON fuel. The time taken for the first 10% burn is greater than the time taken for burning the next 80% of the charge. At cold idle conditions, the particulate emissions are higher than at full throttle conditions and they increase as the fuel aromatic content increases. Such emissions considerations as well as fuel effects on pre-ignition and the well-to-wheel implications need to be considered along with knock in designing optimum fuels for future SI engines.

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INTRODUCTION

The current focus in spark ignition (SI) engine development is to increase efficiency. The main approach to achieving this aim is through down-sizing the engine allied with turbo-charging. SI engines have to run essentially at stoichiometric mixture strengths and over a wide range of

loads and speeds. Fuel energy is directly proportional to the mass of air in the cylinder since the mixture strength is fixed so that the load is determined by the mass of air that is compressed in the engine. The air mass needs to be varied in order to vary the load at any given speed. In a naturally aspirated engine, the engine displacement is determined by the maximum load that needs to be achieved so that the air

mass needs to be reduced by the use of a throttle at low loads. The main inefficiency in SI engines arises from throttling losses. If the engine size is reduced, it can be run even at low loads at or near full throttle but will need to be turbo-charged to increase the trapped air mass in order to attain higher loads. Reducing the engine size also has additional benefits resulting from the reduced weight and volume of the engine. However, this approach is limited by knock, an abnormal combustion phenomenon initiated by the auto-ignition of the “end-gas” ahead of the advancing flame front. This causes a sharp rise in the heat release rate. The engine vibrates with dominant frequencies between 5 kHz and 10 kHz depending on the engine geometry and generates the sharp “pinking” sound characteristic of knock. When knock is severe, it can cause serious engine damage but even when it is not severe is easily perceptible through its characteristic noise and can be a major source of annoyance. Auto-ignition depends on the pressure and temperature history of the end-gas and also on the fuel auto-ignition quality which in turn depends on the fuel composition. Auto-ignition is determined by the chemical kinetics of the mixture and becomes more likely as the pressure and temperature in the end gas increase. Increasing the auto-ignition or anti-knock quality of the fuel would enable higher boost levels to be employed so that higher loads could be attained at a given speed and a greater degree of down-sizing could be achieved.

Fundamental studies such as measurements of ignition delay in shock tubes and rapid compression machines and modeling of chemical kinetics have helped greatly in understanding auto-ignition. Practical fuels are too complex for comprehensive chemical kinetic schemes to be developed for them. Chemical kinetic models for surrogate fuels which are blends of a few pure compounds but match the behavior of a practical fuel are being developed. The auto-ignition quality of practical fuels has been traditionally described by the research and motor octane numbers, RON and MON which are the most important gasoline properties. RON and MON are measured at two very different pressure/temperature conditions in a single-cylinder CFR engine running in strict accordance with ASTM procedures [1]. The manufacture of practical transport fuels is dominated by the requirement to meet the relevant RON and MON specifications. Given the central role of RON and MON in the fuel industry, auto-ignition in engines using practical fuels will need to be described and understood in terms of RON and MON of the fuel in the foreseeable future regardless of any advances in fundamental understanding of auto-ignition and the accuracy and the applicability of kinetic models. However the RON and MON scale is based on primary reference fuels (PRF), mixtures of iso-octane and n-heptane, whose chemistry is very different from that of other fuels and RON and MON do not describe the auto-ignition quality of a practical fuel away from the RON or MON test conditions.

The auto-ignition quality of practical fuels is better described by the Octane Index, a combination of RON and

MON when the pressure/temperature evolution in the unburned gas in the cylinder differs from that found in the RON or MON test [1,2,3,4]. In general if x is a measurement parameter which depends on the auto-ignition quality of the fuel in an engine or a vehicle test, it is possible to find a linear regression equation such that

$$x = aRON + bMON + c \quad (1)$$

x can be the Knock Limited Spark Advance (KLSA) in engine tests [2,4]; or it can be either an acceleration time or Knock Limited Power in vehicles equipped with knock-sensors [1,3,4]

The octane index, OI is then defined with RON and MON and the coefficients from the linear regression.

$$OI = \frac{a}{a+b} RON + \frac{b}{a+b} MON \quad (2)$$

Equation 1 can then be rewritten with the octane index.

$$x = c + (a + b)OI \quad (3)$$

We now introduce a constant K that is defined as:

$$K = \frac{b}{a+b} \quad (4)$$

The octane index can now be rewritten as

$$OI = (1 - K)RON + KMON = RON - KS \quad (5)$$

Here S is the sensitivity of the fuel, $(RON - MON)$. The Octane Index of a fuel is the octane number of the PRF that matches it for auto-ignition behavior in the particular test under consideration. K is a constant which depends on the pressure and temperature evolution in the unburned gas - it is not a primary property of the fuel. The higher the OI, the greater the resistance to auto-ignition and, in the context of SI engines, the better the anti-knock quality of a fuel at the particular condition. For a PRF, since $S = 0$, the OI is the same as its RON or MON.

At least three pieces of information are needed to characterize the auto-ignition quality of gasoline fuels - two measures such as RON and MON which describe the auto-ignition behavior at two very different pressure/temperature evolutions with time and another parameter such as K in OI, which describes the actual engine operating condition in relation to the specification test conditions. Engines can be run at different pressures and temperatures depending on the intake conditions and engine design. Figure 1 shows temperature in the unburned gas plotted against the pressure for the RON and MON tests based on the results from [5] and for a condition discussed in [2]. Modern SI engines are “beyond RON” because the temperature of the unburned mixture, at a given pressure, is even higher than in the RON

test. Alternatively, the pressure for a given temperature is higher in modern engines compared to the RON test. This is a consequence of the higher efficiency of modern engines. Engine designers have always tried to force in more air into the cylinder while minimizing the temperature in order to improve power density and efficiency. Indeed, all the approaches aimed at improving the efficiency of SI engines such as increasing the compression ratio, direct injection and turbo-charging aligned with down-sizing will push modern engines further “beyond RON” [1]. By definition, for the RON test condition, $K = 0$ and $OI = RON$ and for the MON test condition, $K = 1$ and $OI = MON$. The value of K is negative in modern engines so that for a given RON, a higher sensitivity (*lower* MON) fuel has higher OI and hence is more resistant to knock [1, 2, 3, 4, 6, 7, 8, 9, 10, 26]. Mittal and Heywood [11] have demonstrated that in the U.S., the K value has decreased from around 1 in the 1930s when the MON test was introduced, to zero in 2008 because of reduced end-gas temperatures and higher pressures. A similar decreasing trend for K in the U.K. car fleet has also been shown in [1]. This trend of decreasing K is consistent with the experimental observations of ignition delays from shock tube studies [e.g. 12,13] which show that if the pressure is increased for a given temperature, non-paraffinic fuels (those fuels with $S > 0$) become relatively more resistant to auto-ignition compared to PRF. An increase in pressure (as well as in temperature) causes a decrease in the ignition delay, τ_i - the mixture becomes more reactive. In general τ_i can be expressed as

$$\frac{\tau_i}{\tau_{i0}} = f\left(\frac{T}{T_0}\right)\left(\frac{P}{P_0}\right)^{-n} \quad (6)$$

Here, τ_{i0} is the ignition delay measured at a pressure and temperature of P_0 and T_0 respectively. The value of the pressure exponent, n , is around 1.6 for PRF but around 1 or lower for non-paraffinic fuels. It is then assumed that auto-ignition occurs when the integral of the reciprocal of τ_i with time, t , (Equation 4.1) reaches unity.

$$I = \int_0^t \frac{dt}{\tau_i(P,T)} = 1 \quad (7)$$

This approach was first suggested by Livengood and Wu [14] and is now commonly used in knock studies as well as HCCI studies.

If the inlet pressure is increased by turbo-charging at a fixed inlet temperature, ignition delay decreases and the Livengood-Wu integral, I , will reach unity earlier i.e. knock becomes more likely. Moreover, conditions will move further “beyond RON” in Figure 1 so that K will become negative and for a given RON, a *higher* sensitivity fuel will have higher OI i.e. better knock resistance. This has important implications for the specifications, which in turn determine the manufacture, of fuels for such engines. For instance,

current gasoline specifications in Europe as well as in the U.S. assume that higher MON contributes to higher anti-knock quality - in the U.S., fuel anti-knock quality is specified by $(RON+MON)/2$ while in Europe a *minimum* MON of 85 is specified. These specifications would have been appropriate when they were first set but engines have moved towards or “beyond” RON as they have become more efficient so that MON has become irrelevant [11] and is even detrimental to fuel anti-knock quality in modern European and Japanese cars [1,3,6]. This mismatch between fuel specifications and the actual requirement of engines can be expected to become more pronounced with down-sizing and turbo-charging. Hence it is very important to understand fuel effects on knock in turbo-charged engines.

This paper describes a systematic and extensive experimental study on the fuel effects on knock using a single-cylinder turbo-charged engine. Some previous studies [1, 6] on the octane appetites of modern vehicles have included vehicles equipped with turbo-charged engines. A study similar to the one described here but on a different engine has been published recently [26]. The aims of the current study were - a) to verify that, for a given RON, more sensitive fuels were more resistant to knock in boosted engines and b) to explore the potential for down-sizing using fuels of higher anti-knock quality. The highest IMEP that was obtained in these tests was 35.1 bar with an intake pressure of 3.4 bar abs. using one of the high RON, high sensitivity fuels. In addition some part-throttle results on fuel effects on particulate emissions are reported.

EXPERIMENTAL DETAIL

A single-cylinder DISI engine was tested on fuels of different RON and MON. Two compression ratios (CR) viz. 10 and 8.5 were considered and in each build the engine details are listed in Table 1. Reducing the compression ratio reduces the chances of knock and this would allow greater level of boosting and increase the potential for efficiency gains through down-sizing. Of course reducing the compression ratio also reduces the thermodynamic efficiency and the overall effect on efficiency will depend on the details of the engine and the vehicle and the operating regime. The maximum in-cylinder pressure was limited to 170 bar. The engine was tested at the stoichiometric condition, at full throttle, at two speeds, 1500 rpm and 4000 rpm. Engine pressure was measured using a water-cooled AVL pressure transducer. Intake pressure was varied, up to a maximum of 3.4 bar absolute, by using a compressed air system. The temperatures of the coolant, water and air could be controlled to within 1 K. On a given fuel, at each operating condition, a spark timing sweep was conducted and pressure and emissions data were recorded. Knock Limited Spark Advance (KLSA) was defined as described in the next section based on pressure measurements. In this paper, we primarily consider KLSA and Knock Limited IMEP (KLIMEP) at full throttle at different intake pressures. We also briefly discuss fuel effects

on emissions, particularly smoke, at cold idle conditions. The tests used a solenoid actuated multi-hole injector by Bosch (HDEV 5.1). The injector was limited to 150bar injection pressure. Hence at full load conditions the fuel pressure used was 150 bar though the high pressure fuel pump could deliver up to 200 bar injection pressure. The high pressure pump is supplied by a fuel pre-pressure of 5 bar. The pump is driven by an electric motor and has special sealing materials (for alternative fuels). All the fuels were tested at the same operating conditions. Intake air temperature was 25°C, engine-out coolant and oil temperatures were 90°C for the knock studies, 30°C for cold idle studies. These temperatures could be controlled to within 1°C.

The fuels used in the engine tests are listed in [Table 1](#) - in all there are 15 fuels. Amongst the parameters listed in [Table 1](#) are - RON, MON, composition, density and at stoichiometric conditions, the air fuel ratio (AFR) and the lower calorific value per kg of fuel. All the fuels have a wide boiling range and are made from refinery components; ethanol and ETBE are also used as blending components. Fuel PF is a European premium gasoline. The fuel properties were measured independently in three different laboratories and the values listed in [Table 2](#) are averages of these measurements. [Figure 2](#) shows that there was little correlation between sensitivity and RON for the fuel set used. Fuel design was guided by the principle that the correlation between S and RON should be minimized; otherwise it would not be possible to assess the separate effects of RON and S.

The first series of tests was aimed at establishing the octane appetite. Several fuels, in some cases, all fifteen, were tested at the same operating condition and KLSA and KLIMEP were established for each fuel. In the second series some of the higher octane quality fuels were tested at higher intake pressures in order to explore the potential for down-sizing using such fuels. In addition some part-throttle tests were conducted to understand fuel effects on particulate emissions which are getting to be of increasing concern in DISI engines.

RESULTS AND DISCUSSION

Combustion in SI engines is marked by significant cycle-to-cycle variations and cycles with faster flame development will have higher pressures and are more likely to knock [15]. In-cylinder pressure was collected for 300 cycles at each operating condition, with fixed CR, fuel, spark advance, engine speed and intake pressure. The pressure based measurements such as IMEP and CA50, the crank angle at which 50% of the heat is released, are averaged over 300 cycles. An example of a pressure signal filtered between 5 kHz and 10 kHz is shown in [Figure 3](#) for a knocking cycle. There are many different ways of defining knock intensity and threshold from such signals [e.g.16, 17] and several such approaches were used in this work during preliminary investigations. Some of these methods involved averaging over many cycles. The method chosen for defining knock

intensity did not change the ranking of fuels in these preliminary studies. The results described in this paper are based on the definition of knock intensity (KI) as the maximum peak-to-peak amplitude of a knock signal such as the one shown in [Figure 3](#) amongst the 300 cycles of data. [Figure 4](#) shows knock intensity plotted against the spark advance with the intake pressure at 2.2 bar abs. for different fuels at 1500 RPM. It can be seen that for a given fuel KI increases as ignition is advanced. We define the knock threshold at 2.25 bar. The spark advance at which the knock threshold is reached is the Knock Limited Spark Advance (KLSA). The anti-knock quality of the fuel at a given operating condition is defined by its KLSA - the larger the KLSA, the greater the resistance of the fuel to knock.

OCTANE APPETITE, THE K VALUE

[Appendix 1](#) lists KLSA for all the tests done in this study for different fuels and operating conditions. Also listed are the mean IMEP and maximum pressure with the ignition timing at KLSA as well as the RON and MON of the fuels and the intake pressure. In some cases viz with CR of 10, 1500 RPM and intake pressure of 2.2 bar abs., all fifteen fuels were tested. In fact two of the fuels, PF and T6 were tested twice at this condition. We estimate the K value using KLSA for x in [Equation 1](#) as illustrated in [Figures 5, 6 and 7](#). The case considered for this illustration used 11 fuels at 4000 RPM and an intake pressure of 2.6 bar abs. with CR = 10. KLSA correlates to some extent with RON ([Figure 5](#)) but does not correlate with MON ([Figure 6](#)). However KLSA correlates best with OI = RON +0.93S ([Figure 7](#)); the K value is - 0.93 in this case.

The description of knock solely based on OI cannot be perfect. For instance in these tests as also in the RON or MON tests, the effect of differences in burning rates between different fuels is not considered. Similarly, the effect of the differences in charge cooling brought about by the evaporation of different liquid fuels is also not considered; this is particularly relevant in DISI engines. Of course, charge cooling also leads to higher volumetric efficiency and hence higher pressure if the fuel is injected when the intake valve is still open and this might counteract the effect of reduced temperature. Such differences between fuels do exist but are small compared to the differences in auto-ignition chemistry which can be reasonably assumed to be the main source of differences in the knock behavior of practical fuels. Also the estimations of K value as described above are based on a large number of empirical observations and the fit between KLSA and OI is not perfect e.g. the R^2 value in [Figure 7](#) is not unity. Nevertheless some clear trends are observed and can guide the optimization of fuels for future boosted engines. The K value was estimated in this manner for all the cases where at least six fuels have been tested and these cases are listed in [Table 3](#) which also shows the operating conditions, the R^2 value of the fit, the number of experimental points used in the estimation. It can be seen that

K is negative i.e. for a given RON a fuel with greater sensitivity is more resistant to knock in all cases tested. Davies et al. [26] report K values ranging between -0.3 and -0.9 in their study with another turbo-charged engine.

Figure 8 shows that K will become more negative as intake pressure increases as also observed in [9]. In other engine experiments the value of K increases with increasing intake air temperature [9,10]. On the face of it, some of the observations in this study do not follow the trends observed before. For instance, in Figure 8, there is no evidence that as engine speed increases, with all else fixed, the value of K increases as has been observed in multiple previous engine tests [2, 4, 8, 9, 10]. The earlier results seemed to be reflecting the fact that as engine speed increases, heat transfer to the coolant decreases and in-cylinder temperatures increase. Also, K becomes more negative as the compression ratio is increased in previous engine tests [2, 4, 8, 9] while it is not so clear in the current tests. Combustion occurs by auto-ignition in HCCI (Homogeneous Charge Compression Ignition) engines which can be run over a much wider range of pressure/temperature variations than SI engines. The Octane Index approach has also been used to understand fuel effects in HCCI combustion [18,19,20,21,22]. The temperature in the unburned gas, when the pressure is 15 bar, T_{comp15} , was introduced in [18] to mark the position of the temperature/pressure polytrope such as those shown in Figure 1 for any given experiment; the value of 15 bar chosen for the pressure has no special significance. The K value in HCCI experiments could be shown to be primarily dependent on T_{comp15} [18,19,20,21,22]. The variations in engine design and operating conditions such as intake pressure, speed and load affect T_{comp15} directly. T_{comp15} was estimated for each condition in the current tests from the molar mass, estimated from the air consumption rate and residuals, and volume at the start of consumption and the polytropic constant for compression and is shown in Table 3.

Table 3 also lists results from previous studies of knock in engines where T_{comp15} is either known or can be estimated. Figure 9 shows that there is indeed a clear trend for K to decrease as T_{comp15} decreases in knocking engines. There is clear evidence, both from HCCI experiments [20] and knock experiments [8,9] that K decreases as the normalised air/fuel ratio, λ , increases i.e. as the mixture becomes leaner. An approximation for this trend for $\lambda > 2.5$ and $T_{comp15} < 825$ K in HCCI engines is given in [20] by

$$K = 0.00497T_{comp15} - 3.67 - 0.135\lambda \quad (8)$$

The dependence on λ is not very relevant to SI engines which mostly run at or around $\lambda = 1$. However, for the same T_{comp15} , the values of K reported in [19,20,21,22] for HCCI engines are much lower when compared to K from knocking engines listed in Table 3 because HCCI engines are run at very much leaner mixture strengths than knocking engines.

COMBUSTION PARAMETERS AT KNOCK LIMIT

We now discuss combustion rates by considering CA_x where CA_x is the crank angle position averaged over 300 cycles at which x% of the total heat release occurs. Figure 10 shows CA50 plotted against KLSA for CR= 10 at three different intake pressures and two speeds. It has to be remembered that each point shown in these figures is derived from a spark timing sweep and usually represents a different fuel. Figures 11 and 12 show respectively, (CA10-Spark Timing) and (CA90 - CA10) i.e. the crank angle window for early combustion and the main combustion, plotted against KLSA. Figure 10 shows that CA50 decreases for a given speed i.e. heat release occurs earlier as the spark timing is advanced i.e. as the spark is fired earlier in the cycle. At a given engine speed, there is no significant effect of intake pressure and hence engine pressure on CA50. However for the higher engine speed of 4000 RPM, the combustion phasing occurs later in crank angle terms because the combustion intervals (Figures 11 and 12) are longer. If there had been a significant variation in combustion rate for different fuels, for the same spark timing (KLSA) there would have been different values of CA10 or CA50 for different fuels and there would have been a lot of scatter in the trends. Figure 10 suggests that the differences in combustion rate between different fuels is not large though Figures 11 and 12 show more scatter. It is interesting to note that the time taken for the first 10% burn (Figure 11) is greater than the time taken for burning the next 80% of the charge (Figure 12). There could be opportunities for improving the combustion phasing and hence the efficiency and IMEP by increasing the initial burn rate, say, through a more powerful spark. Of course, this will also affect the pressure rise and hence knock. A fuel with faster initial burn (spark to CA10) but more resistance to knock would be a good solution for better indicated efficiency. As engine speed increases, turbulent burning velocity and hence combustion rate is expected to increase in step so that combustion intervals in crank angle terms such as those plotted in Figures 11 and 12 should not depend on engine speed. The absolute time for one crank angle interval would decrease by a factor of 2.667 on going from 1500 RPM to 4000 RPM. The ratio of the combustion interval for 4000 RPM to the combustion interval at 1500 RPM is around 1.35 in Figure 11 and around 1.25 in Figure 12. Thus in absolute time, combustion rate is faster by about a factor of 2 at 4000 RPM compared to 1500 RPM but this increase is not high enough to ensure that the combustion phasing is the same at the two speeds in crank angle terms. Figure 13 shows the peak pressure plotted against KLSA - as expected peak pressure increases as ignition advance increases. The peak pressure reached for a given KLSA is slightly lower for 4000 RPM because of the slower combustion in crank angle terms compared to 1500 RPM as discussed above.

Figures 14 and 15 consider the effect of changing the intake pressure on the knock-limited parameters for a given fuel - three fuels viz PF, T6 and T8 are considered at 1500 RPM since they have been tested over the widest intake pressure range. Figure 14 shows that as expected, for a given fuel KLSA decreases as intake pressure increases since knock becomes more likely. A higher KLSA can be reached for a given fuel as the compression ratio is reduced from 10 to 8.5. Also, Fuels T6 and T8, with RONs of 99.6 and 103 respectively allow higher KLSA to be reached for a given intake pressure compared to the Pump Fuel with a RON of 96.4. Figure 15 shows the knock limited indicated net efficiency plotted against intake pressure. For a given fuel, the efficiency at the knock limit decreases as intake pressure is increased beyond 1.8 bar abs. presumably because KLSA is reduced and combustion occurs later in the cycle (see Figure 10). Fuels T6 and T8 with higher RON and hence higher KLSA yield a higher efficiency at a given intake pressure. However, what really matters is the efficiency improvement over the entire operating range. Increased turbo-charging will improve the overall fuel efficiency and enabling boosting via higher fuel anti-knock quality will contribute to improved fuel consumption as discussed in the next section.

INCREASING THE POTENTIAL FOR DOWNSIZING THROUGH HIGHER FUEL ANTI-KNOCK QUALITY

Figure 16 shows knock limited IMEP plotted against OI for the results considered in Figure 5,6,7. Another example of KLIMEP vs OI is shown in Figure 17 for 1500 RPM, an inlet pressure of 2.2 bar abs. and 8.5 CR. The pump fuel and T6 are identified on both the figures. The gain in KLIMEP with increased OI is quite small for very high OI - there is not much to be gained in increasing the fuel anti-knock quality compared to the pump fuel beyond that of T6 for the two cases shown in Figures 16 and 17. In all there are nine operating points where Fuels PF and T6 were directly compared - see Appendix 1. At each operating condition we calculate the difference in KLSA between the two fuels and the ratio of KLIMEP with Fuel T6 to KLIMEP for Fuel PF. The averages and standard deviations of these values are shown in Table 4. Fuel T6 enables a higher KLSA to be reached and this enables about 4% higher KLIMEP to be achieved compared to the pump fuel PF. Also shown in Table 4 are the comparison of Fuel PF with Fuel B2 which is very similar to PF and Fuel B1 which has a slightly lower RON and higher sensitivity compared to PF and Fuel B3 which has a lower RON and higher sensitivity compared to PF. Table 4 shows that on average, as expected, Fuel B2 shows little difference from Fuel PF in both KLSA and KLIMEP while the less sensitive Fuel B1 has lower KLSA and around 4% lower KLIMEP. Fuel B3 can be expected to be worse (lower KLSA and KLIMEP) than Fuel PF at the conditions where the two fuels have been compared because of its lower OI but

the differences in KLSA as well as KLIMEP are much larger than expected from the difference in OI between PF and B3. We do not have a full explanation for this though it might be worth noting that Fuel B3 is not quite a full boiling range gasoline; it is very volatile with a final boiling point of 103°C - Table 2. Figure 18 shows KLIMEP plotted against intake pressure for fuels PF, T6 and T8 for 1500 RPM. The highest IMEP obtained with Fuel PF was 30.4 bar at an intake pressure 3.2 bar abs. and 8.5 CR. At the same operating condition we might have expected Fuel T6 to yield a KLIMEP of 31.6 bar - about 4% higher. However, Fuel T6 could be run even with an intake pressure of 3.4 bar abs. with 8.5 CR and attained a KLIMEP of 34.1 bar. Fuels T6 and T8 might have been able to be run at even higher intake pressures but this could not be tested because such pressures could not be reached with the hardware available.

DRIVE CYCLE SIMULATION

In order to assess the potential for fuel savings a drive cycle simulation was performed using GT-Drive with "high potential" advanced fuel formulations (namely Formulations T6 and T8) and baseline (95RON fuel). The baseline vehicle was Audi A6 3.2 L Naturally Aspirated (NA) FSI with 1750 Kg curb weight and 7-speed dual clutch transmission. Three different scenarios were investigated with optimized gear shift strategy for the drive cycle of choice:

- a). A 184 kW, 3.0588 liter naturally aspirated direct injection engine optimized for advanced fuel, increased compression ratio (CR=12.75) and downsizing
- b). A 93.5 kW/l, 1.97 liter turbocharged engine with direct injection with compression ratio optimized for advanced fuel and drive cycle
- c). A 120 kW/l, 1.53 liter 2-stage turbocharged engine with direct injection with compression ratio optimized for advanced fuel and drive cycle.

A combined Artemis cycle (Urban, Road and Motorway) was selected to calculate the combined effects of fuel formulation and engine upgrade (downsizing and compression ratio adjustments) for the three scenarios described above. The vehicle speed/time schedule for this cycle is shown in Figure 19. The total travel distance amounts to 51.7 km with a maximum speed of 150.4. The GT-Drive simulation results are presented in terms of fuel consumption (L/100km), Figure 20. The fuel consumption reduction due to both downsizing and fuel formulation (fuel/engine system effect) ranges between 10.7% and 19.5% depending on the baseline and projected hardware upgrade as shown in Figure 21.

Additionally, the incremental benefits due to fuel formulation were calculated as follows:

- i). Compared the 3.2L NA baseline with 95 RON gasoline, fuel formulation T6 resulted 2.67%, 2.99%, and 3.46% for the adapted NA version, the single staged turbo-