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# Significance of RON and MON to a modern DISI engine

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**Abstract:** The anti-knock quality of gasoline fuels is a significant contributing factor to the indicated thermal efficiency (ITE) of spark ignition (SI) engines. Historically, the anti-knock quality of gasoline is characterised by two parameters, research octane number (RON) and motor octane number (MON), which are measured in cooperative fuel research engines (CFR) using iso-octane and n-heptane as the primary reference fuels (PRFs). However, due to significant hardware, operating condition and fuelling differences between the CFR and the modern SI engines, the relevance of RON and MON to modern SI engines needs to be re-assessed. In this study, six fuels were designed with independent control over RON and MON. The other key fuel properties, such as the heat of vaporisation, the oxygen content, the lower heating value and the stoichiometric air-fuel ratio (AFR) were kept similar for all the fuels. Among the six fuels, two fuels represent regular- and premium-grade gasoline fuels with respect to octane quality in the North American market. The objective of this study was to assess the significance of RON and MON to the combustion characteristics of a modern SI engine. A single cylinder 4-stroke direct injection spark ignition (DISI) research engine was used as the experimental tool. The engine tests were conducted at the engine speed of 1800 rpm and the engine load ranging from 4 to 20 bar IMEP. Three market representative engine compression ratios (9.5:1, 10.5:1 and 11.5:1) were selected. In addition, the engine K value was calculated at knock-limited engine conditions. The results showed that, under knock-free engine operating conditions and at a fixed engine compression ratio, variation of fuel RON and MON had almost no differential impact on ITE. Under knock-limited operating conditions, increasing MON did not increase ITE, and in contrast, even led to decreased ITE especially when RON was as low as 93 and the compression ratio was high. Under knock-limited operating conditions, when the RON of the fuel was as high as 98, changing the MON up or down only showed combustion phasing benefits/disbenefits without obvious ITE benefit. This is because the octane rating of the fuel was high and in order to differentiate their anti-knock quality, a higher compression ratio than 11.5:1 was needed. The calculated engine K value shows that RON was a more significant influential factor than MON in determining the engine thermal efficiency. RON was found to have a higher impact on ITE at the higher MON of 88 vs. the lower MON of 83.

**Keywords:** RON; MON; DISI; Combustion; Indicated Thermal Efficiency

## Definitions, Acronyms and Abbreviations

| | |
|---|---|
| **AFR** | Air Fuel Ratio |
| **ATDC** | After Top Dead Centre |
| **BTDC** | Before Top Dead Centre |
| **°CA** | Crank Angle |
| **CAD** | Crank Angle Degree |
| **CFR** | Cooperative Fuel Research |
| **CR** | Compression Ratio |
| **COV** | Coefficient of Variation |
| **DI** | Direct Injection |
| **DISI** | Direct Injection Spark Ignition |
| **EGR** | Exhaust Gas Recirculation |
| **HOV** | Heat of Vaporization |
| **KLSA** | Knock Limited Spark Advance |
| **LHV** | Lower Heating Value |
| **IMEP** | Indicated Mean Effective Pressure |
| **ISFC** | Indicated Specific Fuel Consumption |
| **ITE** | Indicated Thermal Efficiency |
| **MFB** | Mass Fraction Burn |
| **MFB05** | Crank angle where 5% of fuel is burned |
| **MFB50** | Crank angle where 50% of fuel is burned |
| **MFB90** | Crank angle where 50% of fuel is burned |
| **MON** | Motor Octane Number |
| **NEDC** | New European Driving Cycle |
| **ON** | Octane Number |
| **OI** | Octane Index |
| **PFI** | Port Fuel Injection |
| **Pmax** | Peak in-cylinder pressure |
| PRFs | Primary Reference Fuels |
| **rpm** | Revolutions per Minute |
| **RON** | Research Octane Number |
| **SI** | Spark Ignition |
| **TDC** | Top Dead Centre |
| **vol**.% | Volumetric Percentage |
| **VVT** | Variable Valve Timing |
| $\eta$ | Indicated Thermal Efficiency |

## 1. INTRODUCTION

The transportation sector is facing pressures of increased light duty mobility demand and more stringent regulations on fuel economy and greenhouse gas emissions [1]. Even though hybrid and electric vehicles are gaining significant support and popularity, conventional vehicles powered by internal combustion engines will still be the main tool for light-duty transportation in the foreseeable future [2]. Therefore, improving the efficiency of internal combustion engines via better engine design is highly relevant [3-6]. Apart from improving the engine hardware, better fuel properties such as higher anti-knock quality can play a significant role in impacting the engine efficiency.

Historically, the anti-knock quality of gasoline fuels is described by two parameters, research octane number (RON) and motor octane number (MON), which are measured in standardized single cylinder naturally aspirated carburettor SI engines designed in the year of 1929, which are known as  cooperative fuel research (CFR) engines [7-9]. Details of RON and MON test procedures are defined in the ASTM standards D2699-08 and D2700-08, respectively [10, 11].

In the past 90 years since the introduction of CFR engines, internal combustion engines have developed significantly, driven by stringent fuel economy and emission standards [12-14]. Modern SI engines, especially the turbo-charged downsized designs tend to operate at relatively lower temperature but higher intake manifold pressure, resulting from the use of advanced hardware/technologies such as direct injection and charging intercooler [15]. In addition, the physiochemical properties of the reference fuels (iso-octane and *n*-heptane) used in CFR, called primary reference fuels (PRF), differ from gasoline available on the market, which consists of hundreds of hydrocarbons that have different properties such as boiling range, ignition delay, and octane sensitivity. Due to significant hardware and fuelling differences between the CFR and modern engines, the relevance of RON and MON to modern SI engines needs to be re-assessed.

The impact of RON has been studied by many investigators, and it is generally accepted that higher RON is beneficial to improving engine thermal efficiency [16-21]. However, the relevance of MON to modern gasoline engines is being challenged in the recent ten years [12, 22-24]. It was found that, for some engine types and at some operating conditions, a fuel with a low MON for a given value of RON could be beneficial in reducing engine knock tendency [12, 22, 25-27]. To address the disconnect between CFR and modern engines, an octane index (OI) was proposed [1]:

Equation 1: $$\mathrm{OI} = \mathrm{RON} - \mathrm{K} \times (\mathrm{RON} - \mathrm{MON}) = \mathrm{RON} - \mathrm{K} \times \mathrm{S}$$

where K is a weighting factor depending solely on in-cylinder temperature and pressure history experienced by the end-gas prior to the onset of auto-ignition; S, the difference between RON and MON, is the octane sensitivity. A higher OI indicates that the engine is more resistant to knock. If K is negative, a fuel with a high octane sensitivity is beneficial to suppressing engine knocking [28].

The engine K value can be determined through either experiments or modelling. The experiment method relies on the correlation of an engine/vehicle performance parameter relating to fuels' auto-ignition properties such as knock limited spark advance (KLSA) and acceleration time with an RON and MON de-correlated fuel matrix. Details regarding the experiment method can be found in the literature [12, 22, 26, 28-30]. For the modelling method, in-cylinder pressure data is required as an input, based on which the in-cylinder temperature is calculated. The crank angle of auto-ignition for a matrix of PRFs and toluene/n-heptane mixtures using the Livengood-Wu integral is calculated, and then the OI and K value of PRFs and toluene/n-heptane fuel mixtures are determined through the PRF calibration curve. Details about the modelling method can be found in research studies elsewhere [15, 27].

There are a few studies available in the literature, focusing on the K value of SI engines/vehicles. Mittal and Heywood [21] found that K values of the vehicles produced between 1951 and 1991 became lower and even negative due to the use of advanced cooling and breathing techniques, and the replacement of carburettors with fuel injectors. They [31] tested fuels with various RON and MON in a single cylinder port fuel injection (PFI) SI engine under one bar intake manifold pressure. The experimental results showed that K value was negative. K had a strong dependence on the intake air temperature, engine speed, and intake charge pressure.

Based on these findings, Mittal and Heywood [31] recommended modifying the octane rating tests to better bracket the knock limited operating conditions of modern SI engines.

Remmert et al. [27] studied the octane appetite and K value in a 4-cylinder DISI engine. Seven RON and MON decorrelated fuels were tested at several high load conditions. The impacts of external EGR, boost pressure, back pressure and lambda were investigated. They found that under high load conditions (approximately 20-30 BMEP), K value was in the range of -0.26 and -1.14. Davies et al. [15] investigated K value of several engines under high boost and EGR conditions. They found that K value was in the range of -0.86 to 0.5. Kalghatgi [29] reported that the averaged K value at full throttle conditions was -0.38 for 37 SI engines ranging from naturally aspirated to turbo-charged, and 1.2 L small engines to 2.4 L big engines. Orlebar et al. [30] conducted an octane sensitivity study on the model year 2007 Pontiac Solstice. They found that there was a clear negative correlation between charge pressure and K value.

Even though there are relevant publications available about the impact of fuel octane in spark-ignition (SI) engines, the RON and MON of the fuel matrices used in those studies were usually correlated, making the assessment of the individual contribution of RON and MON impossible. To address this knowledge gap, in this study six fuels were designed with independent control over RON and MON. The significance of RON and MON on the combustion characteristics were studied in a single cylinder DISI research engine. The experiments were conducted at the stoichiometric AFR with the engine speed of 1800 rpm and loads ranging from 4 and 20 bar IMEP using the fuel-specific optimum spark timing.

## 2. EXPERIMENTAL SYSTEMS AND METHODS

### 2.1. ENGINE AND INSTRUMENTATION

The experiment was conducted in an AVL single cylinder 4-stroke DISI research engine with 82 mm bore and 86 mm stroke, the setup of which is presented in Figure 1. Its combustion system features a 4-valve pent roof cylinder head equipped with variable valve timing (VVT) systems for both intake and exhaust valves. The cylinder head was equipped with a central-mounted outward opening piezo direct injector. The spark plug was located at the centre of the combustion chamber slightly tilting towards the exhaust side. The compression ratio (CR) of this engine was manually adjusted by placing various sized metal sheets between the cylinder liner and the crankcase.

The engine was coupled to an electric dynamometer, which was able to maintain the engine at a constant speed ($\pm$1 rpm) regardless of the engine power output. The engine was controlled via an IAV FI2RE management system. An AVL Indicom system was used for real-time combustion indication and analysis. A Siemens CATs system was used for data acquisition and recording, and it communicated with the IAV FI2RE and the AVL Indicom systems. The Siemens CATs system was also used for controlling air, fuel, coolant and lubricant conditioning equipment.

A Kistler pressure transducer was used for the in-cylinder pressure measurement, and it was installed in a sleeve on the intake and exhaust bridge. The cylinder pressure was collected via a charge amplifier (ETAS ES630.1). The sampling resolution was 0.1 crank angle (°CA) between -30 °CA and 70 °CA after top dead centre (ATDC), and 1 °CA in rest of the crank angles. Some key temperature and pressure measurement locations used are briefly labelled as ‘T’ and ‘P’ in Figure 1.

The engine intake system was connected with an external air handling device, capable of delivering up to 3 bar boosted air. Air was first filtered, dried, and then delivered to a conditioning unit with a capacity of approximately 200 L, where air pressure and temperature were precisely close-loop controlled. Temperatures of fuel, coolant and lubricant were controlled by individual AVL conditioning systems. Fuel consumption was measured by an AVL fuel mass flow meter.

## 2.2. FUEL PROPERTIES

Table 1 lists the properties of fuels in this study. All the fuels contained approximately 10 vol.% ethanol, and they were free of detergent additives or any other performance additives. F2 and F5 denote market realistic octane quality for a regular-grade (averaged knock index (AKI)=87) and a premium-grade (AKI=93) gasoline fuels in the North American market, respectively. Fuels 1-3 and Fuels 4-6 had RON of 93 and 98, respectively. However, Fuels 1-3 and Fuels 4-6 differed in MON with the sensitivity going from 5 to 15. Fuel 1 and Fuel 5 had the same MON but different RON, the same with Fuel 2 and F6. The other key properties such as the heat of vaporisation, oxygen content, lower heating value, stoichiometric AFR and density were kept similar across all fuels.

### 2.3. EXPERIMENTAL PROCEDURE

Table 2 lists the engine operating conditions. For each fuel at a certain engine operating condition, if the engine was not knock-limited, spark timing was adjusted by aiming the combustion centre (MFB50) at 7.5±0.5 °ATDC, which was an approximation of the maximum brake torque (MBT) spark timing. The term 'MFB50' stands for the crank angle position where 50% mass fraction of the fuel is burned. When engine knock occurred, spark timing was retarded to limit the knock intensity below the maximum tolerated intensity in order to avoid potential engine damage. The maximum tolerated knock intensity at 1800 rpm engine speed is 2 bar cycle-based maximum in-cylinder pressure oscillation. In this case, spark timing was termed as the knock limited spark advance (KLSA). The same valve timing and injection maps were used for all the fuels in this study (see Table 2).

The engine was first warmed up until the coolant, lubricant, fuel, and intake air temperatures were stabilised, which usually took 30 minutes. The engine was then run at a reference operating point (a daily checkpoint) in order to make sure that the engine was in good condition. The main parameters for the daily check were in-cylinder pressure, the peak in-cylinder pressure, MFB50, gaseous emissions and particulate emissions. After the engine passed the daily check, the test was executed according to the test protocol provided in Table 2. The maximum tolerated knock intensity at the engine speed of 1800 rpm was defined as 2 bar cycle-based maximum in-cylinder pressure oscillation. Raw in-cylinder pressure data was filtered by a 3-30 kHz band-pass filter, and then the filtered pressure data was rectified. The cycle-based maximum in-cylinder pressure oscillation was defined as the maximum pressure amplitude of the filtered and rectified in-cylinder pressure of that cycle. The cycle-based maximum in-cylinder pressure oscillation was calculated in real-time by AVL Indicom combustion analyser.

During the test protocol design stage, a design of experiment was conducted for intake and exhaust valve timings. One of the objectives was to avoid scavenging flow of air, which was achieved by carefully choosing the valve opening and closing timings, and also the control of exhaust back pressure. In this study, valve opening and closing timings were controlled via a variable valve timing system. The exhaust back pressure was controlled via an air-driven valve, and the control target was to match the exhaust back pressure with the intake manifold pressure. During the tests, three lambda values were obtained from lambda sensor, Horiba MEXA-7100D gas analyser, calculated value from the air and fuel flow meter readings. If there were scavenging flow of air, the lambda value calculated from the air and fuel flow meter readings would be higher than the readings from lambda sensor and Horiba MEXA-7100D gas analyser. During the tests, these three values matched well (±3% difference) with each other, which provided the confidence that there was no/minimal scavenging flow of air. For each engine operating point, the in-cylinder pressure data for 200 consecutive cycles were recorded. Low-frequency data were averaged and recorded for over a period of two minutes.

### 2.4. DATA PROCESSING

Net IMEP was calculated by the AVL Indicom. In the following test, Net IMEP is referred to as IMEP.

The heat release rate and mass fraction burned (MFB) were calculated and used to characterise the combustion process. In-cylinder pressure and the corresponding cylinder volume data were used to calculate the net apparent heat release rate based on the following equation:

Equation 2: $$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} \times P \times \frac{dV}{d\theta} + \frac{1}{\gamma-1} \times V \times \frac{dP}{d\theta}$$

where $\gamma$ is the ratio of specific heat capacities ($C_p/C_v$); P, V and $\theta$ are instant in-cylinder pressure, cylinder volume and crank angle position, respectively.

According to Heywood's book [32], γ for the unburned mixture (fuel, air and burned residual) is a function of temperature, equivalence ratio, and burned gas fraction, and it is typically in the range of 1.25 to 1.35 in spark ignition engines. γ of the equilibrium burned mixture is a function of equivalence ratio, and it is typically in the range of 1.18 to 1.28. In literature [33, 34], many researchers used a fixed γ (1.28-1.32) for heat releaser rate calculation. In this study, the exponent *n* for the compression and expansion processes ($PV^n$=constant) was calculated, and the crank angle windows for *n* calculation were 60 to 20 °bTDC for the compression stroke and 50 to 80 °aTDC for the expansion stroke. The exponent *n* for the compression process is comparable to the average value of γ for the unburned mixture over the compression process [32]. Therefore, the exponent *n* for the compression process is used to calculate heat release rate during the compression stroke. Because accurately estimating the in-cylinder temperature during combustion is challenging, and also the mixture composition inside the cylinder changes rapidly during combustion, γ for the expansion stroke has to be estimated. The appropriate values for γ during the combustion stroke which gives the most accurate heat-release information are not well defined [32]. In this paper, the exponent *n* for the expansion stroke was used to calculate heat release rate during the expansion stroke. It has to be pointed out that the actual γ (typically between 1.18 and 1.28) during expansion is lower than the exponent *n* for the expansion stroke due to heat loss to the combustion chamber. Nevertheless, it has been found out that the exponent *n* for the expansion stroke was in the range of 1.20 to 1.30; therefore, it is a reasonable estimation.

The definition of MFB is the accumulated released heat in successive crank angles ranging from the start of combustion to a certain crank angle degree divided by the total heat released in the entire combustion process. In the Appendix, Figure 1A shows an example of MFB calculated from differently fixed γ (ranging from 1.24 to 1.3), and γ using the polytropic exponents for the compression and expansion processes. It can be seen that the difference in the MFB profiles calculated from various γ is limited apart from MFB90 (approximately 4 CAD difference). The relatively large difference in MFB90 is mainly because the MFB90 lies in a flat region of the MFB profile, therefore γ have a relatively large impact on the MFB90.

Engine indicated thermal efficiency (ITE) is calculated by the following equation:

$$\text{Equation 3:} \quad \text{ITE} = \frac{30 \times s \times \text{IMEP} \times V_S}{\dot{m} \times LHV_{fuel}}$$

where s is the engine speed (rpm); IMEP is the engine load (Pa); Vs is the engine sweep volume ($m^3$); $\dot{m}$ is the fuel flow rate (kg/h); $LHV_{fuel}$ is the low heat value of the fuel (J/kg).

The averaged ITE at the 1800 rpm engine speed is defined as:

$$\text{Equation 4:} \quad \text{Averaged ITE} = \frac{\sum_1^n \text{ITE}_i}{n}$$

where $\text{ITE}_i$ is the ITE at the operating point $i$; n is the number of operating points being tested at the engine speed of 1800 rpm.

## 3. RESULTS AND DISCUSSION

There are three sections in the results and discussion. First, the impact of MON on engine combustion characteristics is discussed by comparing two groups of fuels, each of which had the same RON but different MON. Second, the impact of RON is discussed by comparing two groups of fuels, each of which had the same MON but different RON. In the end, discussions about the significance and comparison of RON and MON, and K value study are presented.

### 3.1. IMPACT OF MON

Figure 2 shows the effect of MON on ITE at the CR of 9.5:1, 10.5:1 and 11.5:1. The results of two groups of fuels are presented, including F1-F3 with RON of 93 (low RON group) in Figure 2 (a), and F4-F6 with RON of 98 (high RON group) in Figure 2 (b). In each group, there are three levels of MON. In the legend, RON and MON are given for each fuel. For example, 'F1_RON93/MON88' means F1 fuel with an RON of 93 and a MON of 88. In Figure 2 (a), the ITE of F1-F3 (low RON group) was hardly differentiated at the lowest CR of 9.5:1; however, F3 with MON of 79 showed statistically higher ITE than F1 with MON of 88 at the highest CR of 11.5:1, especially at medium to high engine loads. In Figure 2 (b), the differences in ITE among F4-F6 (high RON group) are smaller than these among F1-F3 (low RON group); nevertheless, at the CR of 11.5:1, F6 with MON of 83 led to a higher ITE than F4 with MON of 92 at the highest engine load. In both low and high RON group, the difference between fuels increased with engine load and compression ratio. For the low

RON group, the ITE of fuels started to be differentiated at the compression ratio of 10.5:1; whilst for the high RON group, the ITE of fuels started to be differentiated at the compression ratio of 11.5:1. This is because overall the high RON group had a higher anti-knock quality than the low RON group. It is excepted that, if the engine CR were to be increased further from 11.5:1 to 12.5:1, F6 would show a higher ITE than F4.

Overall, in both the low and high RON groups, a trend exists that lower MON resulted in a higher engine thermal efficiency. This contradicts the notion that high MON is always a positive indicator of fuel's anti-knock quality. Many publications agree with the findings from this study [13, 14, 21, 22, 30]. From the combustion perspective, in-cylinder parameters for the MON test in CFR engines deviate from those in a modern engine. Additionally, the PRFs used in the CFR engines have an octane sensitivity of zero; whilst the most market available fuels have a sensitivity of 5-15. PRFs have a stronger negative temperature coefficient (NTC) behaviour than conventional gasoline. Due to the above reasons, the MON measured in the CFR engine does not suitably characterise/predict the octane appetite of modern SI engines.

In order to develop a better understanding of the performance of these fuels, key combustion characteristics, including spark timing, combustion phasing, peak in-cylinder pressure (Pmax) and exhaust gas temperature are presented. Since the most distinctive differences were observed at the highest CR, only the results from CR of 11.5:1 are presented.

Figure 3 shows the effect of MON on combustion characteristics of F1-F3 (low RON group) at the CR of 11.5:1. The MFB'x' in Figure 3 stands for the crank angle position where 'x' percent of fuel is burned. Overall, the results in Figure 3 matched with the ITE results showed in Figure 2 (a). Since the spark timing used for each fuel in this study was KLSA when the engine was operated at knock-limited engine load, spark timing indirectly reflected the anti-knock quality of the fuels. In a DISI engine, the octane rating and charge cooling effect influence the engine knock. The charge cooling effect of these fuels in this DISI engine is comparable due to the similar heat of vaporisation of fuel and the same injection strategy; therefore, the spark timing differences in Figure 3 is mostly due to anti-knock quality differences. From Figure 3, F3 clearly enabled more advanced spark timing than other fuels at engine loads higher than 12 bar IMEP. For IMEP lower than 12 bar, the spark timing for each fuel was almost identical because the engine was either not limited by knock, or the knock amplitude was less than the upper knock limit for this engine. The results of MFB05 showed a similar trend as seen with spark timing, indicating that the initial combustion rates of F1-F3 were similar. The centre of combustion, or MFB50, is a key engine calibration parameter. If the engine is operated at knock-free load, the optimum combustion is achieved when MFB50 is in the region of 7.5±0.5 °ATDC, which can be seen for F1-F3 at 4-6.5 bar IMEP in Figure 3. The engine started to knock at 8 bar IMEP for all fuels, and spark timing had to be retarded from the optimum phasing in order to protect the engine from experiencing excessive knock. F3 allowed for more advanced MFB50 than F1 and F2, which explained the higher ITE of F3 in Figure 2 (a). Even though there were no differences between F1 and F2 in spark timing and MFB05, F2 enabled more advanced MFB50 at engine load higher than 12 bar IMEP, indicating that F2 had a faster burning rate than F1. Due to the use of more advanced spark timing, F3 led to a much higher peak pressure and lower exhaust temperature than F1 and F2 at medium and high engine load.

Figure 4 shows the effect of MON on combustion characteristics of F4-F6 (high RON group) at the CR of 11.5:1. The combustion characteristics of F4-F6 were highly similar, except for engine loads higher than 16 bar IMEP. The values of spark timing, combustion phase, peak in-cylinder pressure and exhaust temperature logically matched with the ITE results in Figure 2 (b). The results shown in Figure 4 (high RON group) differed from these shown in Figure 3 (low RON group) by the magnitudes of the differences in the key combustion parameters.

For the low RON group, the minimum knock-limited loads were 7.1, 7.0 and 6.5 bar IMEP for F1, F2 and F3 at the CR of 11.5:1, respectively. For the high RON group, the minimum knock-limited loads were 8.7, 8.2, 7.7 bar IMEP for F4, F5 and F6 at the CR of 11.5:1, respectively. It seemed that a fuel with a higher MON had a marginally higher minimum knock-limited load.

Based on the results in Figure 2, Figure 3 and Figure 4, it can be summarised that a fuel with low MON demonstrated better anti-knock property, especially when the fuel is subject to severe auto-ignition, such as at high CR and at high engine loads.

### 3.2. IMPACT OF RON

Figure 5 shows the effect of RON on ITE at the CR of 9.5:1, 10.5:1 and 11.5:1. The results for two groups of fuels are presented, including F1 and F5 with MON of 88 (high MON group) in Figure 5 (a), and F2 and F6 with MON of 83 (low MON group) in Figure 5 (b). The differences in RON in both groups were five. Results showed that a fuel with higher RON was beneficial to ITE. As the engine was more prone to knock, such as at high compression ratio and at high loads, the advantage of high RON became more obvious. Larger differences were observed in the high MON group than in the low MON group. This agreed with the findings in the previous section that, a fuel with a lower MON was more knock resistant than a fuel with the same RON but a higher MON.

Figure 6 and Figure 7 show the effect of RON on combustion characteristics of the high MON group (F1 and F5) and the low MON group (F2 and F6) at the CR of 11.5:1, respectively. Clearly, a fuel with a high RON enabled an advanced spark timing, an advanced combustion phasing, a higher peak in-cylinder pressure and a lower exhaust temperature. These differences were observed at engine loads as low as 8 bar IMEP. At 14 bar IMEP, F5 with five units higher RON than F1 led to 13 CAD more advanced spark timing, 22 CAD more advanced MFB50, 17 bar higher Pmax and 120 °C lower exhaust temperature; whilst at the same engine load, F6 with 5 units higher RON than F2 led to 10 CAD more advanced spark timing, 17 CAD more advanced MFB50, 14 bar higher Pmax and 85 °C lower exhaust temperature. Therefore, the high MON group showed more differences in combustion characteristics than the low MON group.

Overall, the findings in Figure 5, Figure 6 and Figure 7 agreed with the literature that high RON is beneficial to improve engine thermal efficiency [16-21]. Unlike the MON test, the RON test is better at characterising the anti-knock quality of gasoline-type fuels.

### 3.3. DISCUSSIONS ABOUT THE SIGNIFICANCE OF RON AND MON

Figure 8 and Table 3 show the summary of the effects of RON and MON on the average ITE. The average ITE for each fuel at a certain CR was calculated by averaging the ITE at all tested points at the engine speed of 1800 rpm. Overall, RON was found to be a more influential factor than MON in determining the ITE.

In comparison to F1, F5 with five units higher RON led to 3.40%, 9.71% and 8.55% higher averaged ITE at the CR of 9.5:1, 10.5:1 and 11.5:1, respectively, corresponding to 0.68%, 1.94% and 1.71% benefit in ITE for every unit increase of RON. In comparison with F2, F6 with five units higher RON led to 2.12%, 5.36% and 3.71% higher ITE than F2 at the CR of 9.5:1, 10.5:1 and 11.5:1, respectively, representing 0.42%, 1.07% and 0.74% benefit in ITE for every unit increase of RON. This showed that RON was found to have a higher impact on ITE at the higher MON of 88 vs. the lower MON of 83.

When the RON was at 93 (as in with F2), F3 with four units lower MON led to 2.42%, 2.27% and 2.78% higher ITE at the CR of 9.5:1, 10.5:1 and 11.5:1, respectively, which corresponds to 0.60%, 0.57% and 0.69% benefit in ITE for every unit decrease of MON. When the RON was at the level of 98, lowering MON from 88 to 83 did not show benefits in the averaged ITE. This is because the CR of 11.5:1 was not high enough to differentiate the octane quality of the high RON group. This also showed that decreasing MON led to a larger ITE gain at the lower RON of 93 vs. the lower RON of 98.

As mentioned in the introduction, the engine's octane appetite can be expressed as: OI=RON-K×(RON-MON). The engine K value was experimentally determined [13, 22, 27, 30]. The experimental method relies on measuring an engine performance parameter related to a fuel's auto-ignition property with an RON and MON decorrelated fuel matrix. In this study, the $R^2$ of RON versus MON is 0.19, showing a good decorrelation. In this study, the performance parameters, including spark timing, MFB50, Pmax and ITE, were used to determine the K value from the six fuels. More details of the experimental method can be found in the literature [27].

A linear relationship between the performance parameter and OI was assumed, such as Pmax = $\alpha + \beta \times$ OI. K was determined by minimising the sum of squared residuals between the experimental and predicted the performance parameter. Figure 9 shows an example of engine K value determination at 12 bar IMEP and the CR of 11.5:1. The K value under this engine operating condition is -0.12. Under this K value, the $R^2$ of OI and Pmax was maximised at 0.93.

Figure 10 shows the K value of knock-limited engine operating conditions. The results of the maximum and the minimum CR are presented in order to give a clear trend. From Figure 10 (a) and (b), it is clear that as at a fixed CR, K value consistently decreased as engine load increased from 12 bar to 14 bar IMEP under all the engine performance parameters. From Figure 10 (c), it is clear that at a fixed engine load, K value consistently decreased as the CR increased from 9.5:1 to 11.5:1. It can be also observed that the K values calculated by different engine performance parameters are different, which is largely due to the different response of these parameters to engine octane quality.

From the OI equation, it is defined that the contribution of RON to OI is |1-K|; whilst the contribution of MON is |K|. If K >0.5, the significance of RON is less than MON, and vice versa. It can be seen in Figure 10 that all the K values under various CR and engine loads were lower than 0.5; therefore, it is concluded that the contribution of RON was higher than that of MON in this particular single cylinder DISI engine. If K <0, then decreasing MON increases OI, and vice versa. From Figure 10, under most engine performance parameter, the K value of this engine was negative.

The K value results from this study match well with other publications [13, 15, 27, 29]. For example, Remmert et al. [27] studied the octane appetite and K value in a 4-cylinder DISI engine. Seven RON and MON decorrelated fuels were tested at several high engine load conditions. The impact of external EGR, boost pressure, back pressure and lambda were also investigated. They found that under those high-load conditions, K values were in the range of -0.26 and -1.14. Davies et al. investigated the K value of several engines under high boost and EGR conditions and they found that K values were in the range of -0.86 to 0.5 [15]. Kalghatgi [29] reported that the averaged K value at full throttle conditions was -0.38 for 37 SI engines ranging from NA to turbo-charged, and 1.2 L small engines to 2.4 L big engines, and the K value varied with engine running conditions. Kassai et al. [12]studied K value in a Nissan PFI turbocharged engine, and found that K value became more negative with increased boost pressure and lower engine speeds.

## 4. CONCLUSIONS

In this study, six market relevant fuels were designed with independent control over RON and MON. Among the six fuels, two fuels denoted market realistic fuels for the regular grade (AKI=87) and premium grade (AKI=93) gasoline fuels in the North American market. A modern single cylinder 4-stroke DISI engine was used as the experimental tool. The tests were conducted at the stoichiometric AFR with the engine speed of 1800 rpm and the engine load ranging from 4 to 20 bar IMEP using the fuel-specific optimum spark timing. Three engine compression ratios (9.5:1, 10.5:1 and 11.5:1) were tested for all fuels. The following are the conclusions drawn from results and discussions:

1. Under knock-free engine operating conditions and at a fixed engine compression ratio, the fuel RON and MON had no differential impact on ITE.
2. Under knock-limited engine operating conditions, increasing the MON did not result in a higher ITE, and in contrast, even led to the opposite effect especially when the RON was low and the compression ratio was high. For a fuel with an RON of 93, depending on the compression ratio (9.5:1-11.5:1), decreasing the MON from 83 to 79 led to 2.27%-2.78% gain of the average ITE. For a fuel with an RON of 98, lowering the MON only showed combustion phasing benefits at the highest studied engine load. This is because for fuels with RON of 98, in order to differentiate their anti-knock quality, a higher compression ratio than 11.5:1 would be needed.

3. Under knock-limited engine operating conditions, increasing the RON was beneficial to the ITE by enabling more advanced spark timing and better combustion phasing. For a fuel with a MON of 88, depending on the compression ratio (9.5:1-11.5:1), increasing the RON from 93 to 98 led to approximately 3.40%-9.71% gain in the average ITE; whilst for a fuel with a MON of 83, increasing the RON from 93 to 98 led to approximately 2.12%-3.71% gain. Therefore, RON was found to have a higher impact on ITE at the higher MON of 88 vs. the lower MON of 83.

4. RON exhibited a first order influence on the octane quality of gasoline fuels, and MON exhibited a secondary influence. This is supported by the results that the calculated K value of this engine was less than 0.5 under knock-limited operating condition.

5. Engine K value decreased with the increasing of engine load and CR under all the engine performance parameters selected in this study, including spark timing, MFB50, Pmax and ITE. Under the most engine performance parameters at knock-limited operating conditions, the K value of this engine was negative, indicating that a low MON fuel was beneficial.

**Future work**: The engine K value is dependent on the thermodynamic (temperature and pressure) history of the unburnt end gas; therefore, the K value is engine hardware and operating condition related. More K value studies on various SI engines with different hardware architectures and engine operating conditions are needed in order to draw further conclusions.

## Acknowledgement

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# Figures

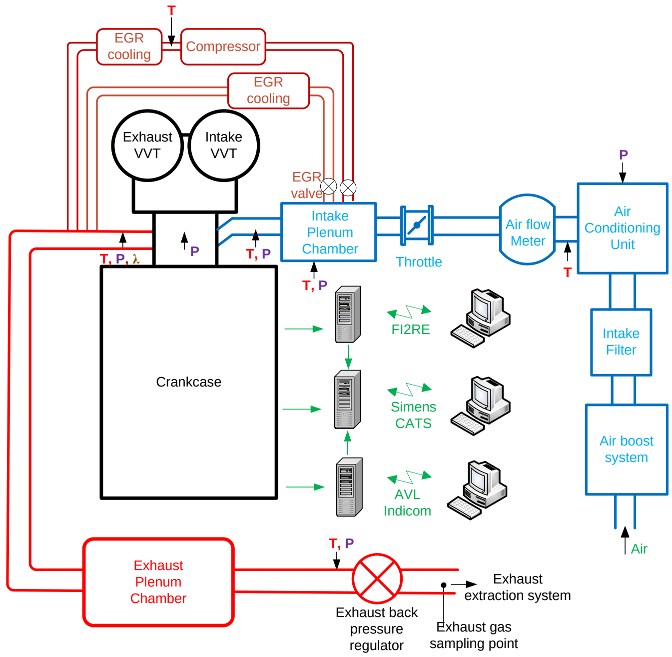
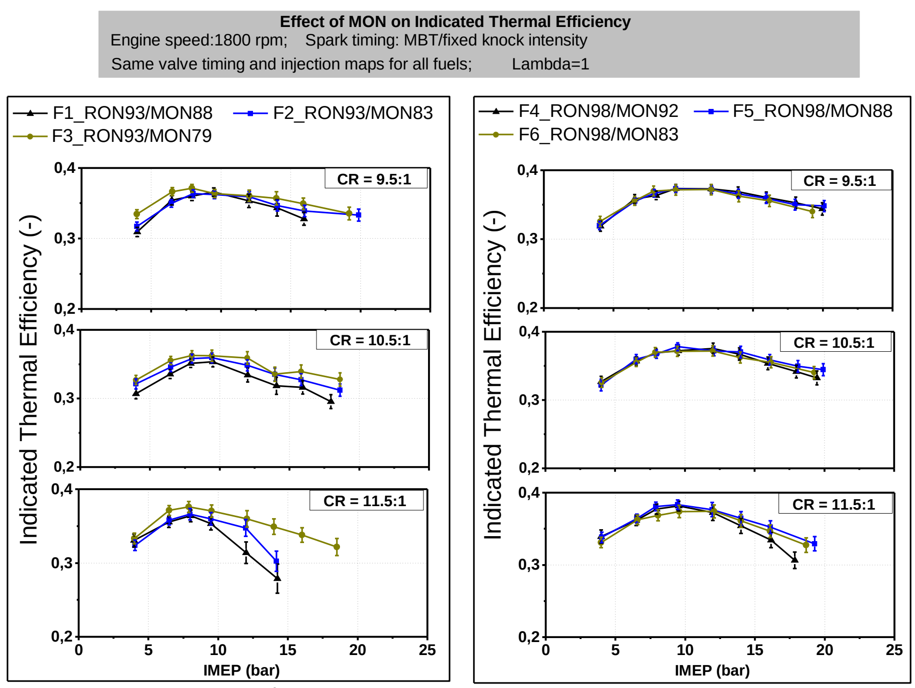



Figure 1: Engine setup




(a) F1, F2 and F3 (b) F4, F5 and F6

**Figure 2:** Effect of MON on indicated thermal efficiency at the CR of 9.5:1, 10.5:1 and 11.5:1. (a) F1, F2 and F3; (a) F4, F5 and F6

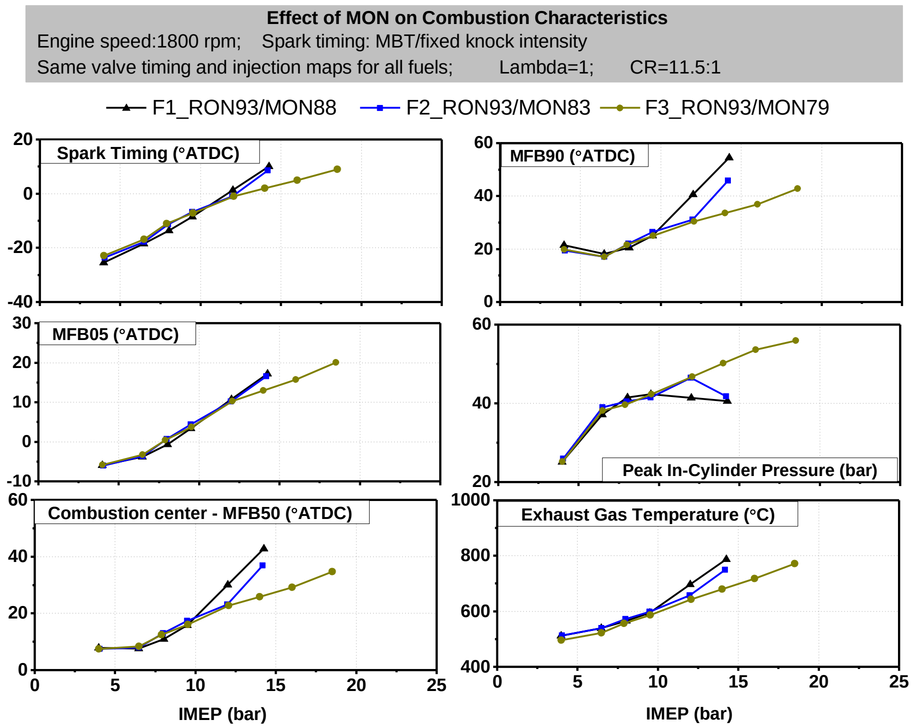



**Figure 3:** Effect of MON on combustion characteristics of F1, F2 and F3 (low RON group) at the CR of 11.5:1

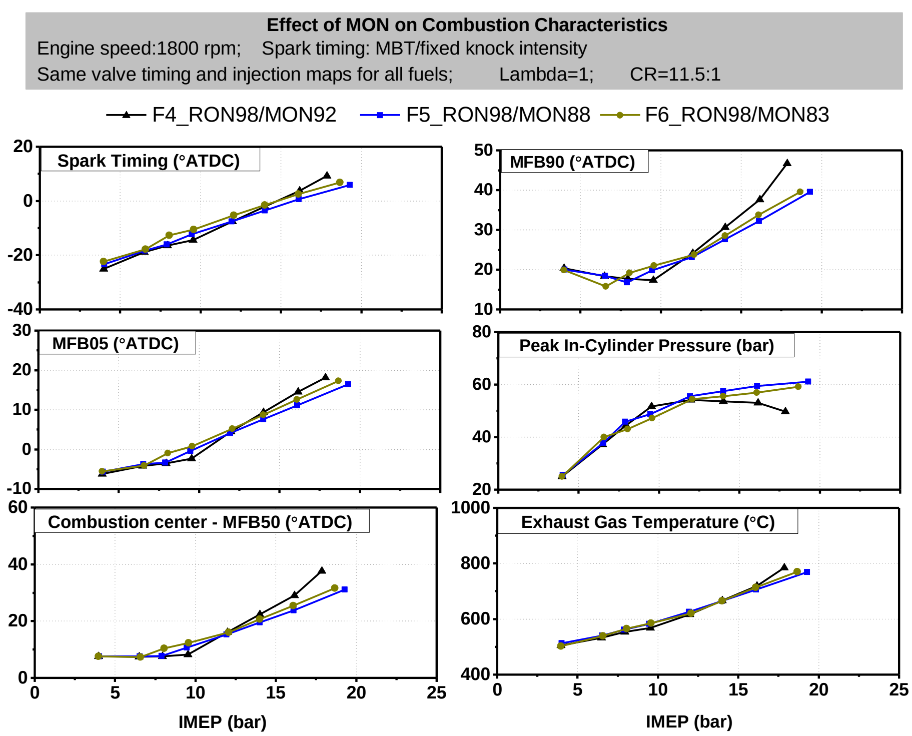
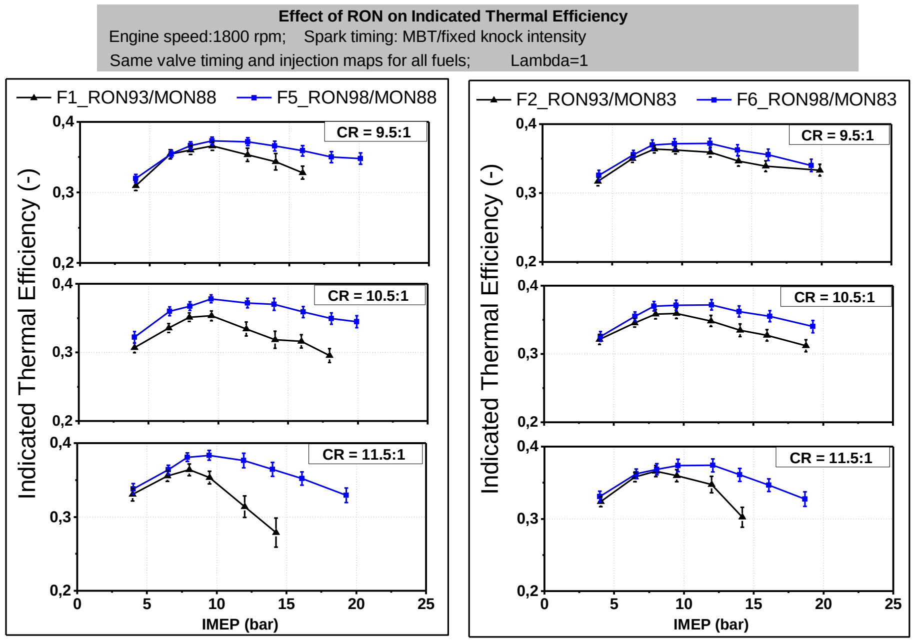



**Figure 4:** Effect of MON on combustion characteristics of F4, F5 and F6 (high RON group) at the CR of 11.5:1




(a) F1 and F5 (b) F2 and F6

**Figure 5:** Effect of RON on indicated thermal efficiency at the CR of 9.5:1, 10.5:1 and 11.5:1. (a) F1 and F5; (a) F2 and F6

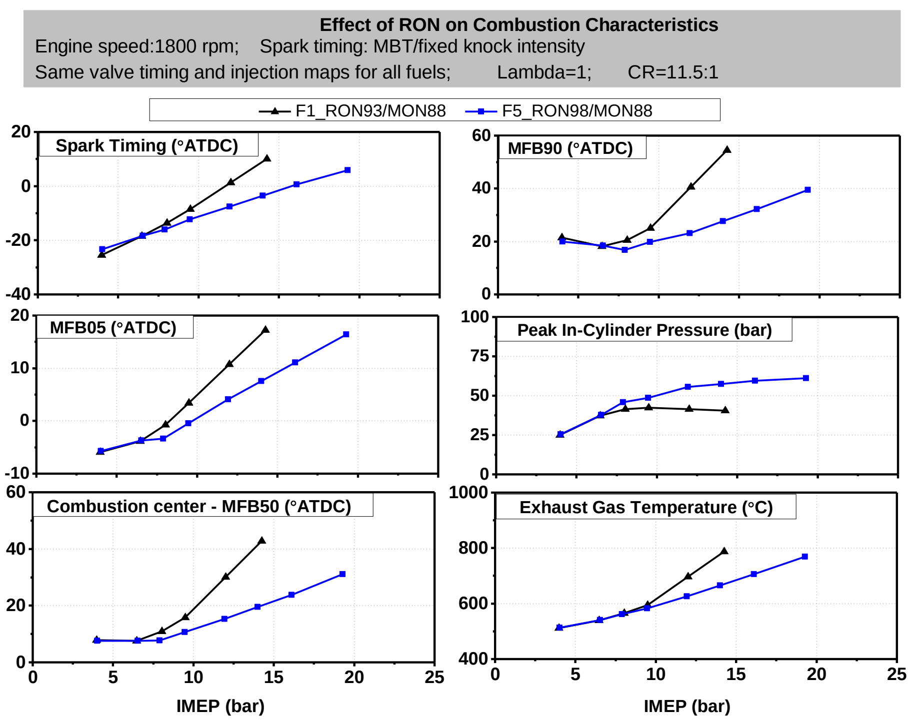



**Figure 6:** Effect of RON on combustion characteristics of F1 and F5 at the CR of 11.5:1

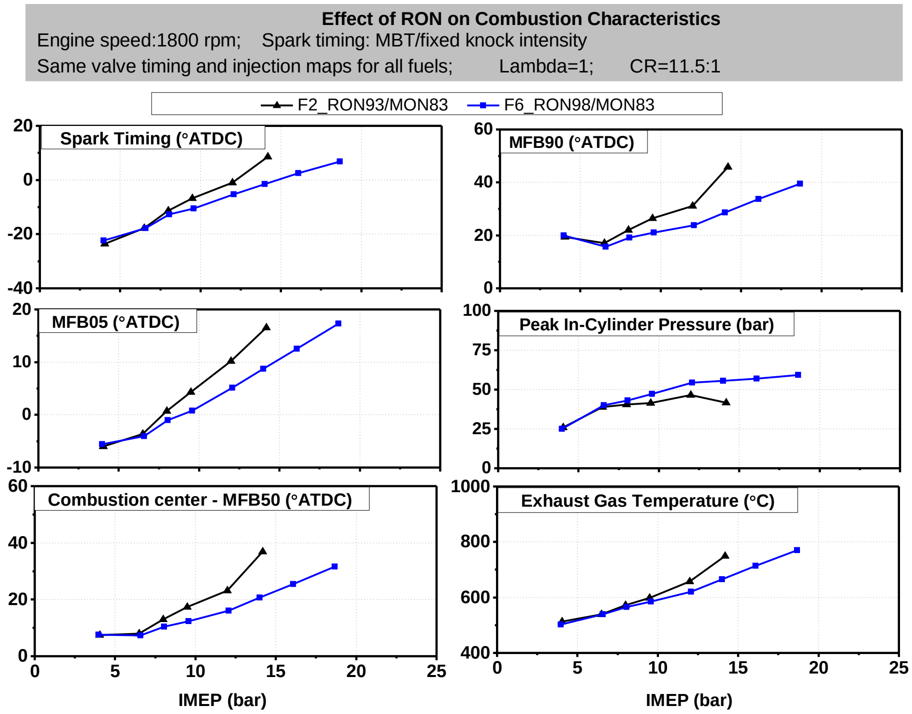



**Figure 7:** Effect of RON on combustion characteristics of F2 and F6 at the CR of 11.5:1

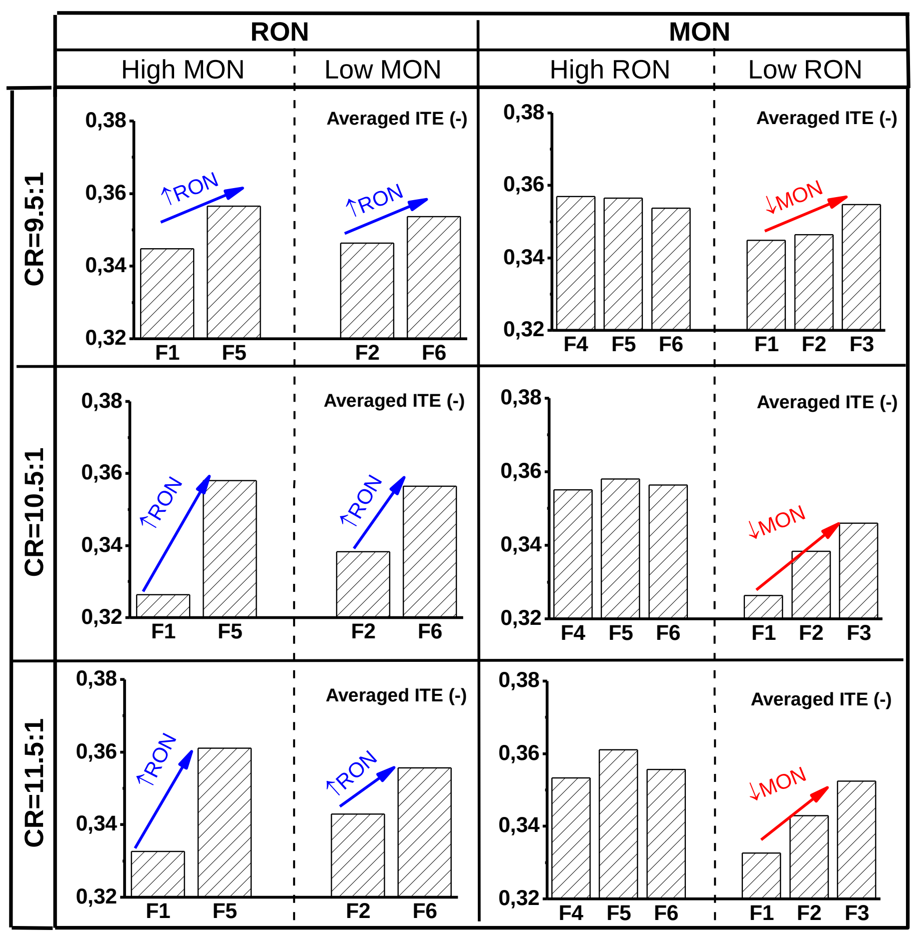



**Figure 8:** Summary of the effect of RON and MON on the averaged ITE at the CR of 9.5:1, 10.5:1 and 11.5:1

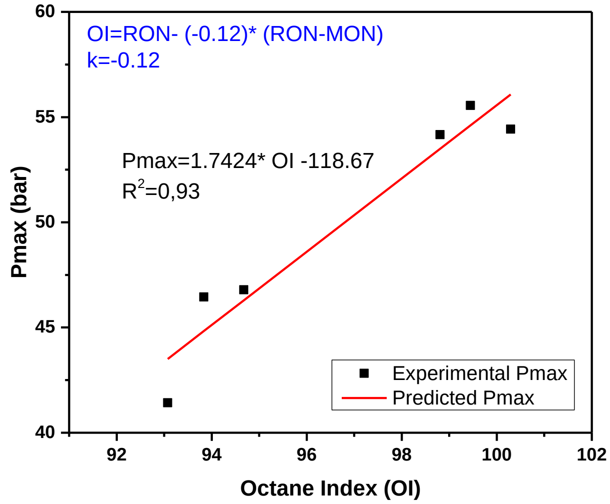



**Figure 9:** Octane Index versus Pmax (an example of engine K value determination @ 12 bar IMEP under CR 11.5:1)

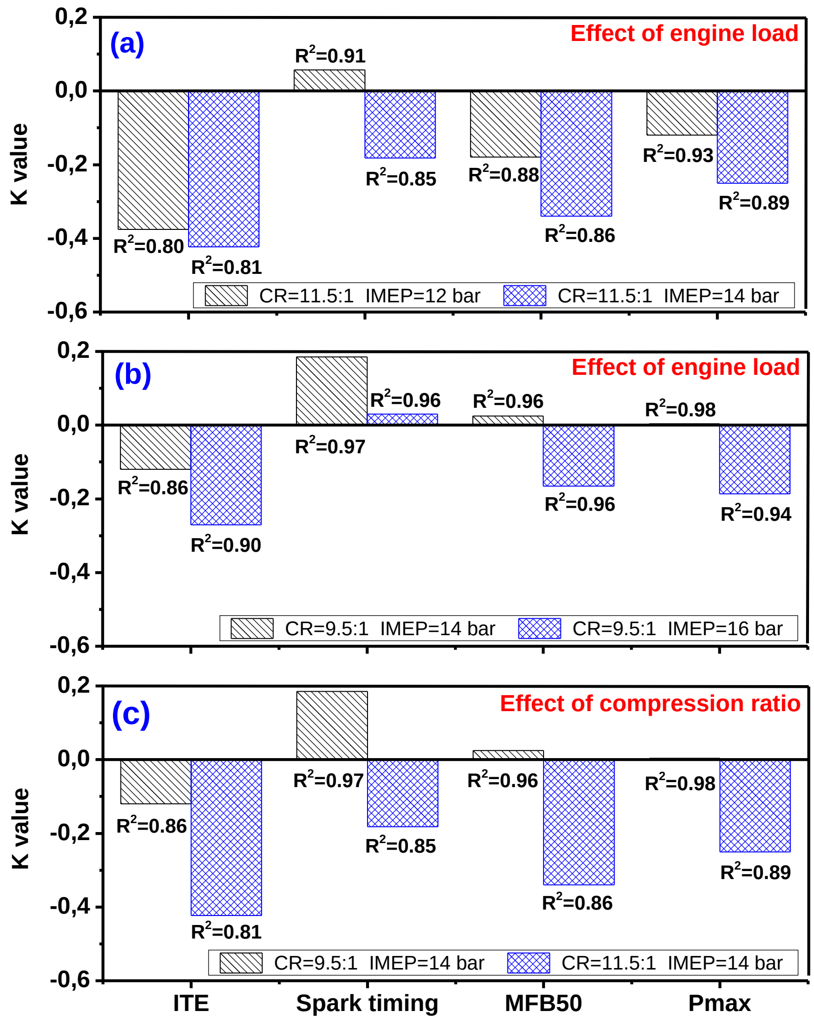



**Figure 10:** K value calculated by various engine performance parameters: (a) effect of engine load at the CR of 11.5:1; (b) effect of engine load at the CR of 9.5:1; (c) effect of compression ratio ($R^2$ for the linear fits shown in the figure)

# Tables

Table 1: Fuel properties

| | | F1 | F2 | F3 | F4 | F5 | F6 |
|---|---|---|---|---|---|---|---|
| **RON** | - | 93 | 93 | 93 | 98 | 98 | 98 |
| **MON** | - | 88 | 83 | 79 | 92 | 88 | 83 |
| **Octane sensitivity** | - | 5 | 10 | 14 | 6 | 10 | 15 |
| **Anti-knock index** | - | 90.5 | 88 | 86 | 95 | 93 | 90.5 |
| **HOV** | kJ/kg | 403 | 423 | 424 | 395 | 424 | 444 |
| **Oxygen content** | wt .% | 3.92 | 4.1 | 3.3 | 4.0 | 4.5 | 3.5 |
| **Lower heating value** | MJ/kg | 42.6 | 41.5 | 42.2 | 43.0 | 41.6 | 42.0 |
| | MJ/L | 30.6 | 30.8 | 30.8 | 30.1 | 30.3 | 31.4 |
| **Stoichiometric AFR** | - | 14.0 | 13.7 | 13.9 | 14.2 | 13.7 | 13.8 |
| **Density** | kg/m$^3$ | 718.0 | 742.0 | 731.1 | 698.6 | 730.0 | 749.0 |

Table 2: Summary of engine operating conditions*

| Speed | IMEP | Intake valve open/close timing @ 1mm valve lift | Exhaust valve open/close timing @ 1mm valve lift | Injection timing | Injection duration split ratio | Estimated absolute intake manifold pressure** |
|---|---|---|---|---|---|---|
| rpm | bar | °ATDC | °ATDC | °ATDC | | bar |
| 1800 | 4 | -12.2/179.2 | -204.4/7.0 | -280 | - | ≈0.60 |
| 1800 | 6.5 | -12.2/179.2 | -204.4/7.0 | -280; -240 | 1:1 | ≈0.75 |
| 1800 | 8 | -12.2/179.2 | -204.4/7.0 | -280; -240 | 1:1 | ≈0.86 |
| 1800 | 9.5 | -12.2/179.2 | -204.4/7.0 | -280; -240 | 1:1 | ≈0.96 |
| 1800 | 12 | -2.2/189.2 | -214.3/-3.0 | -280; -240; -200 | 1:1:1 | ≈1.20 |
| 1800 | 14 | -2.2/189.2 | -214.3/-3.0 | -280; -240; -200 | 1:1:1 | ≈1.48 |
| 1800 | 16 | 12.8/204.1 | -214.3/-3.0 | -280; -240; -200 | 1:1:1 | ≈1.64 |
| 1800 | 18 | 17.8/209.1 | -214.3/-3.0 | -280; -240; -200 | 1:1:1 | ≈1.80 |
| 1800 | 20 | 17.8/209.1 | -214.3/-3.0 | -325; -285; -245; -205 | 1:1:1:1 | ≈2.10 |

* To avoid a scavenging flow of air, the back pressure was set to be the same as the intake manifold pressure in case that intake air boosting was needed. Intake air temperature was conditioned at 34 ℃ by an air-handling device.
** The actual intake manifold pressure required for achieving a target IMEP would depend on the fuel and the engine compression ratio.

Table 3: Averaged ITE benefit under various CR at the 1800 rpm engine speed

| | Averaged ITE benefit (%) | | | | | |
|---|---|---|---|---|---|---|
| | Fuel 2 as base fuel | | | | Fuel 5 as base fuel | |
| | F2 vs. F1 | F2 vs. F3 | F2 vs. F5 | F2 vs. F6 | F5 vs. F4 | F5 vs. F6 |
| CR=9.5:1 | -0.44 | 2.42 | 2.94 | 2.12 | 0.12 | -0.79 |
| CR=10.5:1 | -3.54 | 2.27 | 5.82 | 5.36 | -0.81 | -0.43 |
| CR=11.5:1 | -2.99 | 2.78 | 3.70 | 3.71 | -2.16 | -1.52 |

# Appendix

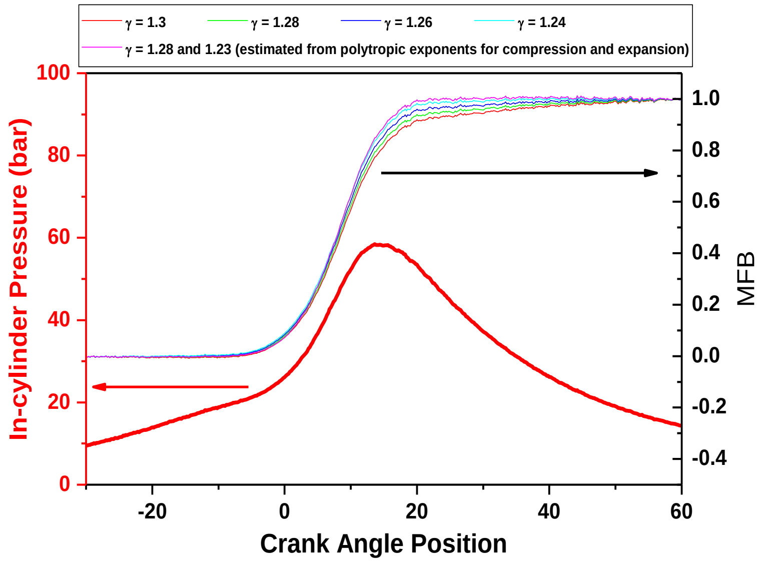



Figure 1A: Sample of in-cylinder pressure trace and corresponding MFB profiles calculated from various γ (Engine speed= 1800 rpm; IMEP=9.5 bar)

## Reference

[1] T. Johnson, Vehicular Emissions in Review, SAE Int. J. Engines, 9 (2016) 1258-1275.
[2] US Department of Energy, Quadrennial Technology Review 2015 - Chapter 8: Advancing Clean Transportation and Vehicle Systems and Technologies, http://energy.gov/sites/prod/files/2015/11/f27/QTR2015-8C-Internal-Combustion-Engines.pdf, (2015).
[3] H. Xu, C. Wang, X. Ma, A.K. Sarangi, A. Weall, J. Krueger-Venus, Fuel injector deposits in direct-injection spark-ignition engines, Progress in Energy and Combustion Science, 50 (2015) 63-80.
[4] C. Wang, R. Daniel, X. Ma, Comparison of Gasoline (ULG), 2, 5-Dimethylfuran (DMF) and Bio-Ethanol in a DISI Miller Cycle with Late Inlet Valve Closing Time, in, SAE Technical Paper 2012-01-1147; 2012.
[5] C.M. Wang, H.M. Xu, J.M. Herreros, T. Lattimore, S.J. Shuai, Fuel Effect on Particulate Matter Composition and Soot Oxidation in a Direct-Injection Spark Ignition (DISI) Engine, Energy & Fuels, 28 (2014) 2003-2012.
[6] C. Wang, H. Xu, T. Lattimore, Impacts of Low-Level 2-Methylfuran Content in Gasoline on DISI Engine Combustion Behavior and Emissions, SAE Technical Paper 2013-01-1317; 2013.
[7] W.R. Leppard, The chemical origin of fuel octane sensitivity, SAE Technical Paper 902137; 1990.
[8] H. Dickinson, The Cooperative Fuel Research and Its Results, SAE Technical Paper 310019; 1929.
[9] H. Horning, The Cooperative Fuel-Research Committee Engine, SAE Technical Paper 310019; 1931.
[10] A. International, Standard test method for research octane number of spark-ignition engine fuel, ASTM D2699-08 (2008).
[11] A. International, Standard Test Method for Motor Octane Number of Spark-Ignition Engine Fuel, ASTM D2700-08 (2008).
[12] G. Kalghatgi, Fuel anti-knock quality-Part I: Engine studies, SAE Technical Paper 2001-01-3584. 2001.
[13] A. Prakash, R. Cracknell, V. Natarajan, D. Doyle, A. Jones, Y.S. Jo, M. Hinojosa, P. Lobato, Understanding the Octane Appetite of Modern Vehicles, SAE Int. J. Fuels Lubr., 9 (2016) 345-357.
[14] CRC, Fuel Antiknock Quality - Engine Response to RON versus MON, Coordinating Research Council Report No. 660, (2011).
[15] T.J. Davies, R.F. Cracknell, B. Head, K. Hobbs, T. Riley, A new method to simulate the octane appetite of any spark ignition engine, SAE Technical Paper 2011-01-1873; 2011.
[16] S. Binjuwair, T.I. Mohamad, A. Almaleki, A. Alkudsi, I. Alshunaifi, The effects of research octane number and fuel systems on the performance and emissions of a spark ignition engine: A study on Saudi Arabian RON91 and RON95 with port injection and direct injection systems, Fuel, 158 (2015) 351-360.
[17] R. Daniel, H. Xu, C. Wang, D. Richardson, S. Shuai, Combustion performance of 2,5-dimethylfuran blends using dual-injection compared to direct-injection in a SI engine, Applied Energy, 98 (2012) 59-68.
[18] C.M. Wang, H.M. Xu, R. Daniel, A. Ghafourian, J.M. Herreros, S.J. Shuai, X. Ma, Combustion characteristics and emissions of 2-methylfuran compared to 2,5-dimethylfuran, gasoline and ethanol in a DISI engine, Fuel, 103 (2013) 200-211.
[19] K. Nakata, D. Uchida, A. Ota, S. Utsumi, K. Kawatake, The impact of RON on SI engine thermal efficiency, SAE Technical Paper 2007-01-2007; 2007.
[20] Y. Duchaussoy, P. Barbier, P. Schmelzle, Impact of gasoline RON and MON on a turbocharged MPI SI engine performances, SAE Transactions, 113 (2004) 1446-1453.
[21] V. Mittal, J.B. Heywood, The shift in relevance of fuel RON and MON to knock onset in modern SI engines over the last 70 years, SAE International Journal of Engines, 2 (2009) 1-10.
[22] G.T. Kalghatgi, K. Nakata, K. Mogi, Octane appetite studies in direct injection spark ignition (DISI) engines, SAE Technical Paper 2005-01-0244; 2005.
[23] T. Davies, R. Cracknell, G. Lovett, L. Cruff, J. Fowler, Fuel effects in a boosted DISI engine, SAE Technical Paper 2011-01-1985; 2011.
[24] P. Risberg, G. Kalghatgi, H.-E. Ångstrom, Auto-ignition quality of gasoline-like fuels in HCCI engines, SAE Technical Paper 2003-01-3215; 2003.

[25] G. Kalghatgi, Fuel/Engine Interactions, SAE International, 2014.
[26] G.T. Kalghatgi, Fuel anti-knock quality-Part II: Vehicle Studies-how relevant is Motor Octane Number (MON) in modern engines, SAE Technical Paper 2001-01-3585; 2001.
[27] S. Remmert, S. Campbell, R. Cracknell, A. Schuetze, A. Lewis, K. Giles, S. Akehurst, J. Turner, A. Popplewell, R. Patel, Octane Appetite: The Relevance of a Lower Limit to the MON Specification in a Downsized, Highly Boosted DISI Engine, SAE International Journal of Fuels and Lubricants, 7 (2014) 743-755.
[28] G.T. Kalghatgi, R. Head, The available and required autoignition quality of gasoline-like fuels in HCCI engines at high temperatures, SAE Technical Paper 2004-01-1969, 2004
[29] G.T. Kalghatgi, Auto-ignition quality of practical fuels and implications for fuel requirements of future SI and HCCI engines, SAE paper 2005-01-0239. 2005.
[30] C.N. Orlebar, A. Joedicke, W. Studzinski, The effects of octane, sensitivity and K on the performance and fuel economy of a direct injection spark ignition vehicle, SAE Technical Paper 2014-01-1216; 2014.
[31] V. Mittal, J.B. Heywood, The Relevance of Fuel RON and MON to Knock Onset in Modern SI Engines, SAE International 2008-01-2414; 2008.
[32] J.B. Heywood, Internal combustion engine fundamentals, McGraw-Hill: New York, 1989.
[33] F. Zhang, S. Zeraati Rezaei, H. Xu, S.-J. Shuai, Experimental Investigation of Different Blends of Diesel and Gasoline (Dieseline) in a CI Engine, SAE International 2014-01-2686; 2014.
[34] M. Dalla Nora, H. Zhao, High load performance and combustion analysis of a four-valve direct injection gasoline engine running in the two-stroke cycle, Applied Energy, 159 (2015) 117-131.