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Combustion and Emission Analysis of SI Engine Fuelled by Saudi Arabian Gasoline RON91 and RON95 with Variable Compression Ratios and Spark Timing

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Abstract

The performance, emissions and combustion characteristics of a spark ignition engine at varying compression ratio and spark timing using two grades of gasoline in Saudi Arabian market, RON91 and RON95 have been presented on this paper. The tests were conducted using a single cylinder, four-stroke spark ignition research engine that operates with different RON fuel. The experiments were conducted at a wide-open throttle with an engine speed of 2500 rpm at stoichiometric conditions for three different compression ratios (9.5:1, 10:1, 10.5:1). At each compression ratio, the spark timing was varied till knocking is detected. The performance characteristics such as brake power, specific fuel consumption, thermal efficiency, in-cylinder pressure, rate of heat release, and exhaust emissions were analyzed. The data shows that the increasing of compression ratio and spark timing increased the brake power and decreased the brake fuel consumption for both type of fuels. Increasing compression ratio was more advantageous for RON95 due to the auto-ignition resistance event. The NO_x emission was insignificantly decreasing as compression ratio increases with lower NO_x emission from RON91. Compression ratio had considerable effect on CO emission for both fuels.

Subject Areas

Mechanical Engineering

Keywords

Compression Ratio, Spark Timing, RON, Exhaust Emissions

1. Introduction

The growing demand on internal combustion (IC) engines, in particular spark ignition (SI) engines, coupled with tightening emission regulations have driven the development of more powerful, efficient and cleaner SI engines [1]. In the last decades, SI engines received considerable effort to enhance the combustion process, reduce fuel consumption, improve fuel formulations and decrease undesirable pollutant emissions. There are several ways to improve SI engines efficiency and emissions including engine design, formulation of fuels, compression ratio, fuel injection and mixing, ignition timing, treatment of reactant and exhausts, etc. [2].

The compression ratio of the engine that is determined according to fuel formulation and engine geometry represents one of the most important parameters that affect the performance of SI engines. Compression ratio is defined as the ratio of the maximum cylinder volume to the minimum compressed volume [3]. The impact of compression ratio on engine thermal efficiency and exhaust emissions is significant. According the Carnot principle, engine theoretical thermal efficiency is enhanced by increasing the compression ratio [4]. However, in SI engines, increasing the compression ratio is restricted by the fuel-knocking limit. Engine knocking phenomena occurs due to the undesirable auto ignition of the air-fuel mixture, which leads to an irregular combustion process. This engine knock adversely influences the engine performance including low thermal efficiency and tendency to mechanical failure. Fuel octane number is another important parameter in SI gasoline engines concerning the fuel anti-knock quality [5]. Higher fuel octane number allows higher compression ratio and offers better knock resistance [3]. In addition, spark timing is a potential parameter regarding engine knocking, particularly at variable engine load operation.

Several experimental investigations on local Saudi Arabian RON grades were carried out previously by the authors to study the effect of important combustion parameters on engine performance and emissions formation. These studies include determining the effect of spark timing [6] and fuel injection system [7] on engine performance using Saudi Arabian market gasoline: RON91 and RON95. Results from these studies showed that spark timing and fuel injection system influence combustion and emissions significantly. It was observed that fuel octane number has a major effect on engine efficiency and exhaust emissions. In these researches, the increasing of spark timing increased the brake power and decreased BSFC for both type of fuels with RON95 showing higher brake power and lower BSFC than RON91. In addition, it has been found that RON91 emitted lesser NO_x emissions and higher CO emissions than RON95. Regarding the fuel injection system, mixed results were obtained to different engine speed and fuel system [7].

Studies on the effect of compression ratio on performance, combustion characteristics and emissions of SI engines concerning different fuel formulation including gasoline blends, bio-fuels, hydrogen, liquefied Petroleum gas, compressed natural gas, etc., have been conducted by several investigations. Several researches in the literature studied the effect of compression ratio on SI engines performance fuelled with gasoline. Attrad *et al*. [8] conducted experiments to investigate the SI engine performance and exhaust emissions run at different compression ratios. The tests were performed on a two-cylinder SI engine fuelled with RON98 under compression ratios ranging from 9:1 to 13:1. They reported that with the increase of the compression ratio, higher brake power output and better fuel economy were obtained. They stated that the increase of compression ratio resulted in higher in-cylinder pressure due to the rapid initial burn rate. Costa and José [9] investigated the influence of compression ratio on a SI engine performance experimentally. The investigation compared 10:1, 11:1 and 12:1 compression ratios of a four-cylinder engine with gasoline and hydrous ethanol as fuel. It was found that increasing compression ratio resulted in a significant increase of the in-cylinder pressure and brake power, and a decrease in the fuel consumption, hence improved thermal efficiency. Sayin *et al*. [10] experimented on a SI engine operated with gasoline and iso-butanol/gasoline blend fuel run at compression ratios of 9:1, 10:1 and 11:1. The results show that the increase of compression ratio led to better thermal efficiency and minimum fuel consumption. In addition, lower CO and THC emissions were observed at higher compression ratios.

Numerical and experimental investigations were carried out by Smith *et al*. [11] to study the effect of compression ratio from 8 to 13.4 on SI engines performance. The observations indicated that increasing compression ratio revealed an improvement in the brake thermal efficiency by 5.1%. Aina *et al*. [12] conducted an experimental and a theoretical study to evaluate the effect of compression ratio on SI engine efficiency. In this study, as compression increased, it was found that a decrease in specific fuel consumption by 7.75%, an improvement in the brake thermal efficiency by 8.49%, and an increase in brake power by 1.34% were obtained. A number of studies have been also conducted regarding the effect of compression ratio on the performance of Homogeneous Charge Compression Ignition (HCCI) engines. Christensen *et al*. [13] tested several compression ratios over the range of 10:1 to 28:1 on HCCI with a constant AFR of 3.0 using a dual port injection system. The results showed that reduction of NO_x emissions was observed with increased compression ratio. However, lower combustion efficiency was obtained with an increasing compression ratio.

In recent years, employment of alternative fuels for internal combustion engines has been developing extensively. Alternative fuels have gained in popularity due to many reasons that depends on the fuel type including reduction of pollutant emissions, increase of knock resistance, increase of thermal efficiency and sustainability. Alternative fuels and blends for SI engines include bio-fuels, hydrogen, liquefied Petroleum gas, compressed natural gas, iso-butanol, etc. The compression ratio effect on the performance, combustion and emission characteristics of gasoline SI engines operated on alternative fuels have been conducted by several researchers. Abdel-Rahman *et al.* [14] reported that using ethanol blended gasoline fuel improved thermal efficiency with the increase in compression ratio. They observed that there is an optimum value of compression ratio for maximum engine thermal efficiency based on the fuel blend. Zhao *et al.* [15] observed the performance of a SI engine operated with hydrogen-enriched natural gas at varied compression ratio. The results indicated that higher compression ratio improved the combustion conditions.

In view of the literature review, the effect of compression ratio related to fuel type need to be studied clearly. Therefore, in this study, two available gasoline grades from Saudi Arabian market (RON91 and RON95) were investigated with respect to compression ratio to analyze and evaluate the effects of compression ratio on SI engine performance and exhaust emissions. Experimental investigation using the two local RON grade gasoline fuels were conducted under three different compression ratios at various spark timing from 20˚CA BTDC till knocking was detected at WOT and engine speed of 2500 rpm. The performance, exhaust emission and combustion characteristics were studied and compared. The experimental findings will benefit in the development of engine performance regarding fuel formulation.

2. Experimental Investigation

2.1. Experimental Setup

A Lotus Single Cylinder Research Engine (SCRE) located at the King Abdulaziz City for Science and Technology (Combustion Lab) was utilized to conduct this research. The engine is a four-stroke, spark-ignition (SI) gasoline engine with water-cooled and naturally aspirated. **Table 1** demonstrates Engine specifications and run settings. In order to study the influence of the compression ratio (CR) on the engine combustion and performance, three compression ratios (9.5:1, 10:1, 10.5:1) were employed by using three modified piston crown that were prepared. **Figure 1** shows the flow chart of the experiments and **Figure 2** shows piston crowns for three compression ratios.

Figure 3 shows the test cell comprises of SCRE and the 30 kW Eddy-current type dynamometer. **Figure 4** shows a schematic diagram of the setup. The fuel

Table 1. Specification of SCRE.

Type of engine	Single cylinder, 4 stroke and SI engine
Fuel injection system	PI
Stroke (mm)	88.2
Bore (mm)	80.5
Compression ratio	9.5:1, 10:1 and 10.5:1
Displacement (cm^3)	448.9
IVO/IVC	451˚/589˚
EVO/EVC	131˚/269˚

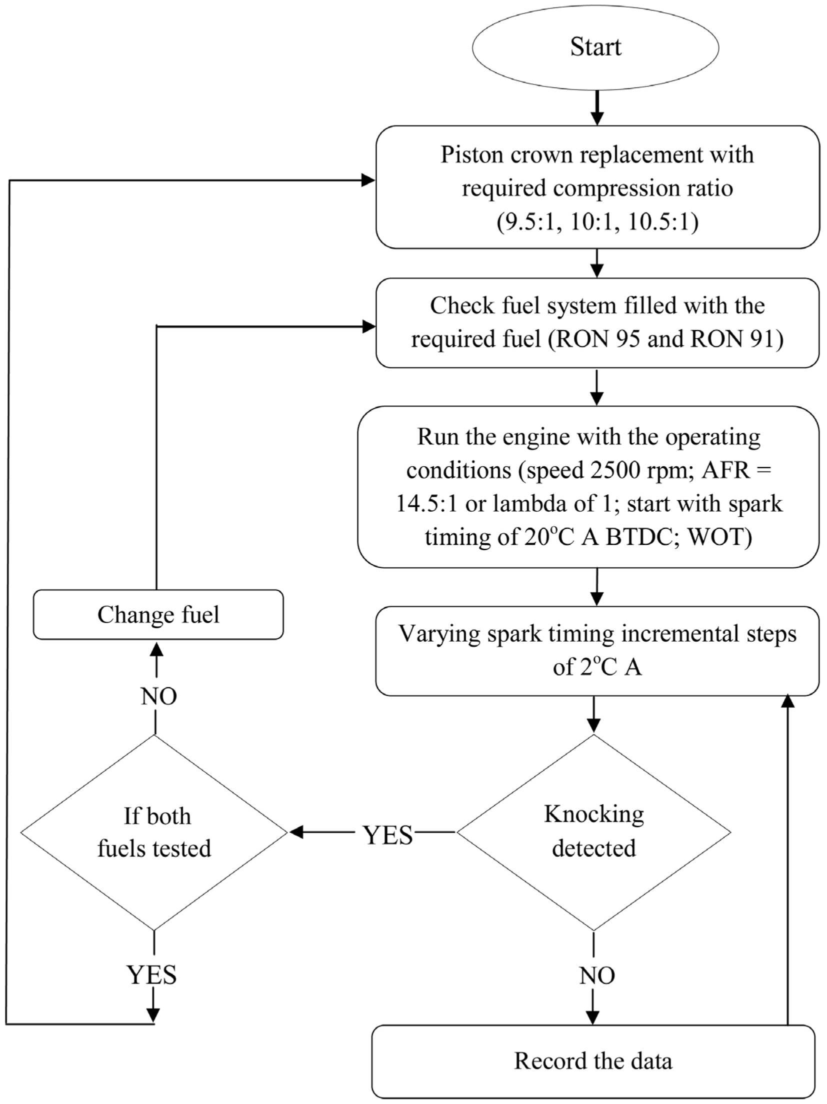

Figure 1. Flow chart of the experiments.

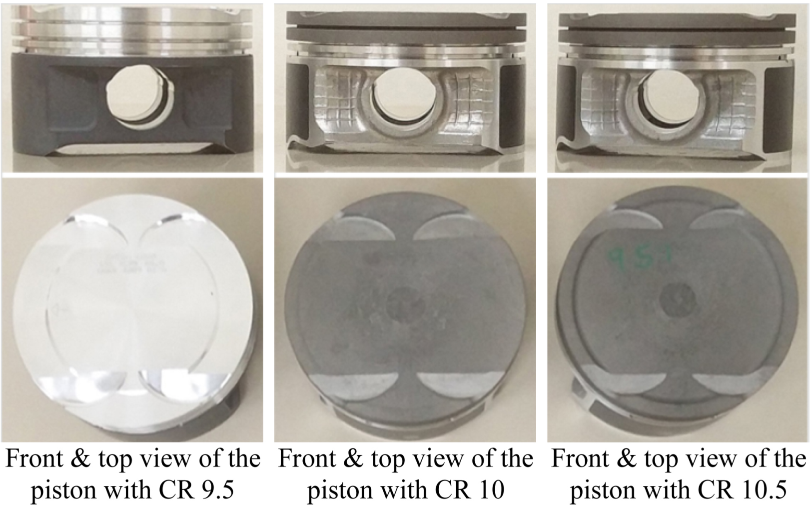

Figure 2. Piston crowns for various compression ratio.

Figure 3. Single cylinder engine test cell: 1—eddy current dynamometer; 2—single cylinder; 3—valve cam box; 4—Balance gear box; 5—Exhaust pipe; 6—Intake filter.

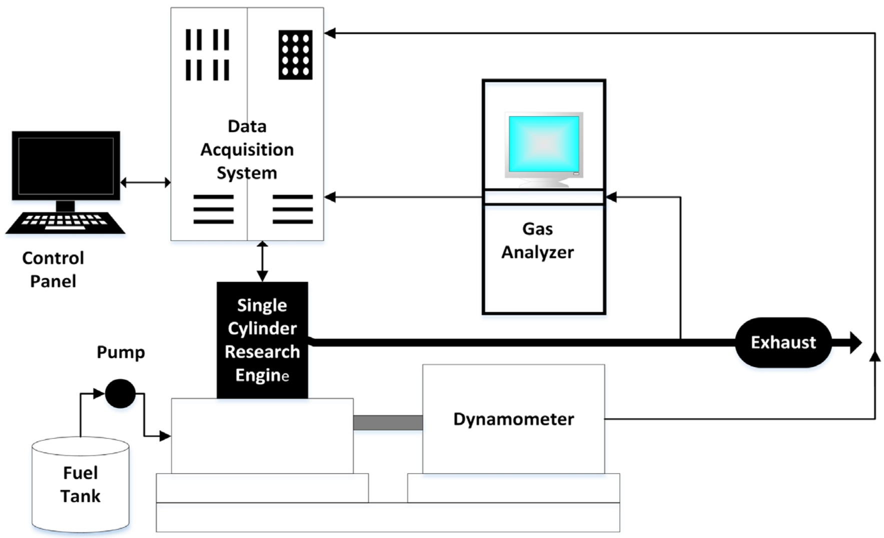

Figure 4. A schematic diagram of the experimental setup.

consumption rate was logged using a positive displacement meter (PLU) over a flow rate between 0.03 to 20l/h. Water cooled piezoelectric sensor (Kistler 6061B) was used to measure the pressure data inside the cylinder. The ECM meter (AFRecorder 2000) with accuracy of 0.01 - 0.03 was used to measure the air fuel ratio (AFR). Apiezo resistive sensor (Kistler 4045A10) was used to measure intake and exhaust pressure, and both sensors were subsequently recorded using AVL's Indicom software V1.6 based data acquisition system. An emission analyzer (MEXA-7170DEGR) from Horiba was used to measure exhaust gas emissions of CO, THC and NO_x. A fuel injector made by Bosch actuated by engine control unit was used for the port fuel injection. To control the spark timing electronically, a Lotus V8 controller was used.

2.2. Uncertainty Analysis

Uncertainty calculations were done as the method described by Shyam [16]. The recording of 100 cycles of in-cylinder pressure at a resolution of crank angle 0.5 ˚CA was averaged to compute combustion-related parameters. Table 2 shows the measurements accuracy and uncertainty of apparatuses and Table 3 shows calculated parameters utilized in this experimental investigation. The generalized approach for uncertainties calculations is used to determine the uncertainties in all calculated parameters is as follows:

For a generalized result $R = f\left(\overline{p_1}, \overline{p_2}, \cdots, \overline{p_j}\right)$, the total uncertainty in R is given by Equations (1) and (2):

$$U_r = \sqrt{\left(\theta_1 * u_1\right)^2 + \left(\theta_2 * u_2\right)^2 +, \cdots, + \left(\theta_j * u_j\right)^2} \tag{1}$$

$$\theta_j = \frac{\partial R}{\partial p_i} \tag{2}$$

where u_j is the uncertainty in the quantity $\overline{p_j}$ and θ_j is the sensitivity coefficient of f with respect to $\overline{p_j}$. Total experiment uncertainty for both measured and calculated parameters are ±1.06% and ±1.7% respectively.

3. Results and Discussion

A series of experiments were carried out for three different compression ratios of (9.5:1, 10:1, 10.5:1) at a constant speed of 2500 rpm and wide throttle open at

Table 2. Uncertainty of accuracy of measurements.

Parameters	Range	Accuracy	Uncertainties
Speed	0 - 12,000 RPM	±1	±0.0001%
Torque	0 - 95 N·m	±0.25	±0.26%
Fuel consumption	0.03 - 20 l/h	±0.1	±0.5%
Temperature	0˚C - 1000˚C	±1	±0.1%
Pressure sensor	0 - 250 bar	±0.5	±0.2%

Table 3. Uncertainty of calculated results.

Parameters	Uncertainties
THC	±0.3%
BSFC	±0.1%
bp	±0.7%
CO	±0.1%
NO_x	±0.2%
BTE	±0.3%

stoichiometric mixture using a single cylinder research engine. The engine was operated at various spark timing and varied from 20˚CA BTDC until knock is detected in incremental steps of 2˚CA. As can be noticed in investigation results, in some cases, the engine could not be operated at retarded and/or advanced ST; this is due to that fuel with higher octane number enhanced knock tolerance and allowed wider range of operation at high compression ratio and advanced spark timing that allow more favorable variables. As a result, fewer measurements are obtained for the case of high compression ratio and low octane number. Furthermore, in order to distinct the effect of compression ratio on the engine performance and exhaust emissions, the optimal spark timing of 32˚CA BTDC was selected. The fuel delivery system is port injection with injection pressure of 2.3 bar and injection starting (SOI) at 360˚ crank angle BTDC. Thermal efficiency, bp, BSFC, in-cylinder pressure, heat release, and exhaust emissions were obtained and analyzed. In the case of changing testing fuel, adequate time of operation was allowed before taking measurements to make sure that the fuel utilized beforehand is removed from the fuel system.

3.1. Brake Power

The engine brake power is defined as the actual engine output power measured at the output shaft. Engine brake power for RON91 and RON95 fuels at three compression ratios (9.5:1, 10:1, 10.5:1) with different spark timing are shown in Figure 5 and Figure 6, respectively. It can be seen that the brake power increases when spark timing increases for both fuels. While as compression ratio increases, the outcomes are different related to different RONs. With RON95, the brake power increases as compression ratio increases. These findings are in agreement with Costa *et al.* [9] and Aina *et al.* [12]. At the peak power for both

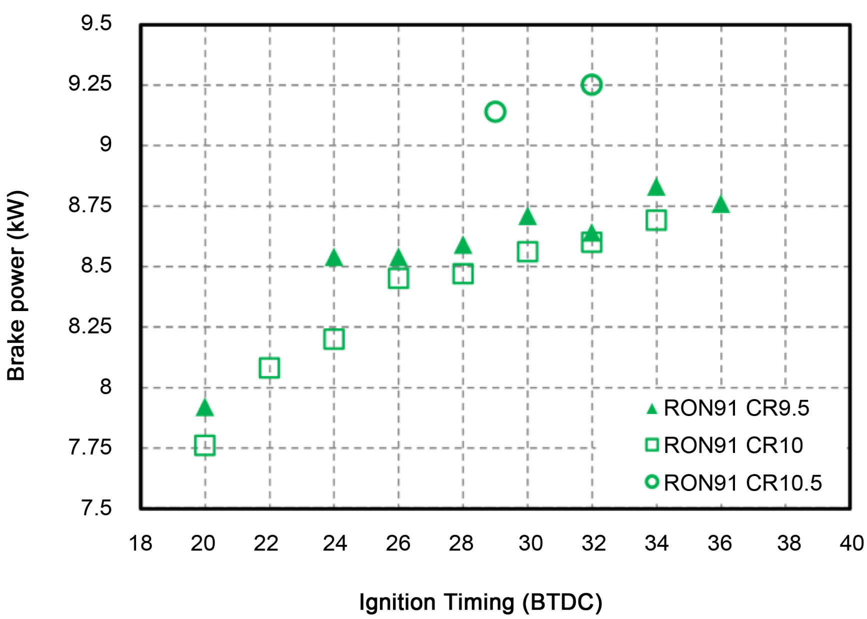

Figure 5. Engine brake power with RON91.

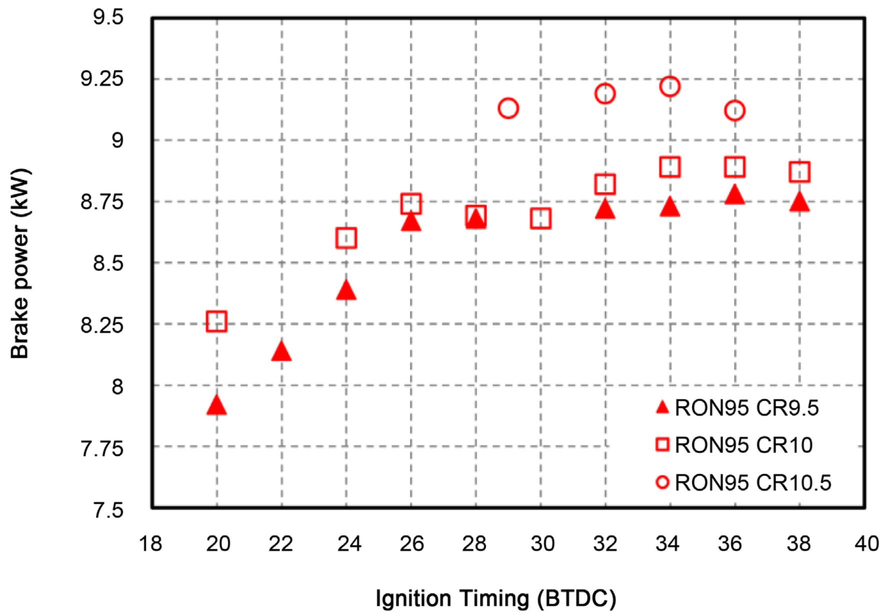

Figure 6. Engine brake power with RON95.

fuels, the RON91 showed better performance at CR9.5 by 1.14% compared to CR10. Unlike RON95 in which the higher compression ratio case CR10 showed improved performance by 1.2%. The advantage of RON95 is obvious in terms of brake power with CR10 and CR10.5. This can be explained by the fact that auto-ignition resistance is higher while having comparable heating value to RON91. The peak brake power achieved by RON91 was 9.25 kW at 32˚CA BTDC ST and CR10.5. While with RON95 peak power was 9.2 kW at 34˚CA BTDC ST and CR10.5. For both RONs, maximum power were achieved with the highest compression ratio operated CR10.5.

3.2. Thermal Efficiency

Thermal efficiency is described as the ratio of the brake power to the fuel thermal input. It is used to evaluate the conversion efficiency of the fuel thermal energy to mechanical energy. Figure 7 shows the thermal efficiency achieved by the engine of RON91 fuel and Figure 8 shows the thermal efficiency of RON95 fuel at three compression ratios (9.5:1, 10:1, 10.5:1) with different spark timing. Using RON91, thermal efficiency increased with increasing compression ratio except at 30˚CA BTDC for compression ratios of 9.5 and 10. The thermal efficiency increased as advancing the spark timing for CR10 and CR10.5. While at CR9.5, thermal efficiency increased as spark timing increases except between 24 and 28˚CA BTDCST. The maximum efficiency was obtained as 28.1% with CR10.5 at 32˚CA BTDCST, while the minimum was obtained as 21.5% with CR9.5 at 20˚CA BTDCST. In the case of RON95, thermal efficiency behavior showed significant variation in terms of relation between compression ratio and spark timing. When spark timing is less than 30˚CA BTDC, thermal efficiency with CR10 is higher than of CR9.5, as expected. However, as advancing the spark

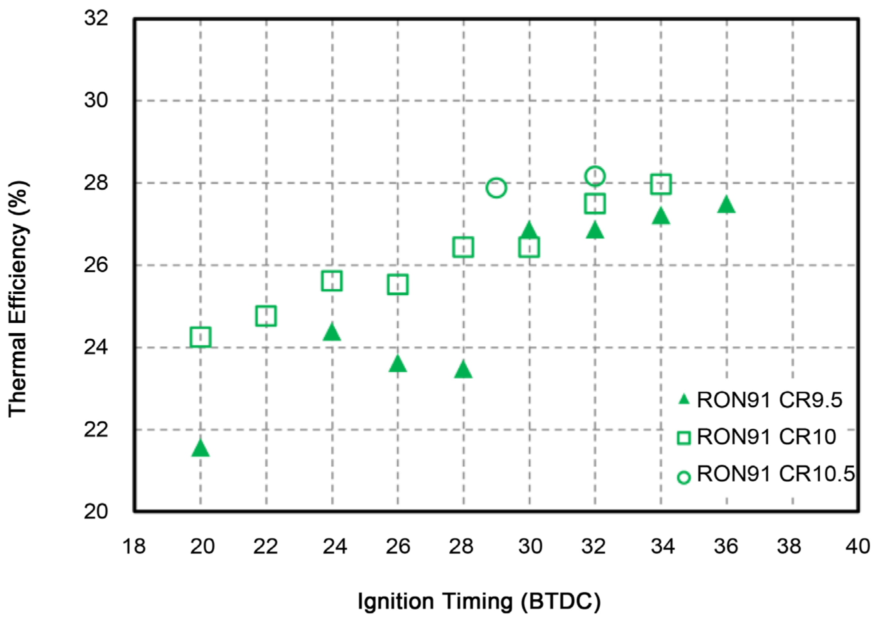

Figure 7. Engine thermal efficiency with RON91.

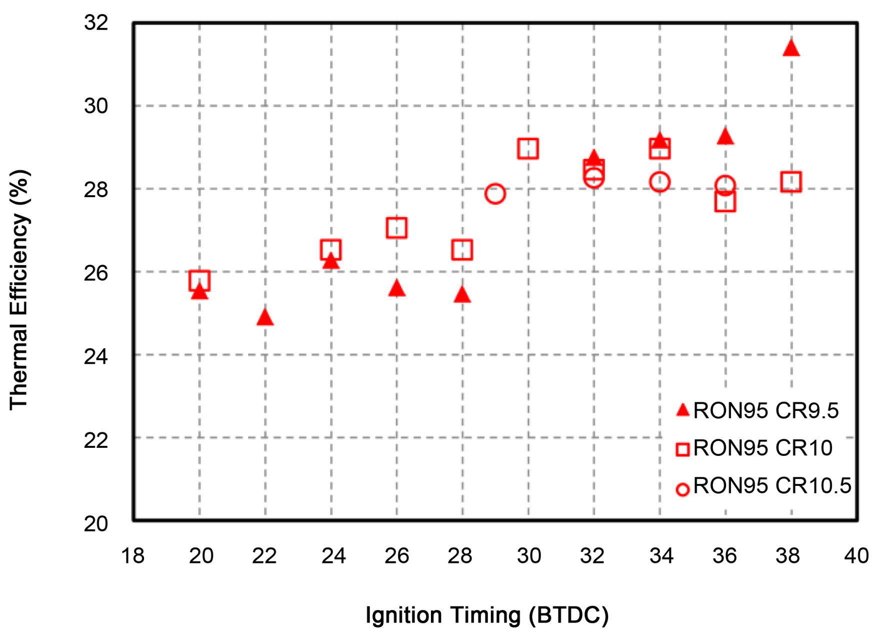

Figure 8. Engine thermal efficiency with RON95.

timing, CR9.5 showed superiority to CR10 and even CR10.5. There were not significant changes in the thermal efficiency of different compression ratios at 32˚CA BTDC ST.

3.3. Brake Specific Fuel Consumption

The brake specific fuel consumption is described as the rate of fuel consumed to the engine power generated. Figure 9 and Figure 10 show the brake specific fuel consumption of RON91 and RON95 fuels at three compression ratios (9.5:1,

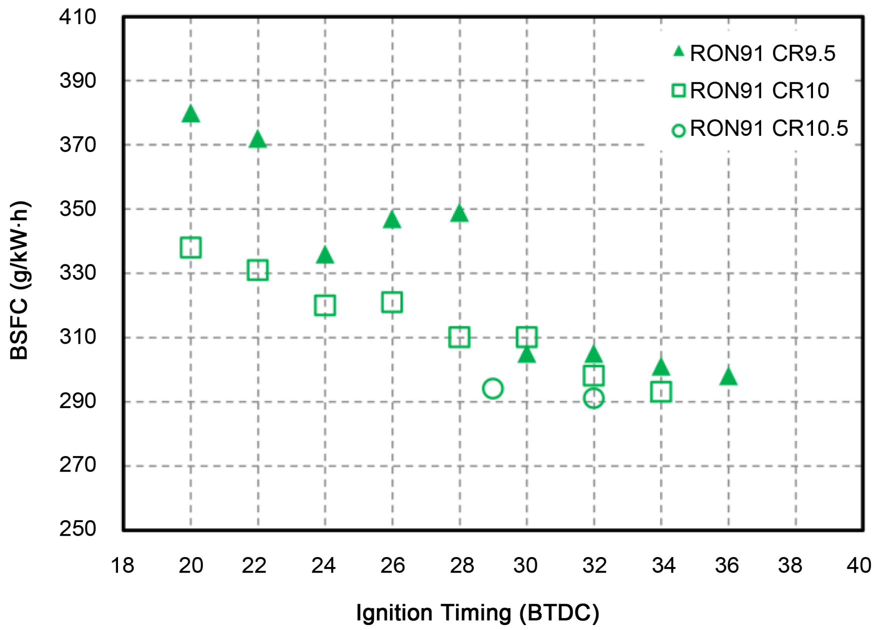

Figure 9. Engine brake specific fuel consumption with RON91.

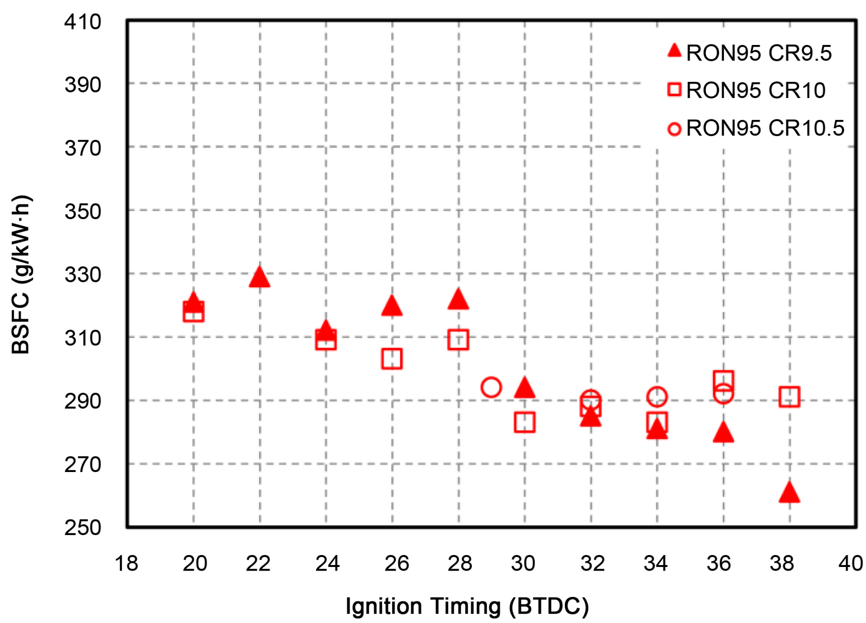

Figure 10. Engine brake specific fuel consumption with RON95.

10:1, 10.5:1) with different spark timing. Basically, BSFC is the inverse of thermal efficiency. The observations show that with RON91 fuel, the minimum fuel consumption rate at 290 g/kWh was achieved with CR10.5 with spark advance of 32˚CA BTDC. As compression ratio reduced to 10 and 9.5, BSFC increases especially at the more retarded ignition. The best BSFC for all compression ratios were achieved at the very advanced ignition with values of 290, 294 and 298 g/kWh respectively for CR10.5, CR10 and CR9. These results are expected and consistent with the results of other studies including Costa *et al.* [9] and Aina *et* *al*. [12]. Further reduction of fuel consumption was achieved when using higher RON fuel. The RON95 acquired 10% to 13% lower fuel consumption relative to RON91. However, in the case of RON95, even though at more retarded ignitions, CR9.5 showed worst fuel consumption, this trend reversed as spark advance progress. The minimum consumption of 260 g/kWh was found with CR9.5 at 38˚CA BTDCST. This value is 8% and 11% less than the minimum BSFC with CR10 and CR10.5, respectively.

3.4. Emissions

Three toxic engine emission species that need to be controlled were measured in this investigation including; carbon monoxide, CO, oxides of nitrogen, NO_x and total hydrocarbon, THC. Figure 11 and Figure 12 show CO emissions ofRON91 and RON95 fuels for the three compression ratios and different spark timing. In the case of RON91 fuel, CO emissions increased as compression ratio increases. However, it was observed that CO emissions decreased as spark timing advanced with CR9.5 and CR10 up to 28˚CA BTDCST. The behaviour is different with RON95 fuel in which CO emissions first decreased with spark advancing up to 30˚CA - 34˚CA BTDC and then increased with further advancement of ignition. At 32˚CA BTDC ST, CO emissions increased as compression ratio increased for both fuels and the advantage of RON95 over RON91 in reducing CO emissions is obvious with 12% less CO produced at higher compression ratios. Generally, for greater spark timing, combustion time is extended thus allowing better completion resulting in lower CO emissions. Additionally, with lower compression

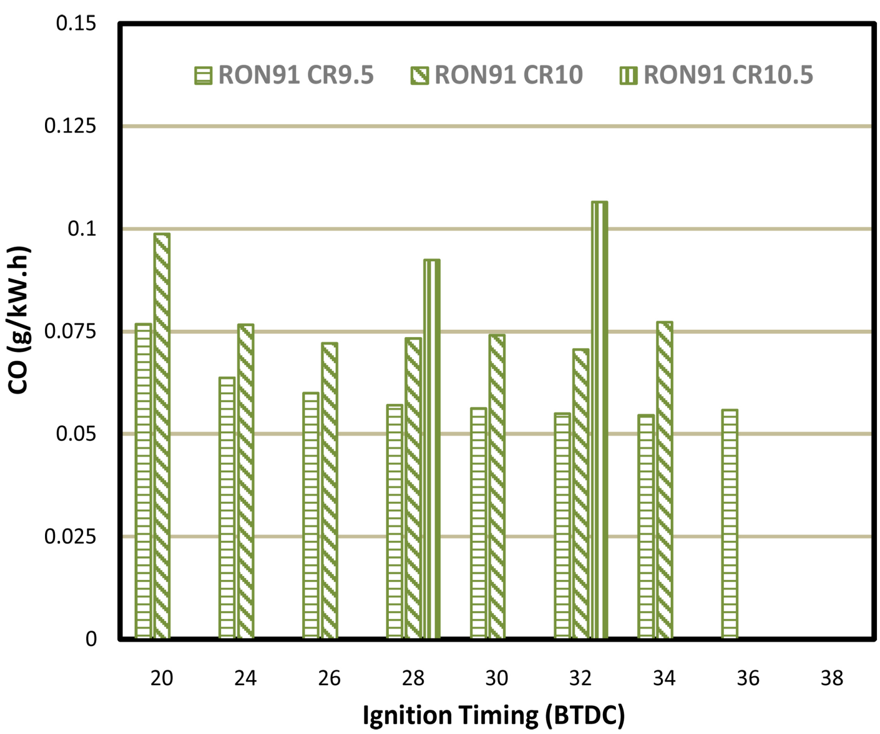

Figure 11. Carbon monoxide emissions with RON91.

ratio, peak cylinder temperature is reduced which can lead to less probability to dissociation of CO_2 to CO.

Figure 13 and Figure 14 show NO_x emissions of RON91 and RON95 fuels respectively, for the three compression ratios and different spark timing. An

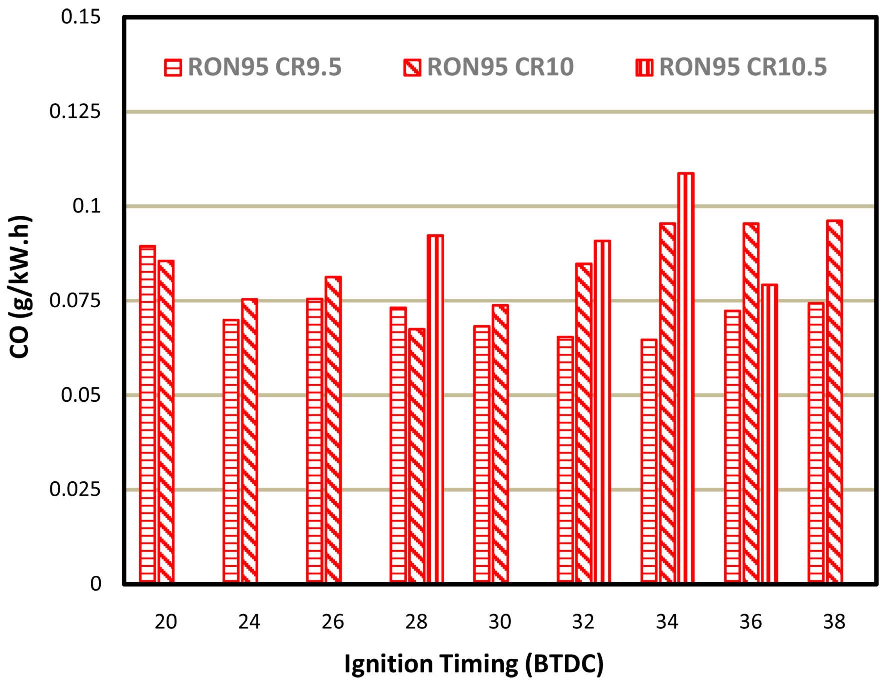

Figure 12. Carbon monoxide emissions with RON95.

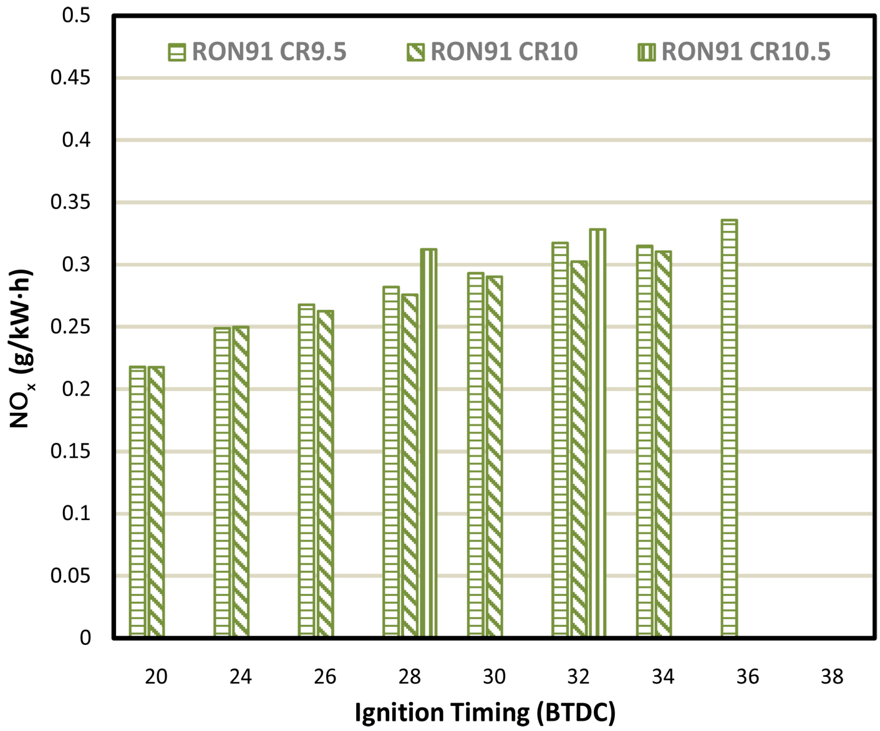

Figure 13. NO_x emissions with RON91.

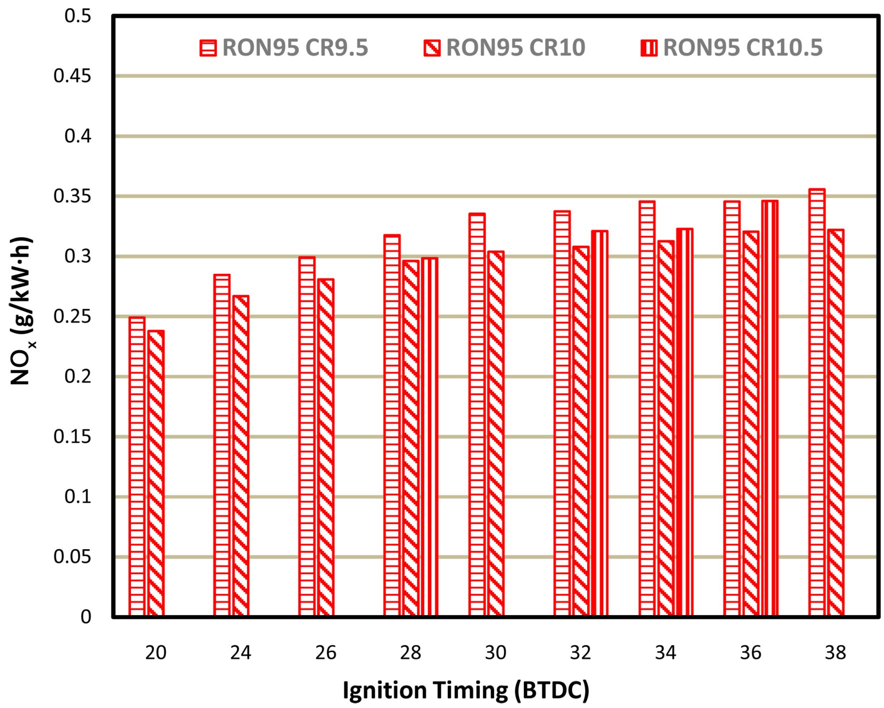

Figure 14. NO_x emissions with RON95.

overall observation showed that there is little variation of NO_x emissions with respect to compression ratio. For both fuels, when the compression ratio increased from CR9.5 to CR10, a slight decrease of NO_x emissions was observed. However, NO_x emissions increased with ignition advancement. RON91 offered better NO_x level with 6% - 15% less than those of RON95 over the whole range. As NO_x emissions are proportionate to combustion temperature, it was clear that advancing ignition caused peak cylinder temperature to rise and therefore more NO_x emissions.

Figure 15 and Figure 16 show the variations of total hydrocarbon emissions (THC) concerning compression ratio at different spark timing for RON91 and RON95 fuels respectively. There was not a clear trend with respect to the effect of compression ratio on THC. In the case of RON91, THC emissions increased as the compression ratio increases from CR9.5 to CR10 but the trend is reversed at CR10.5. While with RON95, THC emissions deceased as the compression ratio increases between 24˚CA BTDC to 30˚CA BTDC ST. There was not significant discrepancy at advanced ST for all compression ratios with RON95. It was noticed that higher THC emissions is produced from RON91 compared to RON95 at CR10 and CR10.5 at advanced spark timing, while RON95 produces higher THC emissions than RON91 at CR9.5. On average, maximum THC emission was found with CR10 while the minimum was with CR9.5. On the other hand, with RON95, the maximum and minimum THC emissions were found with CR9.5. In both minimums, the spark timing was 34˚CA BTDC and 36˚CA BTDC which coincide with the higher power regions. These indicates that CR9.5 is advantageous over other CR in terms of minimizing THC emissions

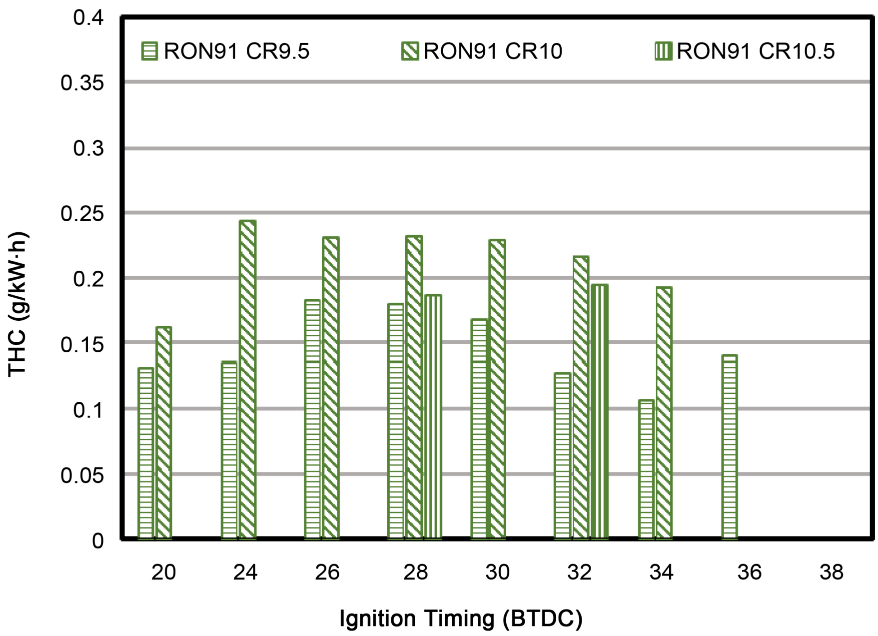

Figure 15. Total hydrocarbon emissions with RON91.

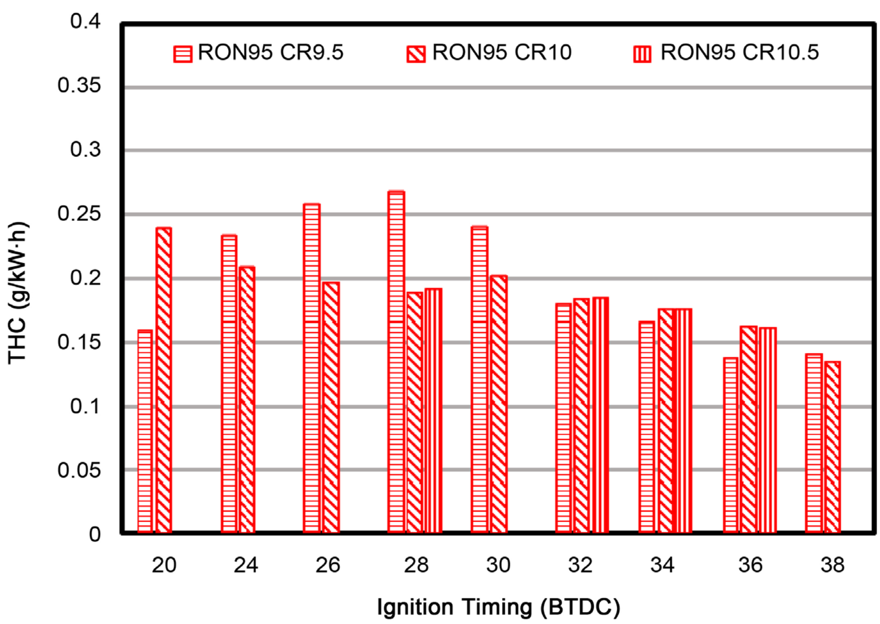

Figure 16. Total hydrocarbon emissions with RON95.

when the optimum spark timings are reached. It can also be seen that the average and the minimum THC with RON91 are lower than that of RON95 by around 15%.

3.5. Combustion Analysis

The in-cylinder pressure and heat release rate (ROHR) behaviours are discussed. Figure 17 shows the cylinder pressure curves for both fuels (RON91 and RON95) of several compression ratios at ST 32˚CA BTDC. It can be seen that

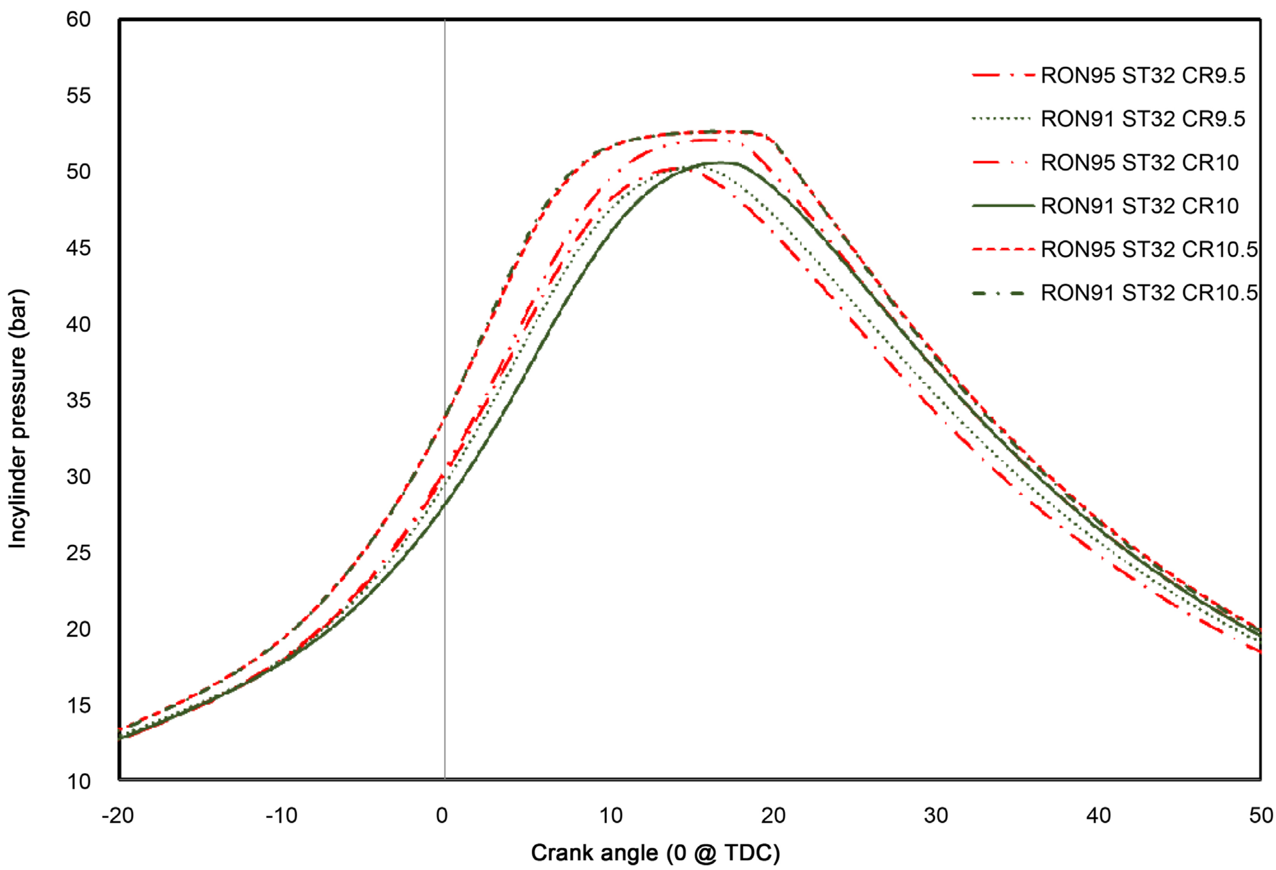

Figure 17. In-cylinder pressure of fuels (RON91 and RON95) at 32˚ crank angle BTDC ST for varied compression ratios.

for both fuels, the peak pressure increases as the compression ratio rises. Furthermore, RON95 fuel showed higher in-cylinder pressure than RON91 at CR10, but when the compression ratio increased to CR10.5; both fuels displayed almost the same in-cylinder pressure values. Table 4 shows the peak pressure locations at different compression ratios for the two fuels. It can be seen that RON91 make lower and delayed peak pressure happening, in particular at CR10.

ROHR of RON91 and RON95 fuels is calculated from the pressure data recorded by the evaluation software from Lotus-Engine based on the single Weibe heat release function. Figure 18 shows ROHR plots and Table 5 demonstrates the peak of ROHR values and events, which are the parameter of interest. Generally, higher peak ROHR at earlier occurrence of RON95 was observed with increasing compression ratio compared to RON91. With CR10.5, two peaks in the ROHR plot are noticeable for two of heat release phases representing the premixed and diffusion combustion phases respectively. RON95 displayed a more noticeable diffusion combustion phase than RON91. This is attributed to the RON95 has lower calorific value per unit volume of fuel.

Cycle variability in the combustion process is a considerable limitation on engine operation. These instabilities were noticed and identified as a fundamental combustion tricky that cause fluctuations in many engine parameters and accordingly the power output of SI engines [17]. Many researches stated that cyclic

Table 4. Cylinder pressure peak values and event timings over all compression ratios at 32˚CA BTDC ST.

Compression ratios	RON 91		RON 95	
	Event timing (˚CA ATDC)	Peak pressure (bar)	Event timing (˚CA ATDC)	Peak pressure (bar)
9.5	15.3	50.36	14.3	50.245
10	17.5	50.585	16.1	52.123
10.5	17.4	52.679	17.4	52.647

Table 5. ROHR peak values and even timing over all compression ratios at 32˚CA.

Compression ratios	RON 91		RON 95	
	Event timing (˚CA ATDC)	ROHR peak (kJ/m^3 ˚CA)	Event timing (˚CA ATDC)	Peak of ROHR (kJ/m^3 ˚CA)
9.5	7	116.73	7	117.44
10	10	123.02	6	125.89
10.5	4	124.48	2	126.5

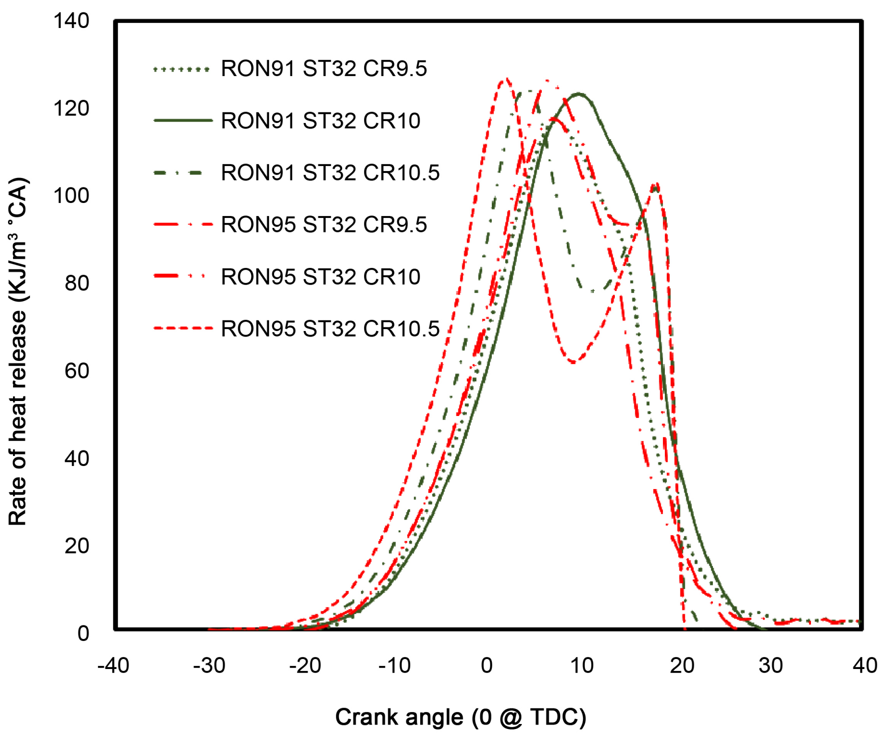

Figure 18. ROHR of RON91 and RON95 at 32˚CA BTDCST for different CRs.

variability in gasoline engine affects the engine fuel economy and it decrease the mean effective pressure by as much as 20% [18]. The coefficient of variation (COV) of the indicated mean effective pressure (IMEP) quantifies the indicated mean effective pressure fluctuation from cycle to cycle. It is a favoured variable for measuring combustion stability in SI engines. The combustion process of SI engine is considered stable when COV_{IMEP} is less than 10%. The COV_{IMEP} for 100 successive engine cycles was calculated as in Equation (3) at each operating condition [19]. **Table 6** demonstrates that COV_{IMEP} decreased with advancement in spark timing and rise in compression ratio from CR9.5 to CR10 for both fuels. However, COV_{IMEP} increased when increasing compression ratio from CR10 to CR10.5. It can be seen that the COV_{IMEP} of the RON95 fuel is lower than that of RON91 fuel. Therefore, the engine seems to be running more stable with RON95 fuel.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP} \times 100 \quad (3)$$

Table 6. Variation of the COV_{IMEP} at compression ratios and spark timing for both fuels.

Spark timing (˚CA BTDC)	COV_{IMEP} (%)					
	RON91			RON95		
	CR9.5	CR10	CR10.5	CR9.5	CR10	CR10.5
20	2.760	3.610		2.670	2.520	
24	2.460	2.360		1.880	1.620	
26	1.730	1.780		1.700	1.530	
28	1.290	1.600	3.890	1.390	1.120	1.550
30	1.050	1.170		0.947	0.901	
32	0.971	0.926	0.880	0.744	0.702	0.896
34	1.120	0.737		0.713	0.720	0.895
36	0.957			0.872	0.741	1.080
38				1.020	0.883	

4. Conclusions

In this paper, an experimental investigation was performed to address the effect of fuel formulation at variable compression ratio over different spark timing on the performance and exhaust emissions of a spark ignition engine. Two available gasoline grades from Saudi Arabian market (RON91 and RON95) were investigated. Experimental study was operated at stoichiometric condition using a single cylinder engine under three compression ratios of 9.5:1, 10:1, 10.5:1 and varying spark timings from 20˚CA BTDC with increment of 2˚CA till knocking is detected at an engine speed of 2500 rpm and wide throttle open.

The experimental results shows that optimal brake power can be achieved at higher compression ratios for both grade of fuels. Maximum power was achieved at compression ratio of 10.5 for RON91 and RON95 as 9.25 kW and 9.2 kW, respectively. Compression ratio had a significant effect on the engine fuel consumption. Increasing the compression ratio has generally decreased the BSFC for both fuels up till the optimal spark timing. The best BSFCs for all compression ratios were obtained at the leading ignition with readings of 290, 294 and 298 g/kWh, respectively for CR10.5, CR10 and CR9.5, of RON91. In terms of exhaust emissions, the compression ratio effect was insignificant regarding NO_x and THC emissions. Nevertheless, the RON91 produced lower NO_x and HC rates than RON95. However, unlike NO_x and THC, the compression ratio effect on CO emission was significant. Increasing the compression ratio would increase the CO emissions due to the rise in the peak cylinder temperature. RON95 gave lower CO emissions than RON91 in particular at higher compression ratios with 12% less CO emission produced at compression ratio of 10.5. Combustion performance shows higher peak ROHR at earlier occurrence of the RON95 fuel with increasing compression ratio compared to RON91 fuel.

Based on all of the results, increasing the fuel octane number improved the engine power and BSFC with the increased compression ratio. Increased compression ratio augmented mixture burning for both fuels besides increased octane number allowed more resistance to knocking. Therefore, these consequences improved the engine performance and reduced fuel consumption while there was a trade-off among exhaust emissions of NO_x, CO and THC.

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Abbreviation

ATDC= After top dead center
AFR= Air-fuel ratio
BSFC= Brake specific fuel consumption
BTDC= Before top dead center
bp= Brake power
CR= Compression ratio
CO= Carbon monoxide
HCCI= Homogeneous charge compression ignition
IC= Internal combustion
NO_x= Nitrogen oxides
PI= Port injection
RON= Research octane number
ROHR= Rate of heat release
ST= Spark timing
SOI= Start of injection
SCRE= Single cylinder research engine
THC= Total hydrocarbon
WOT= Wide open throttle