Influence of compression ratio on performance and emissions of natural gas fuelled SI engine

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ABSTRACT

Recent research of world energy consumption shows that transport is the main source of harmful exhaust gas emissions. Because of the environmental concerns, automotive industry is exploring alternative, more acceptable fuels. Compressed natural gas is one of the "cleanest" fossil fuel and can be used both in spark ignited and compression ignited engines with a number of benefits. Since natural gas has much higher octane rating than gasoline it is expected that higher compression ratios can be used. The goal of the research is to determine the change of performance of engine with the increase of compression ratio while keeping the exhaust emissions on the acceptable level and avoiding knock combustion. Methane with known composition from a pressure cylinder is used instead of natural gas and the results are comprised of indicating results (in-cylinder and intake pressure in a crank angle space), emissions, temperatures and mass flows on various intake and exhaust positions.

KEYWORDS

Natural Gas, CNG Engine, SI Engine, Compression ratio, Methane

INTRODUCTION

Recent research in the field of alternative energy solutions shows a great potential of the compressed natural gas (CNG) as an alternative fuel especially in road transportation. Besides being the alternative fuel the use of CNG can reduce harmful exhaust emission in comparison to conventional diesel and gasoline engines [1]. Although CNG is also a fossil fuel it produces significantly less CO_2 compared to the gasoline or diesel fuel (up to 25%) [2]. Great advantage of CNG in comparison to gasoline is the high octane number (ON = 120) which could allow the use of natural gas in engines with high compression ratio. CNG supply system can be easily installed on intake pipe of conventional gasoline spark ignited (SI) engine and can be used instead of gasoline. The available research that showed experimental results of using natural gas in conventional SI engines presented some advantages of CNG compared to the conventional gasoline fuel, but it also presented some disadvantages. In [3] authors show that the SI engine fuelled with CNG produces 18.5% less power compared to the gasoline due to the lower volumetric efficiency, which is caused by using of fuel with lower molar mass.

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Lower molar mass finally leads to the lower heat release. Comparison between using CNG and gasoline in retrofitted gasoline vehicle (Compression ratio = 9.2) is given in [4]. Authors showed that engine fuelled by CNG produces lower power, achieves lower brake specific fuel consumption (BSFC), emits less CO and HC emissions, but 33 % more NO_X emission compared to the engine fuelled by gasoline. Similar results are obtained with SI engine that had compression ratio of 9.5 in [5]. An effective way to reduce NO_X emission and improve thermal efficiency is lean burn [6, 7], but ultra-lean operation can result with misfire and unstable operation [7]. The increase of thermal efficiency is also possible by the increase of compression ratio or by applying turbocharging/supercharging [7].
Commercial transportation engines with higher compression ratio are mostly compression ignited (CI) engines. Natural gas cannot be used as a single fuel in CI engines because of the high octane number (low cetane number). In CI engines natural gas is usually used with the addition of pilot fuel that has high cetane number (e.g. diesel fuel) which then initiates combustion. This type of combustion is called dual fuel [8] and is not a part of the research presented in this paper.
The overview of the previous research indicates a lack of available experimental results for CNG SI engines with compression ratio over 15. Available literature associated to using natural gas in SI engines shows experimental results for compression ratios 8.5, 10.5 and 12.5 in [9], for compression ratio 11.5 in heavy-duty Euro VI engine in [10], and for compression ratio 14 in [11, 12]. Researchers in [13, 14, 15] used similar compression ratios, while researches in [16] compared four compression ratios between 8 and 14.7. This researchers showed trends of performances and emissions of natural gas fuelled SI engine with increasing CR, but behaviour of the engine at higher compression ratio is quite unknown. Therefore, the experimental research of behaviour of CNG SI engine at higher compression ratio is the main motivation for this work. The main objectives of the research are to investigate behaviour of SI engine fuelled by natural gas at two different compression ratios and compare achieved performances and exhaust emissions. Research is performed at one compression ratio similar to conventional gasoline engine (CR = 12) and one compression ratio similar to conventional diesel engine (CR = 16). For better insight into the behaviour of the engine, two lambda values are used ($\lambda = 1.2$ and $\lambda = 1.4$). These values were chosen because the literature review showed that increase of lambda above 1.18 ($\phi < 0.65$) decreases NO_X emission, while lambda higher than 1.54 ($\phi < 0.65$) causes unstable engine operation in the natural gas fuelled single cylinder SI engine [17].

EXPERIMENTAL SETUP AND TEST CASES

The presented study was performed on an experimental setup located in the Laboratory of IC engines and motor vehicles at the Faculty of Mechanical Engineering and Naval Architecture in Zagreb that has been built by using Hatz 1D81 engine as a basis. Engine properties are given in Table 1.

Table 1 – Specifications of test engine

Item	Specification
Type	Single cylinder
Piston displacement	667 cm^3
Stroke	4-stroke
Bore	100 mm
Stroke	85 mm
Connecting rod length	127 mm
Compression ratio	Adjustable (original 20.5)

For the proposed research experimental engine is converted from CI diesel engine to SI engine operation. Instead of diesel fuel injector, spark plug is mounted into the cylinder head. Engine is also equipped with the high pressure sensor (AVL GH14DK) mounted on the cylinder head and low pressure sensor (AVL LP11DA) mounted in front of the intake valve. The change from original compression ratio (20.5) to compression ratio of 16 is obtained by addition of thicker cylinder head gasket, while the compression ratio of 12 is achieved by changing (machining) the height of the piston. For the preparation of the fuel mixture the gas injector is installed into the intake pipe, close to the intake valve. Instead of natural gas, pure methane (99.95 % vol.) from the pressure bottle is used as a fuel. Figure 1 shows schematic diagram of the experimental setup. As can be seen on Figure 1, experimental setup is not equipped with natural gas flow meter. Therefore, fuel mass flow is calculated from measured lambda value and measured air mass flow. Air mass flow is measured by flow meter TSI 2017L, while Lambda value and NOx emission are measured by ECM analyser NOx 5210T. CO exhaust emission is measured by BOSCH ETT8.55 analyser and THC emission is measured by heated THC analyzer Environment S.A. GRAPHITE 52M. Thermal efficiency and rate of heat release (RoHR) are calculated from measured values by using in-house developed code.

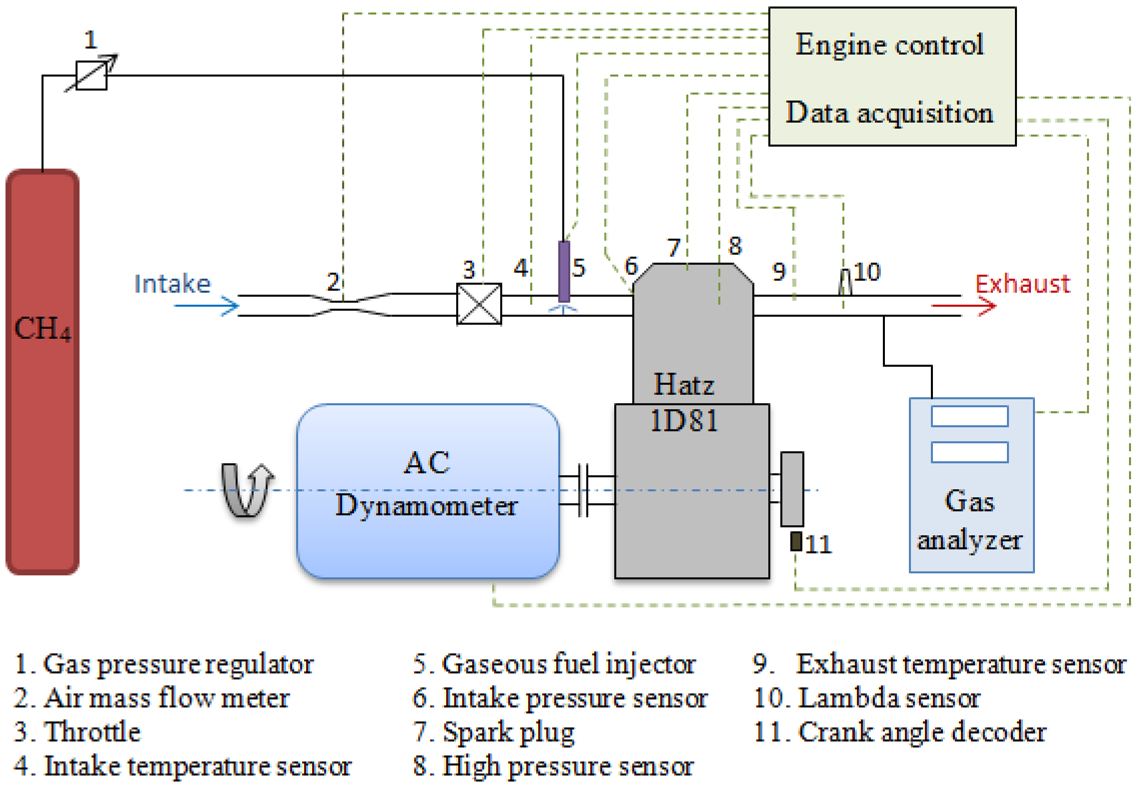


Figure 1 Schematic diagram of experimental setup

Engine test was performed at wide open throttle (WOT) position, at three different engine speeds (1200 min^{-1}, 1600 min^{-1}, 2000 min^{-1}) and at two different lambda values (λ = 1.2, λ = 1.4). To determine the optimal spark timing (ST), spark sweep was performed for every predefined operating point (compression ratio, engine speed and air fuel ratio). During measurements of all operating points the ambient conditions (temperature, ambient pressure, humidity, etc.) were similar. The summary of all test cases (Operating points) are shown inTable 2 and Table 3.

Table 2 – Test cases at compression ratio 12

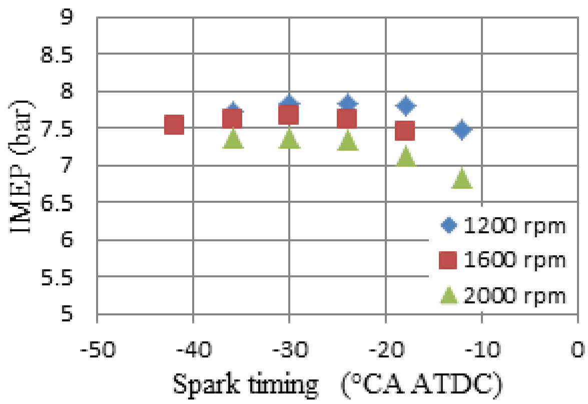
Operating point	CR	n (min^{-1})	p_{intake} (bar)	λ	Throttle
1.1 1.2	12	1200	1	1,2 1,4	100%
1.3 1.4	12	1600	1	1,2 1,4	100%
1.5 1.6	12	2000	1	1,2 1,4	100%

Table 3 – Test cases at compression ratio 16

Operating point	CR	n (min^{-1})	p_{intake} (bar)	λ	Throttle
2.1 2.2	16	1200	1	1,2 1,4	100%
2.3 2.4	16	1600	1	1,2 1,4	100%
2.5 2.6	16	2000	1	1,2 1,4	100%

RESULTS AND DISCUSSIONS

As describe in the previous section, a spark sweep was performed in order to determine the optimal combustion phasing at every operating point determined by engine speed, lambda values and compression ratio. Figure 2 shows the results of indicated mean effective pressure (IMEP) for λ = 1.2 and both compression ratios, while Figure 3 shows IMEP for λ = 1.4. The results show that higher compression ratio leads to higher IMEP at same lambda value and engine speed. The higher IMEP is obtained at lower engine speeds probably because of the larger volumetric efficiency and lower pumping losses. It can also be noticed that the influence of changes in spark timing on IMEP is larger at higher compression ratio (larger difference in IMEP for smaller change of ST). Optimal spark timing at CR = 12 and λ =1.2 is approximately -30 °CA ATDC while the increase of lambda value leads to shift of optimal spark timing towards more advanced values. At higher CR optimal spark timing is more retarded for same lambda value.



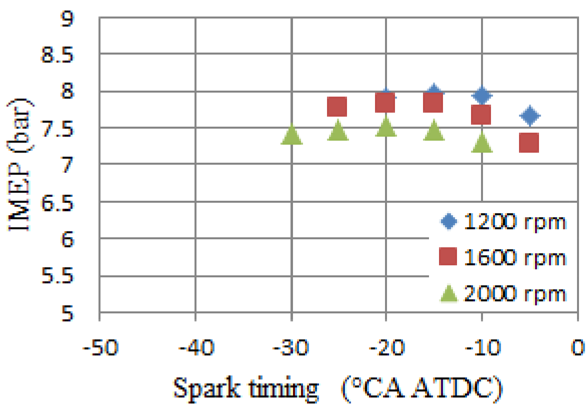


Figure 2 IMEP at λ = 1.2; CR = 12 (left) and CR = 16 (right)

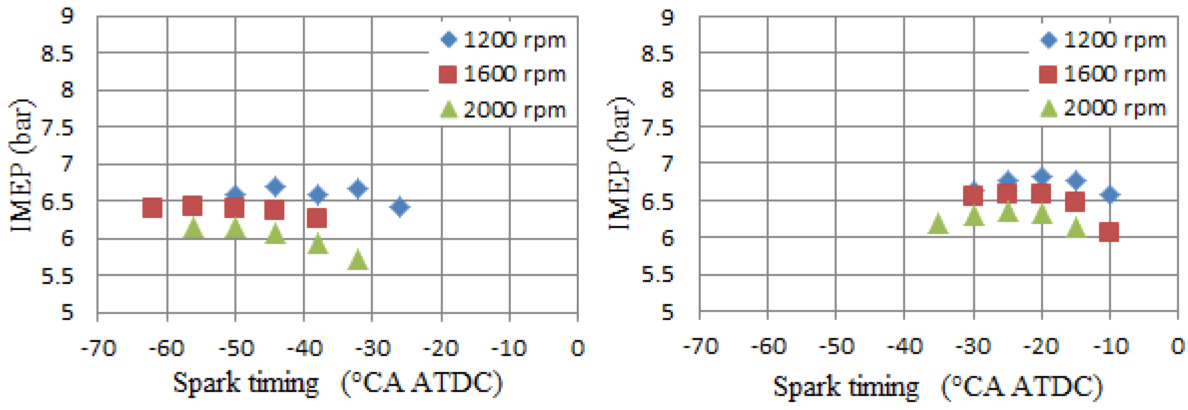


Figure 3 IMEP at λ = 1.4; CR = 12 (left) and CR = 16 (right)

Figure 4 and Figure 5 show the coefficient of variation (CoV) of IMEP for tested operating points. The results of CoV of IMEP show typical behaviour in which very early and very late spark timing result with increased CoV of IMEP with minimum value somewhere in between. At higher compression ratio in general CoV of IMEP is lower than at lower compression ratio for both lambda values. Also, the increase of lambda value increases CoV of IMEP. It has to be mentioned that in all cases the CoV of IMEP is below 5%.

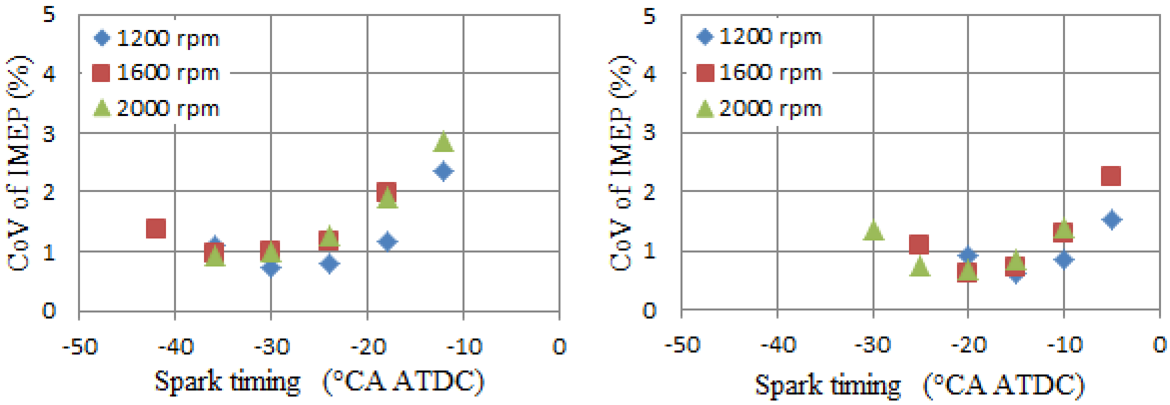


Figure 4 CoV of IMEP at λ = 1.2; CR = 12 (left) and CR = 16 (right)

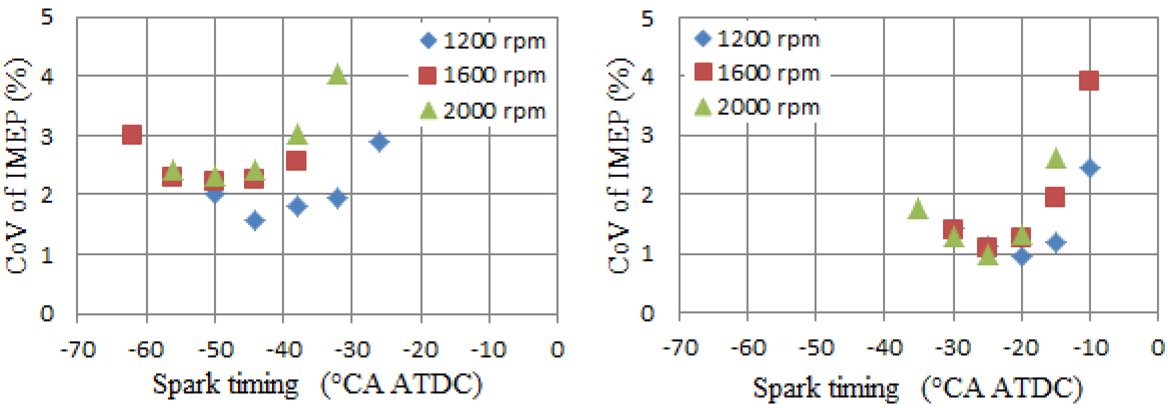


Figure 5 CoV of IMEP at λ = 1.4; CR = 12 (left) and CR = 16 (right)

The change of spark timing when changing CR and lambda is required because the changing conditions influence combustion duration on one hand and ignition delay on the other. Ignition delay for all cases is shown in Figure 6 and Figure 7. Ignition delay is calculated as a difference between CA5 and spark timing. As can be seen, advanced spark timing results with higher ignition delay. By comparison of compression ratios, ignition delay is lower at higher compression ratio. This is expected and caused by higher in-cylinder temperature at defined spark timing. But the difference in ignition delay at different compression ratios is more pronounced at larger lambda. Also, the increase of lambda value increases the ignition delay.

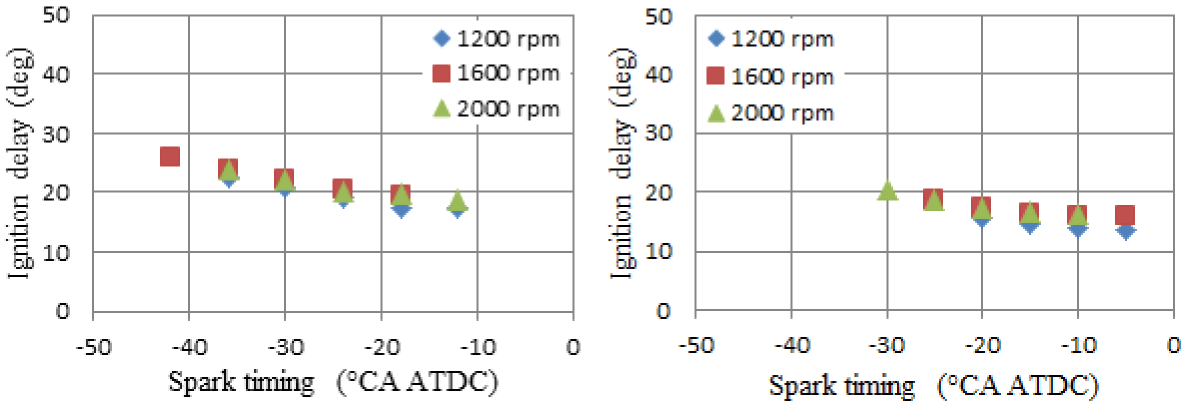


Figure 6 Ignition delay at λ = 1.2; CR = 12 (left) and CR = 16 (right)

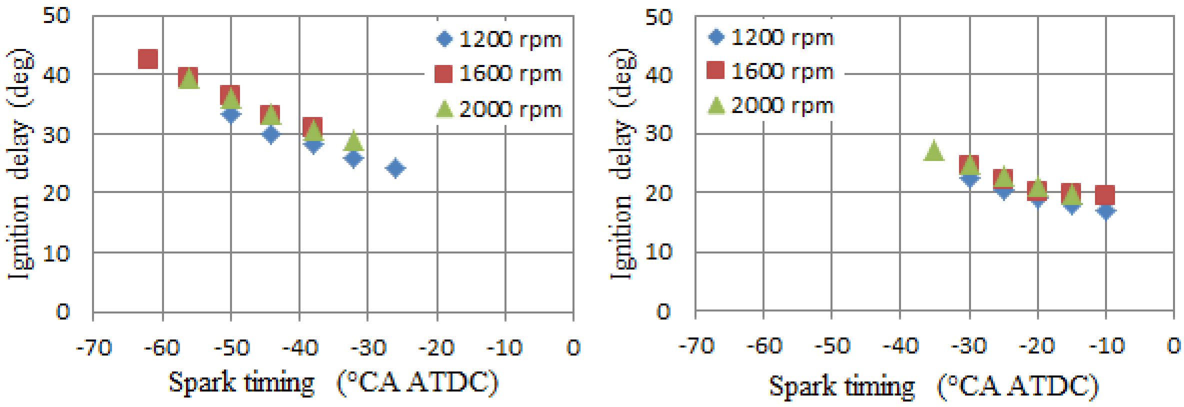


Figure 7 Ignition delay at λ = 1.4; CR = 12 (left) and CR = 16 (right)

With respect to combustion duration the results are shown in Figure 8 and Figure 9. The higher compression ratio decreases combustion duration for ~ 10 °CA at λ =1.2 and ~ 20 °CA at λ =1.4. Engine speed also has influence on combustion duration. Higher engine speed increases combustion duration in all test cases but its influence is most significant at lower CR and higher lambda (λ =1.4). By combining results of ignition delay and combustion duration one can notice that at higher lambda and lower CR the increase of combustion duration and ignition delay is so significant that very early spark timing is required to maintain combustion phasing. But with too early spark timing there is a risk of misfire as the spark discharge is at low temperature and pressure.

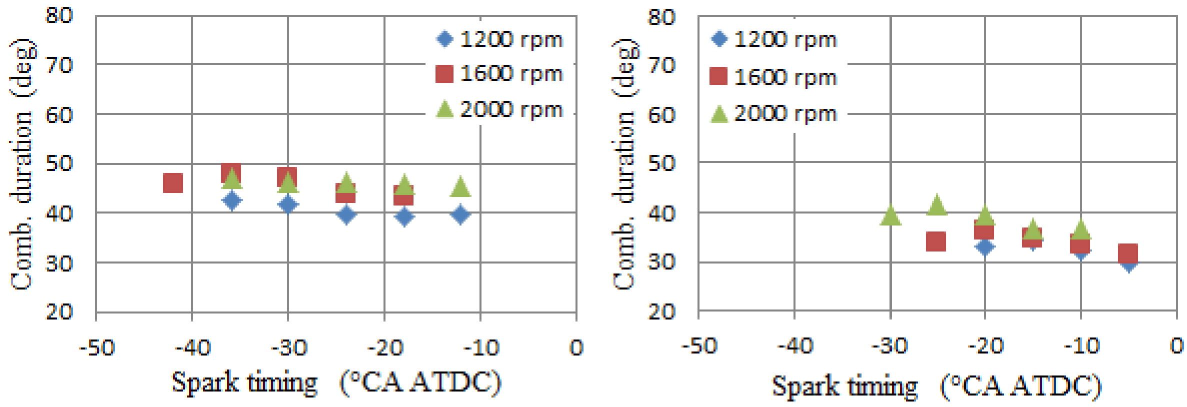


Figure 8 Combustion duration at λ = 1.2; CR = 12 (left) and CR = 16 (right)

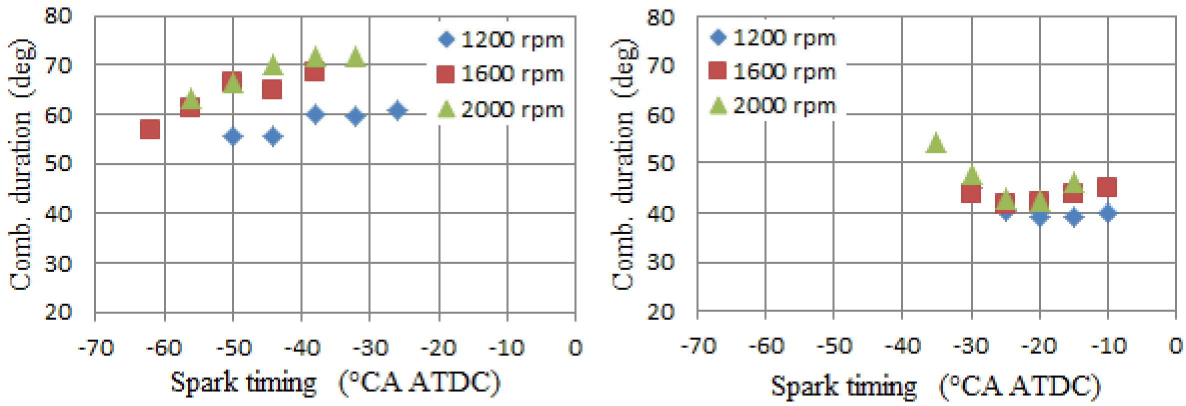


Figure 9 Combustion duration at λ = 1.4; CR = 12 (left) and CR = 16 (right)

The consequence of changes in combustion profile and CR is the change in engine efficiency. The indicated efficiency of all cases is shown in Figure 10 and Figure 11. Indicated efficiency is higher at higher compression ratio by about 3% points. Small drop in indicated efficiency is noticed with increase of engine speed which might be caused by lower volumetric efficiency of engine and higher pumping losses.

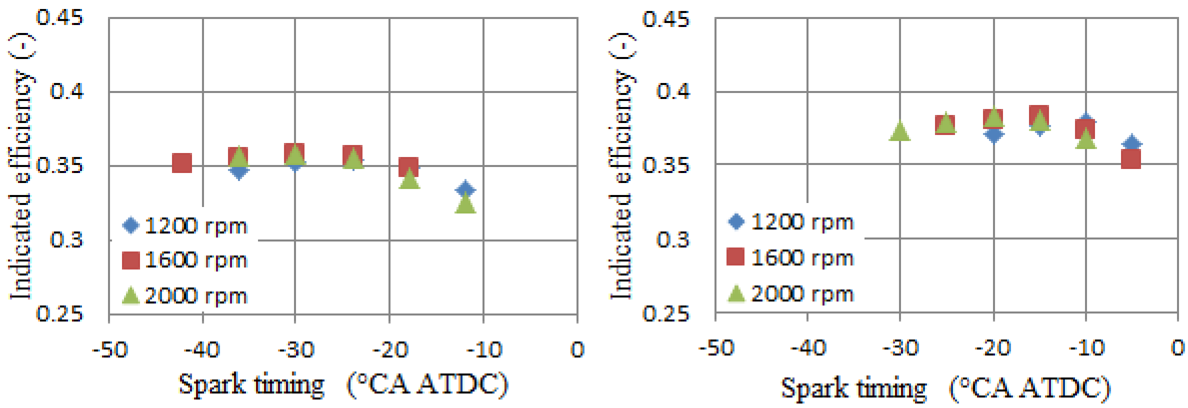


Figure 10 Indicated efficiency at λ = 1.2; CR = 12 (left) and CR = 16 (right)

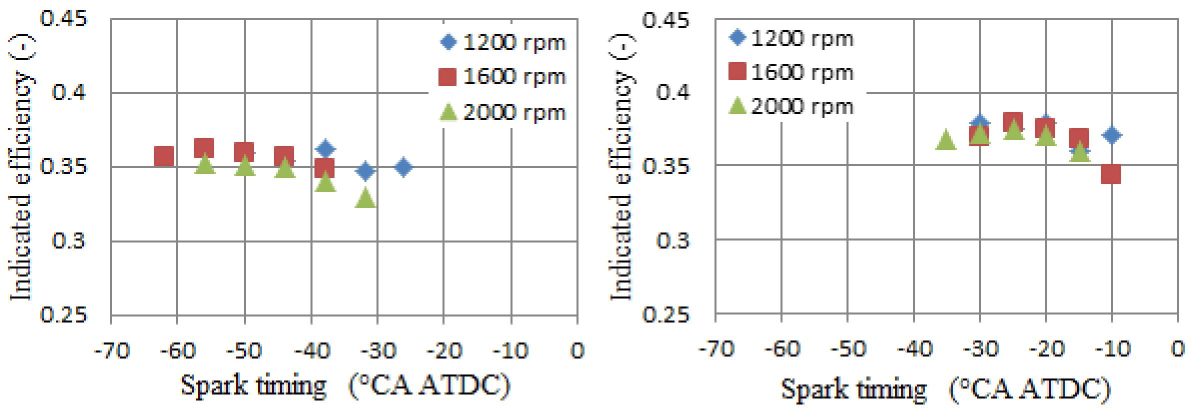


Figure 11 Indicated efficiency at λ = 1.4; CR = 12 (left) and CR = 16 (right)

Harmful exhaust emissions (THC, CO and NO_X) are shown in figures 12-17. In

Table 4 currently valid European Union (EU) standard of allowed exhaust emissions of heavy-duty engines are given for a reference. The regulation prescribes values for Steady-state test cycle (WHSC) only for CI engines while for Transient test cycle (WHTC) there are limits for Spark ignited engines (Positive ignition engine) and CI engines. However, for CI engines the limits values are similar for both cycles and therefore the values of WHTC (PI) can be used as a reference in this research although the tests performed here are steady state tests.

Table 4 - Heavy-duty European exhaust emissions standard - Euro VI [18]

	CO	THC	NMHC	CH_4	NO_X	NH_3	PM Mass	PM Number
	mg/kWh	mg/kWh	mg/kWh	mg/kWh	mg/kWh	ppm	mg/kWh	#/kWh
WHSC (CI)	1500	130	/	/	400	10	10	$8x10^{11}$
WHTC (CI)	4000	160	/	/	460	10	10	$6x10^{11}$
WHTC (PI)	4000	/	160	500	460	10	10	Not confirmed yet

Figure 12 and Figure 13 show measured values of THC (total hydrocarbons) in dependence of spark timing. As can be seen, all measured values are significantly higher than the limits shown in

Table 4 which means that some after treatment is required. The THC emission is higher at lower lambda which is caused by slower in-cylinder flame propagation and probably flames quenching. THC emission is higher at lower engine speed which is probably caused by positive valve overlap and fuel slip into exhaust. Effect of fuel slip in tested engine is previously investigated in [19] and there is shown significantly higher amount of fuel slip at

1000 and 1500 rpm compared to the higher engine speeds. At higher compression ratio THC emission is higher which is probably caused by higher influence of crevices where some amount of fuel remains unburned.

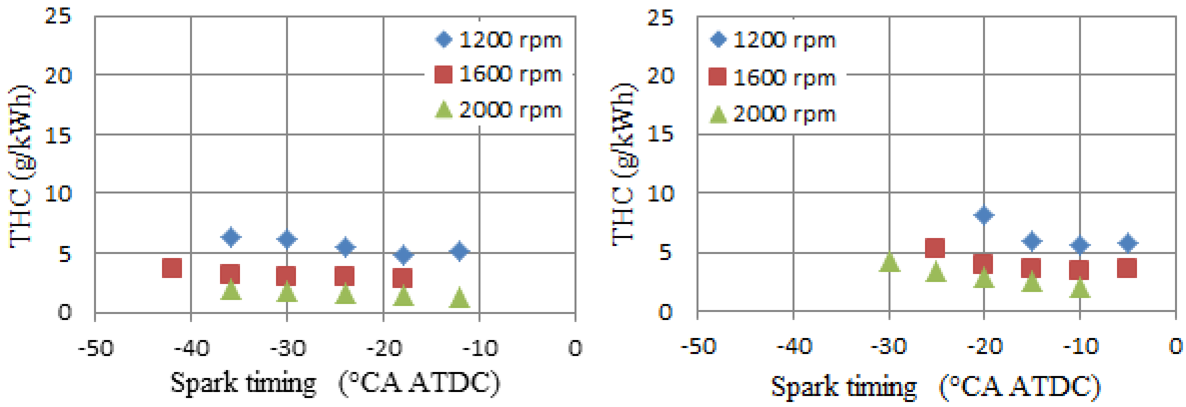


Figure 12 THC at λ = 1.2; CR = 12 (left) and CR = 16 (right)

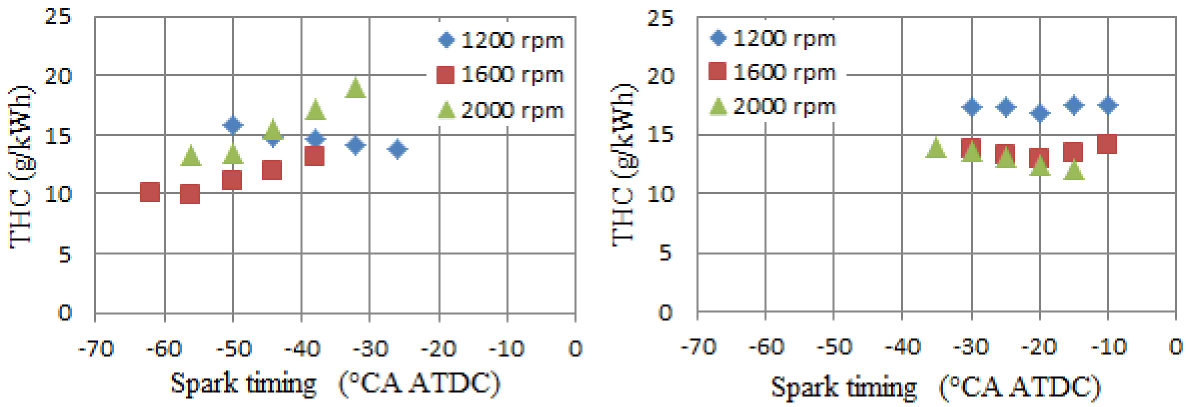


Figure 13 THC at λ = 1.4; CR = 12 (left) and CR = 16 (right)

Measured CO emission is presented in Figure 14 and Figure 15. Maximum emission of CO is at 1200 rpm and decreases with increase of engine speed at both of lambda values at CR = 12. As expected, higher compression ratio produces higher in-cylinder temperature and lower CO emission accordingly. The increase of lambda value resulted with the increase of CO emission which is in contradiction to some results shown in literature [7, 20]. The formation of CO is under the direct influences of bulk gas temperature and oxygen concentration. Since both lambda values represent a lean mixture there is enough oxygen for CO oxidation. Therefore the CO is influenced by temperature. This influence is two folded. With decreasing temperature the CO to CO_2 oxidation is reduced but also the formation of CO from higher hydrocarbons is also reduced. It seems that the temperature levels obtained at λ = 1.4 only slightly reduced the formation of CO from hydrocarbons, but significantly reduced the oxidation of CO to CO_2 and therefore the CO is increased.

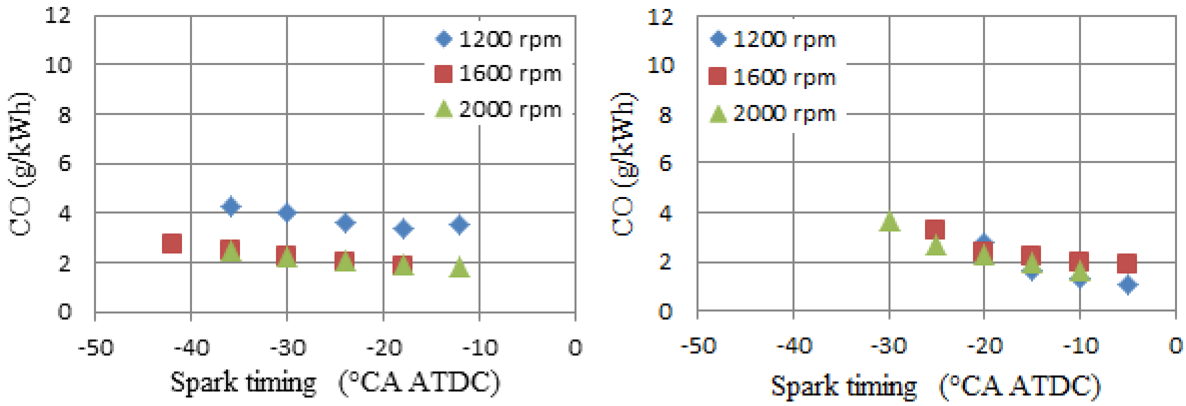


Figure 14 CO at λ = 1.2; CR = 12 (left) and CR = 16 (right)

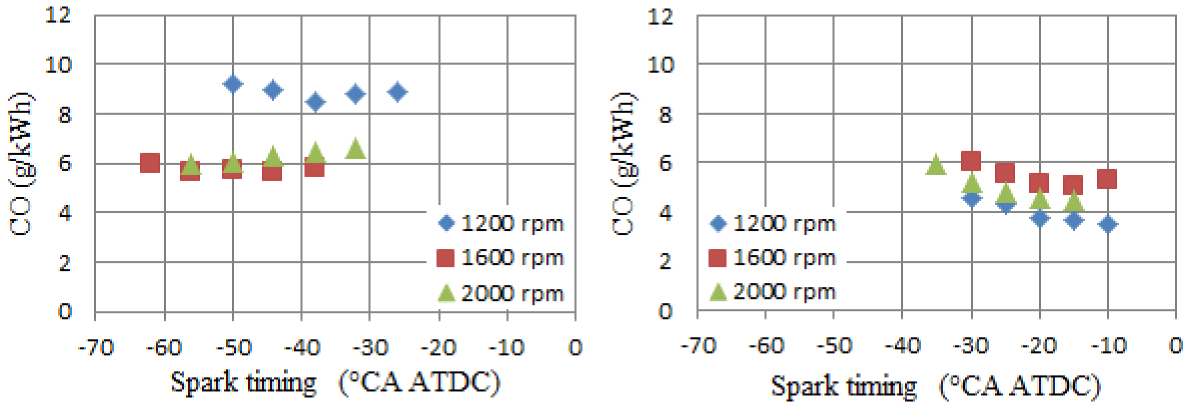


Figure 15 CO at λ = 1.4; CR = 12 (left) and CR = 16 (right)

The NO_X emission is shown in Figure 16 and Figure 17. The increase of engine speed from 1200 rpm resulted with the decrease of NO_X emission. Also retarding of spark timing significantly lowered NO_X emission, but in these cases indicated efficiency also drops. Lower NO_X emission is achieved with leaner mixture in cases at λ = 1.4. Comparison of compression ratio shows a slightly higher emission of NO_X at higher compression ratio which is caused by higher in-cylinder temperature but the difference is much lower than in the comparison of lambda. All measured values are significantly higher than allowed values (0.46 g/kWh) according to the EU standard.

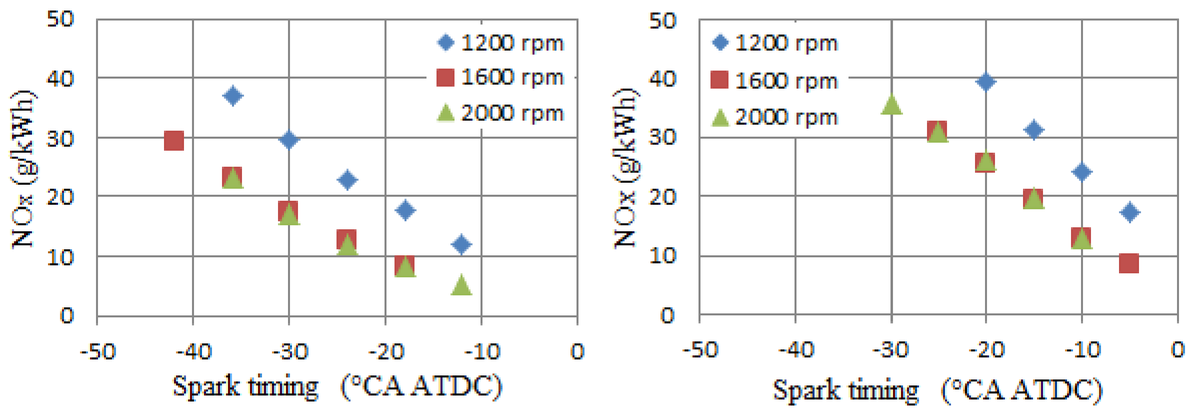


Figure 16 NO_X at λ = 1.2; CR = 12 (left) and CR = 16 (right)

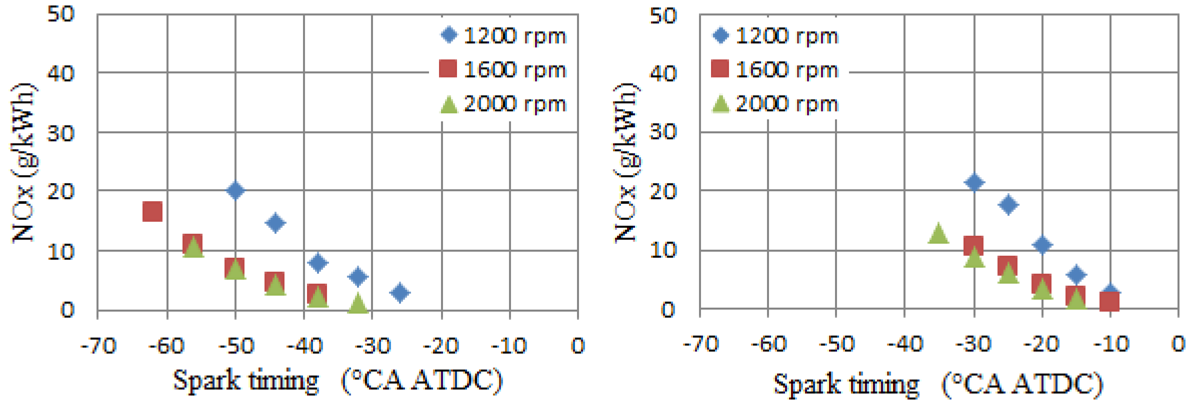


Figure 17 NO_X at λ = 1.4; CR = 12 (left) and CR = 16 (right)

Optimised results

The previously showed results explain how some of the parameters of the engine fuelled with natural gas (methane) change in dependence of spark timing at two different compression ratios at two different lambda values. Since the spark timing is an operating parameter that is optimized with given target in the following section the results are shown for optimum spark timing, where the optimum is obtained with target of highest efficiency. Figure 18 shows indicated efficiency, THC, CO and NO_X emission in dependence of engine speed CR and lambda.

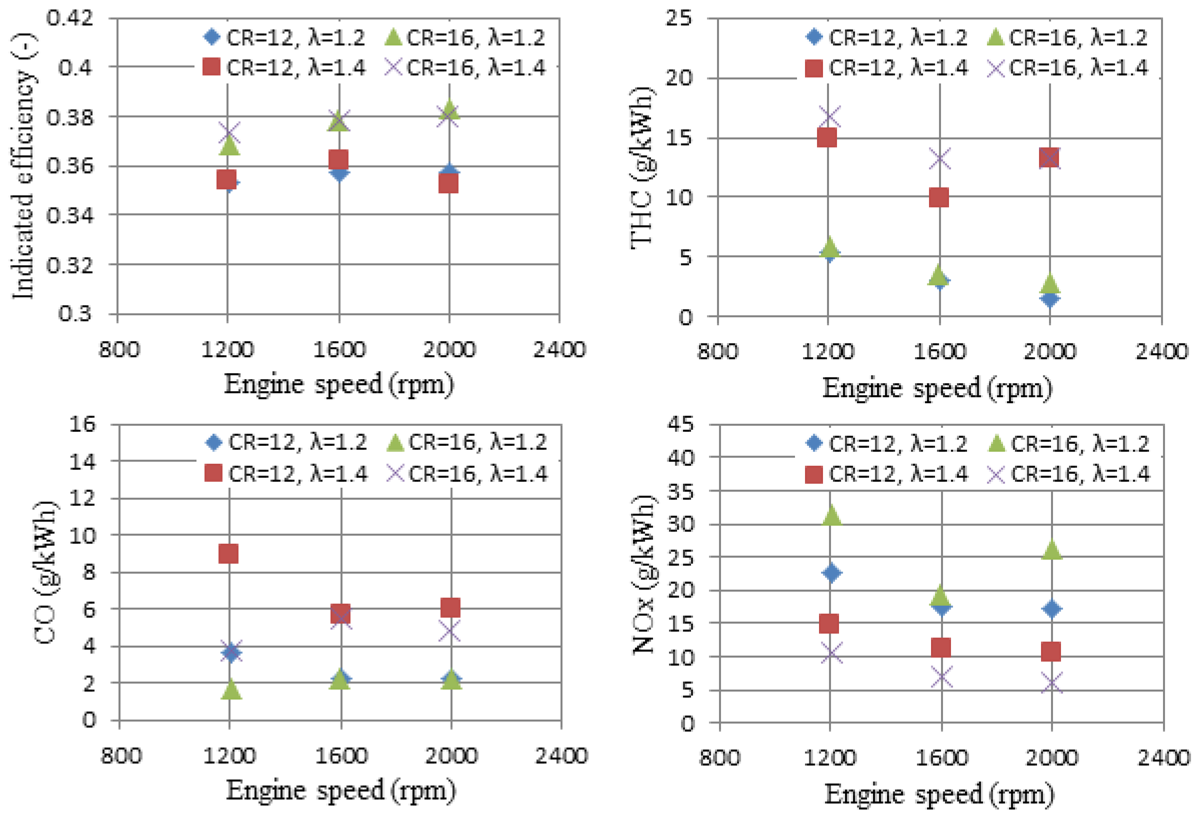


Figure 18 Summary results: Performance and emissions

The optimised results clearly show that indicated efficiency is highly influenced by CR. There is some influence of lambda and engine speed, but the influence of CR is greater with indicated efficiency increasing as the CR increases. It has to be mentioned that in neither of the cases knock was not observed which means that the engine could tolerate further increase of CR. Emissions of hydrocarbons are under a significant influence of lambda value and much lower influence of compression ratio or engine speed. With leaner mixture the emissions of total hydrocarbons increase as well as with increase of CR or decrease of engine speed. Emissions of CO also change with the change of CR and lambda. Here the influence is similar where the increase of CO is obtained with decreasing CR or increasing lambda. The emissions of NO_x decrease with the increase of lambda and with the decrease of CR. There is obviously some interfering effect of CR and lambda since the increase of NOx with simultaneous lowering of lambda and increase of CR (λ = 1.2, CR=16) is much larger than one would expect by looking at other three results. All the emissions of THC and NO_x are above EU limits (NO_X < 0.46 g/kWh, THC < 0.66 g/kWh) which means there will be a requirement for after treatment. Since after treatment of NO_x in lean environment is challenging it might be interesting to test the further increase of lambda (lean burn) or running in stoichiometric conditions.

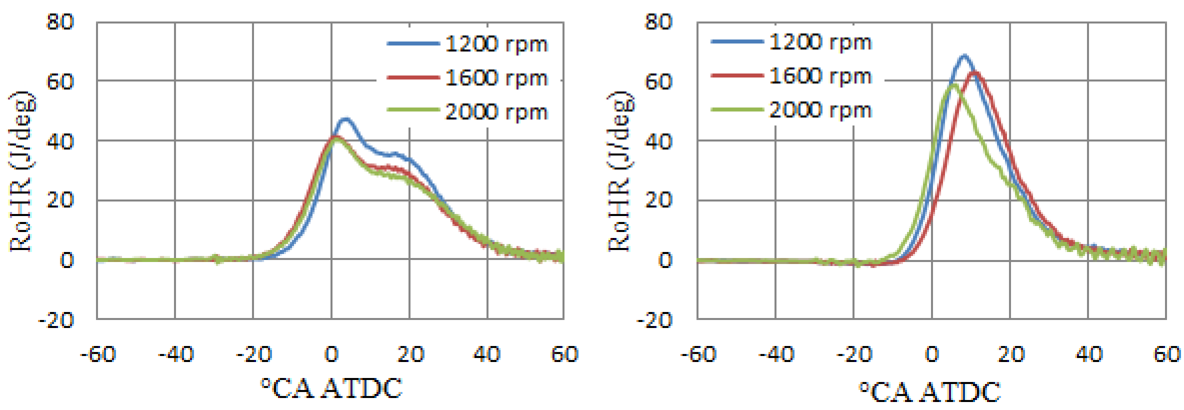


Figure 19 RoHR at λ = 1.2; CR = 12 (left) and CR = 16 (right)

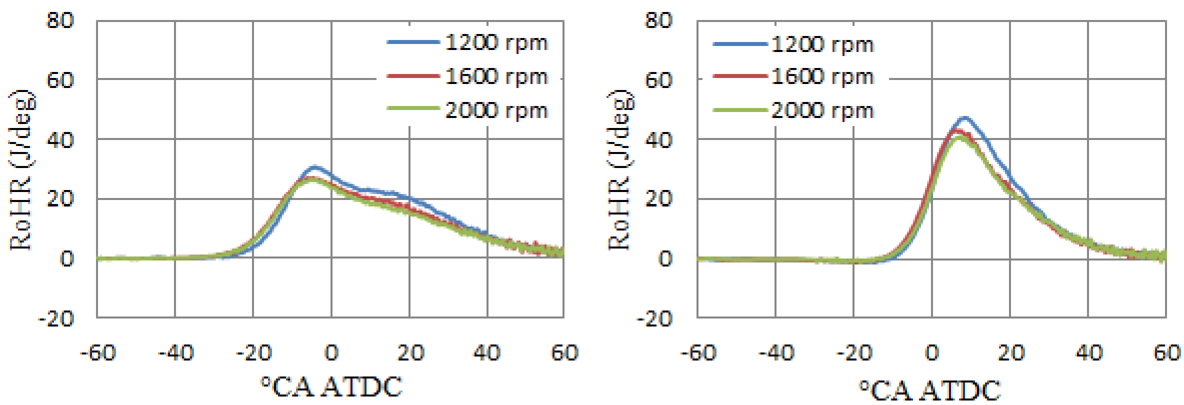


Figure 20 RoHR at λ = 1.4; CR = 12 (left) and CR = 16 (right)

Figure 19 and Figure 20 show rate of heat release (RoHR) for optimized operating points. It can be observed that the increase of compression ratio results with higher peak ROHR values which will lead to larger pressure rise rates and noise. Also increasing lambda value decreases peak ROHR. It can also be observed that with decreasing peak ROHR the peak is obtained at advanced crank angle which means that for optimum operation as the combustion intensity is decreased it has to be more advanced.

CONCLUSIONS

In presented study, the effects of compression ratio, air excess ratio and engine speed on performance and emissions of natural gas fuelled spark ignition engine are examined. The main objective of the work was to show the advantages and disadvantages of using different compression ratio and the possible limitations of this increase. The main conclusions of the paper are:

- Lower compression ratio requires advanced spark timing and achieves lower IMEP and lower indicated efficiency compared to the higher compression ratio.
- At higher compression ratio ignition delay is lower, combustion duration is shorter and CoV of IMEP is lower compared to the low compression ratio.
- Spark timing has significant influence on ignition delay, especially at lower compression ratio.
- IMEP is higher at lower engine speeds probably because of the influence of volumetric efficiency and pumping loses.
- The THC emission is higher at lower lambda which is caused by slower in-cylinder flame propagation and probably flame quenching
- At higher compression ratio THC emission is higher which is probably caused by higher influence of crevices where some amount of fuel remains unburned.
- Higher compression ratio produces higher in-cylinder temperature and lower CO emission.
- The increase of lambda from 1.2 to 1.4 resulted with the increase of CO emission
- Lower NO_X emission is achieved with leaner mixture in cases at λ = 1.4. Comparison of compression ratio shows a slightly higher emission of NO_X at higher compression ratio which is caused by higher in-cylinder temperature

Because of environmental concerns and EU standards, natural gas fuelled SI engines without exhaust gas after treatment system should operate with lean burn strategy. As can be seen from the result, indicated efficiency is at the same level for both lambda values (λ = 1.2, λ = 1.4). This efficiency probably can be retained with further increase of lambda value. Further research should include higher lambda and higher compression ratio to determine the further possibilities of natural gas fuelled SI engines.

NOMENCLATURE

BSFC	Brake specific fuel combustion
AC	Alternating current
ATDC	After top dead center
CA	Crank angle
CA5	The crank angle at which 5% of total heat release occurs (°CA ATDC)
CI	Compression ignited
CH_4	Methane
CNG	Compressed natural gas
CO	Carbon monoxide (%)
CO_2	Carbon dioxide (%)
CoV	Coefficient of variation
CR	Compression ratio (-)

HC	Hydrocarbon
IC	Internal combustion
IMEP	Indicated mean effective pressure (bar)
J	Joule
NH_3	Ammonia
NMHC	Nonmethane hydrocarbon
NO_x	Nitrous oxide
PI	Positive ignition
RoHR	Rate of heat release
SI	Spark ignited
ST	Spark timing
V	Displacement
WHSC	Steady-State Test Cycle
WHTC	Transient Test Cycle
WOT	Wide open throttle
deg	Degree
kWh	Kilowatt-hour
g	Grams
p_{intake}	Intake pressure (bar)
rpm	Revolution per minute
min	Minute
n	Engine speed (rpm)
ϕ	Equivalence ratio
λ	Air excess ratio (-)

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