CHARACTERISATION OF THE COMBUSTION PROCESS IN THE SPARK IGNITION AND HOMOGENEOUS CHARGE COMPRESSION IGNITION ENGINE

by

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Homogeneous charge compression ignition (HCCI) engine is a potential solution for reducing air pollution and for satisfying legal limits regarding the emissions from internal combustion engines. The HCCI engines have advantages of lower emissions of NO_x and particulate matter, compared to the standard combustion modes, while on the other hand one of the major disadvantages is the difficulty of control of start of combustion, since the start of combustion is highly sensitive to the intake air temperature. Additional advantage of the HCCI engine is the ability to operate with wide range of fuels. In order to demonstrate this potential in this study the HCCI mode of operation is compared to the spark ignition mode of operation. The study aims to compare and characterise two different combustion modes on the same engine with different CR and different fuels at similar operating conditions. For that purpose the engine tests are performed at the same indicated mean effective pressures for the spark ignition and HCCI combustion mode at the same engine speed, while the tests are performed at three different engine speeds and three different loads. The measurements were performed on the experimental set-up that consists of single cylinder Diesel engine modified to enable operation in spark ignition and HCCI modes. The characterisation includes the comparison of in-cylinder pressure, temperature and rate of heat release obtained by spark ignition and homogeneous charge compression ignition combustion mode and presents comparisons of engine efficiencies and of emissions of HC, CO, and NO_x.

Key words: *spark ignition engine, HCCI engine, methane, experimental engine testing*

Introduction

In order to reduce air pollution and to meet legal limits regarding the emissions of pollutants the new combustion processes for internal combustion engines (ICE) are constantly being developed. The HCCI combustion is suggested as a potential solution for mentioned challenges. The HCCI combustion process is a form of low temperature combustion [1] and has the advantages of lower emissions of NO_x, compared to both compression ignition and spark ignition (SI) mode of operation, and at the same time of high efficiency (diesel like efficiency numbers). Major disadvantage of the HCCI engine is the control of the start of combustion due to its sensitivity to the intake air temperature [2]. Peucheret *et al.* [3] con-

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ducted experiments on HCCI engine fuelled with natural gas. Besides high compression ratio (CR = 14.5), in order to achieve combustion in HCCI engine, the intake air had to be heated in the range of 140-230 °C. The HCCI engine fuelled by biogas was tested within the study performed by Bedoya *et al.* [4, 5]. It was experimentally shown that HCCI mode of operation has stable combustion with intake pressure around 2 bar and intake air temperature above 200 °C for excess air ratio λ = 2.5-4. Kozarac *et al.* [6] conducted a numerical study regarding the use of exhaust gas re-circulation by applying the negative valve overlap for reducing the requirement for high intake air temperature. The effect of the CR on HCCI combustion fuelled by CH4 was studied by Aceves *et al.* [7]. Chemical kinetics code was used to simulate HCCI combustion of CH4/air mixtures at CR of 14:1, 16:1, and 18:1. They determined the range of operating conditions in which the indicated efficiency is above 50% with NO_x emission lower than 100 ppm. The combustion and performance characteristics of the HCCI engine fuelled by different fuels (n-heptane, n-butanol and isopropanol) was investigated in the study performed by Uyumaz [8]. Testing was performed at a constant engine speed and the effects of the inlet air temperature were investigated. It was concluded that the start of combustion was advanced with the increase of inlet air temperature for all tested fuels. This testing showed almost zero NO emissions in operation with certain fuels. It was also presented that the emissions of CO, and HC decreased with the increase of the inlet air temperature. Exploration of the high-load limits in the low temperature gasoline combustion engine for CR = 16 for early-direct injection partial fuel stratification and premixed fuelling over a wide range of intake pressure is performed by Dec *et al.* [9]. They achieved maximum indicated mean effective pressure (IMEP) of 16 bar with an intake pressure of 2.4 bar. Hasan *et al.* [10] showed that boosting of the intake air pressure has a significant effect on the combustion and performance of HCCI engines. Boosting of the intake pressure tends to significantly increase the peak cylinder pressure during the compression stroke, but it also improves the power output of the HCCI engine. Combustion reactivity increases with the increased boost pressure which significantly advances combustion phasing. In [11], maximum IMEP of 14 bar was obtained by supercharging to 2 bar boost pressure with the natural gas as a fuel in an engine with CR = 17. In [12], numerical simulation of boosted HCCI engine was carried out to study the fuel reactivity at different intake pressures, with findings that are similar to the once obtained in [10, 11]. In [13], it was shown that the combustion duration increases with the large dilution rates for both air and exhaust gas recirculation (EGR). Similarly, the study shown in [14], which is focused on the auto-ignition characteristics of dimethyl ether at boosted intake pressure with EGR, presented that that combustion duration predominantly depends on the EGR addition. Also, it was shown that the required intake temperature had to be increased when EGR was added in comparison with the case without EGR.

Although there are a number of published studies of the combustion process in the HCCI engine, there is no experimental comparison of the combustion process in HCCI mode with the SI mode on the same engine that operates at the same points in the engine map which makes results comparable. This study presents a comparison of these two combustion modes in ICE at similar operating points at optimal operating conditions. Experimental tests that employ an approach significantly different from other research found in the literature were performed, where IMEP was held constant for different fuels, CR and modes of operation and optimisation of operating points at different combustion modes was performed resulting in the new knowledge regarding the combustion process phenomena of the engine. In this study, the engine is fuelled with CH_4, a fuel with high octane number, and RON 95 gasoline. Different CR were used for different fuels and combustion modes, *i. e.* in HCCI mode the CR = 18 was used when the engine was fuelled by CH_4 and CR = 16 when the engine was fuelled by gasoline, while in SI mode the CR = 12 was used with gasoline and CR = 18 with CH_4. In HCCI mode the intake air temperature was used for control of combustion timing and therefore the intake air was heated by the external heater. Due to the physical properties of the CH_4, it was not possible to start the combustion at CR = 16 since it required excessive intake air temperature for auto-ignition of the mixture (over 400 °C), thus the CR was increased to 18. Therefore, the comparison of SI and HCCI mode fuelled by CH_4 is performed at CR = 18. Fuelled with gasoline, the SI mode is limited to CR = 12 due to its lower knock resistance, while the HCCI mode had to be run at CR = 16 because CR = 12 would also require excessive intake air temperature.

Experimental set-up

In the Laboratory for ICE and Motor Vehicles of the Faculty of Mechanical Engineering and Naval Architecture of the University of Zagreb, an experimental set-up for testing of ICE is developed. The core of the set-up is a modified single cylinder Diesel engine Hatz 1D81Z whose main parameters are listed in the tab. 1. The engine is modified so that it can operate in SI and HCCI mode, while the CR is changed by changing the piston and/or by using the head gasket with different thickness. To achieve CR = 12, the top of the original piston was machined by 3 mm and used in combination with head gasket thickness of 1.3 mm. The CR = 16 was achieved with the top of the piston machined by 0.6 mm and the head gasket thickness of 1.3 mm, while the CR = 18 was achieved with the same piston as in CR = 16, but with the head gasket thickness of 0.7 mm.

Table 1. Engine specification

Data	Value
Number of cylinders	1
Bore [mm]	100
Stroke [mm]	85
Connecting rod length [mm]	127
Intake valve open [°bTDC]	20
Intake valve close [°aBDC]	50
Exhaust valve open [°bBDC]	26
Exhaust valve close [°aTDC]	20
Compression ratio [–]	12 16 18

The scheme of the experimental set-up with all of its significant parts is presented in the fig. 1. In this study, the engine is fuelled by CH_4 from the pressurised gas bottles that had 99.5% CH_4. Port fuel injector HANA H2001 was used for the delivery of the gaseous fuel. The engine was also fuelled by RON 95 gasoline taken from the gas station. Gasoline was delivered to the engine by a port fuel injector BOSCH EV-6-E with fuel supply at a constant pressure of 3 bar. The fuel flow was measured by an OHAUS Explorer mass scale for gasoline and with Coriolis mass-flow meter Endress+Hauser Proline Promass A 100 for CH_4. To achieve SI mode of operation, the spark plug was mounted on the engine head at the location of the original diesel fuel injector and accompanied by the corresponding ignition system.

The in-cylinder pressure is measured with an AVL GH14DK sensor which is synchronised with the low pressure sensor AVL LP11DA installed into the intake manifold. During measurement, 300 consecutive cycles were sampled and then used in the analysis. To enable the control of the intake air temperature the Osram Sylvania air heater with 18 kW of installed power was used in the intake system after the intake air settling tank. For the measurement of emissions, the equipment listed in tab. 2**.** is used. The emissions of CO and CO_2 are measured by the non-dispersive infrared method, the emissions of HC are measured by the flame ionisation detector, while the NO_x measurement is performed by a ceramic NO_x sensor. The intake air-flow is measured by the laminar mass-flow meter TSI 2017L [15-17].

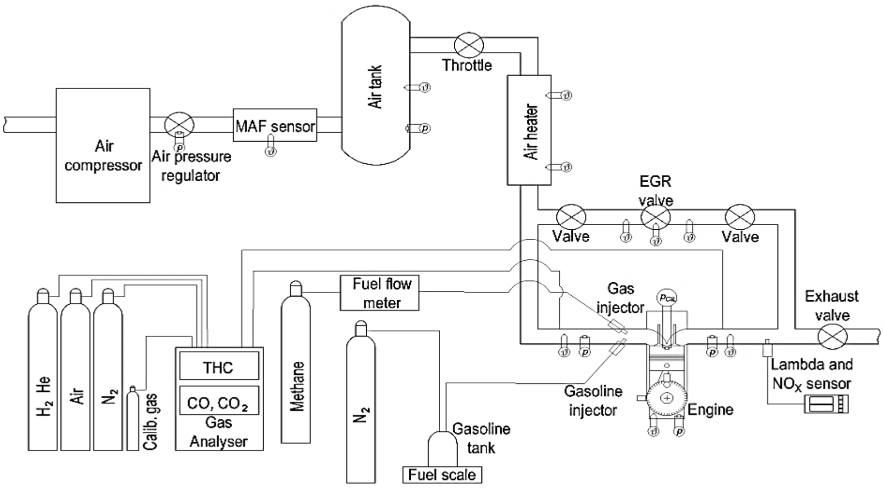

Figure 1. Scheme of the experimental set-up

Table 2. Emissions measurement equipment

Component	Device	Range	Accuracy
NO_x	ECM NO_x 5210t analyzer	NO_x: 0-5000 ppm	± 5 ppm (0 to 200 ppm), ± 20 ppm (200 to 1000 ppm), ± 2.0% (elsewhere)
		λ: 0.4 to 25	± 0.008 (at 1 λ), ± 0.016 (at 0.8 to 1.2 λ), ± 0.018 (elsewhere)
		O_2: 0 to 25%	± 0.4 (0 to 2% O_2), ± 0.8 (elsewhere)
Total hydrocarbon (THC)	Environment graphite 52M	0-10000 ppm	<1% of the displayed value between 15% and 100% of the full Scale
CO	Environment MIR 2M	10-50000 ppm	Zero drift: < 1% full Scale / 24 h Span drift: < 1% full Scale / 24 h Linearity: < 1% from 20 to 100% full Scale
CO_2	Environment MIR 2M	100-250000 ppm	

Experimental testing

This study presents the results of testing of the engine in SI and HCCI mode at operating points which are optimised for combustion phasing (CA50) so that criteria for IMEP, ringing intensity (RI), knock and coefficient of variation of IMEP – CoV(IMEP) are satisfied. The limit for CoV(IMEP) is set to 10% [18] for both combustion modes and in all measured operating points, the CoV(IMEP) was significantly below the imposed limit. The method for achieving the optimal operating point in SI mode, fig. 2(a), was to obtain the maximum IMEP while simultaneously satisfying the limits of the CoV(IMEP) and knock [17]. The measure of knock was maximum amplitude pressure oscillation (MAPO) which was at all operating points under the limit of 0.5 bar since the chosen operating points presented low load in SI mode. The MAPO is defined as the absolute peak value of the band-pass filtered pressure trace as described in [19].

Criteria for determining optimal operating in HCCI mode was maximum IMEP while satisfying the limit on RI [20], as shown in fig. 2(b). In order to determine the optimal operating point during the engine testing, an online program for monitoring of the engine parameters such as combustion phasing (CA50), RI, maximum pressure rise rate (MPRR), maximum in-cylinder pressure, MAPO, combustion noise level, *etc*. is created by using indicating hardware and software package AVL IndiCom [21]. The comparison of the operation in different combustion modes is performed at three different levels of IMEP labelled as

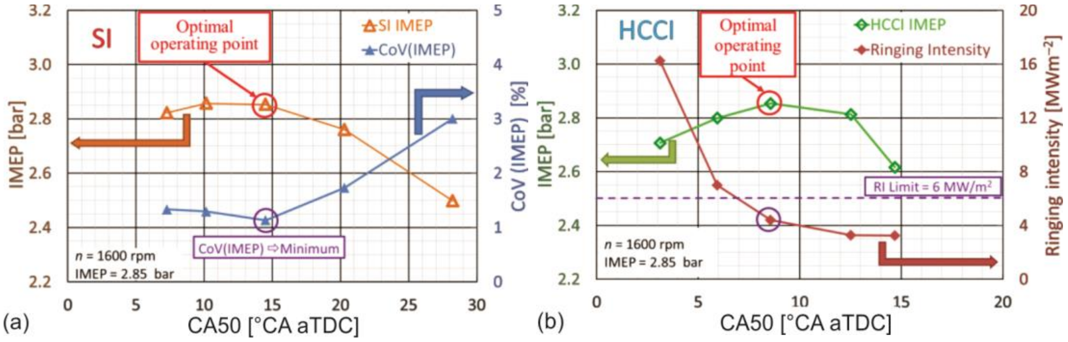

Figure 2. A method of determining the optimal operating point in SI and HCCI mode of operation

IMEP 1 (around 2.17 bar), IMEP 2 (around 2.81 bar), and IMEP 3 (around 3.51 bar). For each level of IMEP the engine was tested at three different engine speed: 1200, 1600, and 2000 rpm. Because it was not always possible to obtain the exact IMEP, there are some variations in obtained IMEP. Since there are three IMEP levels and three engine speeds, nine operating points are tested for each combination of fuel, combustion mode, and CR, which are: SI mode fuelled by gasoline at CR = 12 (hereinafter SI-G-CR12), SI mode fuelled by CH_4 at CR = 18 and at λ = 1 (hereinafter SI-1-M-CR18) and at λ = 1.2 (hereinafter SI-2-M-CR18), HCCI mode fuelled by gasoline at CR = 16 (hereinafter HCCI-G-CR16) and HCCI mode fuelled by CH_4 at CR = 18 (hereinafter HCCI-M-CR18). With the results, a comparison of in-cylinder pressure, temperature, and rate of heat release (RoHR) will be shown together with the measured emission levels of the HC, CO, and NO_x. The control mechanism for the CA50 in SI and HCCI mode is different. In SI mode, the CA50 is controlled by the timing of spark discharge (spark timing) shown in fig. 3(a), while in the case of HCCI mode, the CA50 is determined by the intake air temperature which is controlled by the installed air heater, fig. 3(b). Combustion phasing is advanced by increasing the intake air temperature and *vice versa*.

The engine load in SI mode of operation is controlled by a throttle valve position that changes the intake manifold pressure, fig. 4(a), while the air to fuel mixture was stoichiometric or slightly lean (λ = 1.2 for SI-2-M-CR18).

In HCCI mode of operation, the engine load is controlled by changing λ, where richer mixture results in higher engine load, while the intake manifold pressure was kept on the atmospheric pressure. In HCCI-G-CR16 λ is equal to 3.2, 2.8 and 2.4 for IMEP 1, IMEP 2 and IMEP 3, respectively, while in HCCI-M-CR18 λ is equal to 2.5, 2.1 and 1.9.

Results and discussion

Optimal CA50 for all tested conditions are presented in fig. 5. In SI mode fuelled by gasoline and at CR = 12, the CA50 is closer to the TDC as the IMEP increases. By increasing the load in HCCI mode fuelled by CH_4 engine is more sensitive to ringing so in order to achieve stable combustion within the given limits retarding of the CA50 was necessary. The CA50 decreases, or is constant, as the engine speed increases in SI mode, while in HCCI mode CA50 in nearly constant regardless of the engine speed.

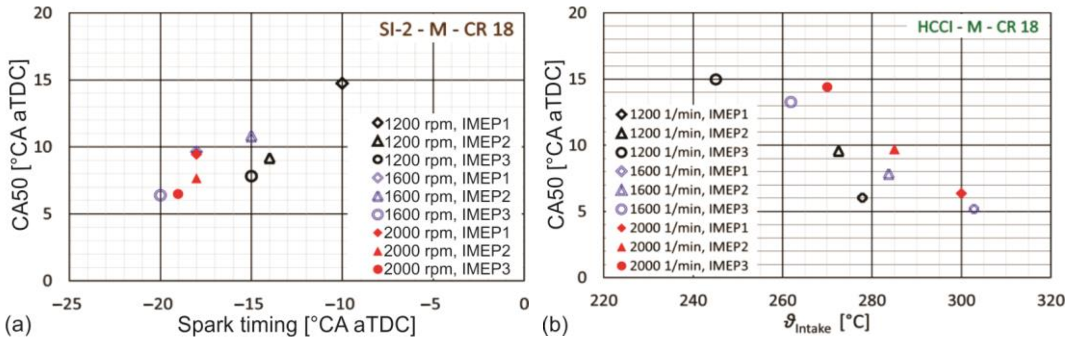

Figure 3. Combustion phase control in SI-2-M-CR18 (a) and in HCCI-M-CR18 (b) mode of operation

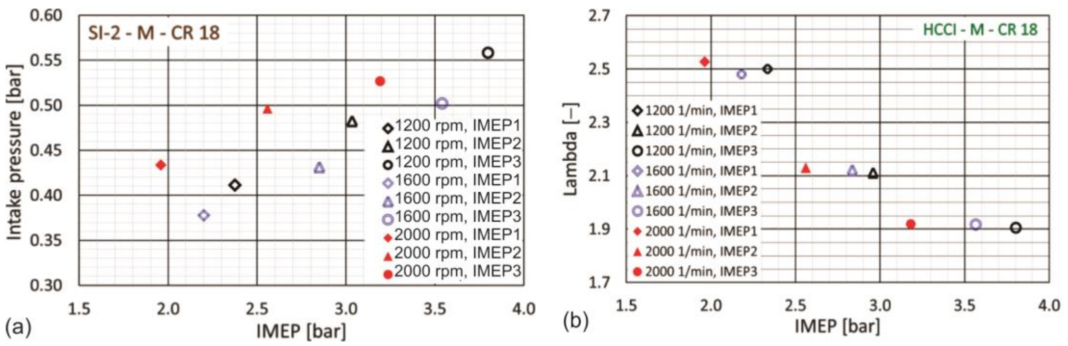

Figure 4. The method for control of engine load in SI-1-M-CR18 (a) and in HCCI-M-CR18 (b) mode of operation

The MPRR (d*p*/d*α*-*MPRR*), fig. 5(b), in all SI cases increase as the level of IMEP increases at constant engine speed, while the MPRR is nearly constant at a constant IMEP and different engine speeds. In the HCCI mode, the change in MPRR with the change of IMEP differs. In HCCI mode fuelled by gasoline the increase of IMEP increases the MPRR, while in HCCI mode fuelled by CH_4 is contrary, since combustion phasing is delayed. When the modes of combustion are compared, all HCCI cases fuelled with gasoline have higher MPRR. This is partially caused by the increase of CR, but the main reason is a much faster chemically driven combustion of the HCCI mode that results with much higher ROHR [22]. With CH_4, this comparison of MPRR depends on the load. At lower load, the HCCI mode shows higher MPRR, while at higher load the SI mode has higher MPRR. The reason for the behaviour at low load is the same as in the comparison of the gasoline cases, while at the high load the optimal operating points of the HCCI mode are obtained at significantly delayed CA50 which then counteracts the faster combustion process and therefore the MPRR is much lower. The change of fuel from gasoline to CH_4 leads to the higher MPRR in SI mode while in HCCI mode that depends on the load. At lower load, the gasoline shows lower MPRR, while at higher load the operation with CH_4 results in lower MPRR.

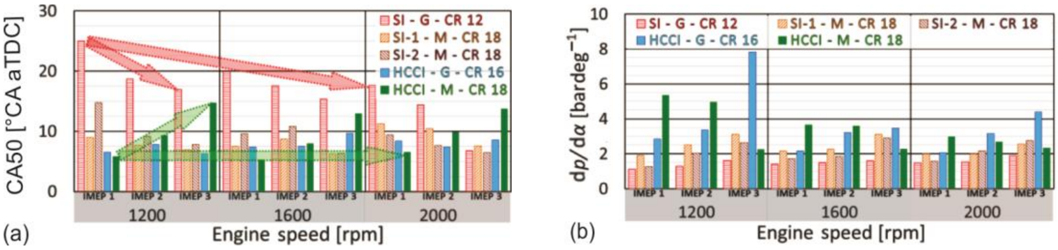

Figure 5. Comparison of the combustion phasing (a) and pressure rise rate (b) in SI and HCCI mode (*for colour image see journal web site*)

One of the criteria for the optimal combustion phasing is the RI which is used as a measure of the knock in HCCI combustion [20]. The limit value for RI in this research was set to 6 MW/m^2 [23], and all of the measured operating points for HCCI mode were below that limit. During the experiments, it was observed that when the HCCI engine was fuelled by gasoline at CR = 16 the RI was very sensitive to the change of intake air temperature, compared to HCCI mode fuelled by CH_4 at CR = 18. This means that in HCCI-G-CR16 mode a very small increase of the intake temperature leads to the significant increase of RI and therefore the operation near the limit was not possible. This sensitivity in HCCI-M-CR18 mode was much lower and it was easier to control combustion phasing which resulted in operating points that are much closer to the RI limit.

The comparison of the in-cylinder pressure of the lower engine speed and load for all five operating modes is presented in fig. 6(a). Since in HCCI mode the load is controlled by λ and the intake pressure is set to ambient conditions, and in SI mode the load is controlled by the throttle position with λ = 1 or 1.2, both HCCI modes of operation have much higher in-cylinder pressure during the entire cycle. Also, HCCI mode of operation has higher CR for both fuels and therefore the increase of pressure is stronger. As a consequence of this high pressure during the entire cycle, the peak pressure is also higher in HCCI mode. The highest in-cylinder peak pressure was observed in HCCI mode fuelled by gasoline, over 60 bar, fig. 6(b) which had fairly advanced combustion phasing, fig. 5(a). By increasing the level of IMEP at the same engine speed, peak in-cylinder pressure increases in all SI mode points while in HCCI mode the trend differs and depends on the combustion phasing. In HCCI mode fuelled by gasoline, since the CA50 remains almost constant, the increase of load leads to the increase of peak in-cylinder pressure, while in HCCI mode fuelled by CH_4 the trend is opposite since the CA50 is delayed with the increase of load. For the same level of IMEP at higher engine speed the peak in-cylinder pressure in all cases slightly decreases or remains constant. The operation with HCCI mode used in this work has on average 39.7% higher peak in-cylinder pressure than in SI mode.

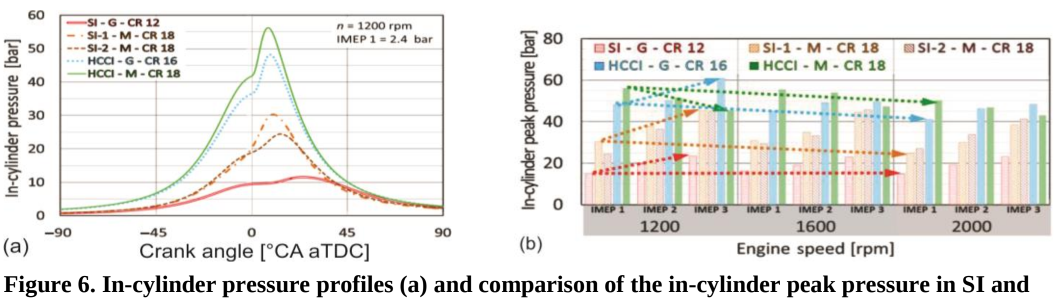

Figure 6. In-cylinder pressure profiles (a) and comparison of the in-cylinder peak pressure in SI and HCCI mode (b) (*for colour image see journal web site*)

Indicated efficiency, fig. 7, which is in correlation with the indicated specific fuel consumption, is in this study higher in HCCI-G-CR16 mode than in HCCI-M-CR18 and corresponding SI mode for all measured operating points. The average value of indicated efficiency for SI-G-CR12 and SI-1-M-CR18 mode is 26%, for SI-2-M-CR18 is 28%, for the HCCI-G-CR16 mode is 35% and for the HCCI-M-CR18 mode is 31%.

Therefore, if compared to the gasoline fuel, the HCCI mode is on average 25% more efficient than SI mode. The difference between the modes is slightly lower for CH_4 fuel since the HCCI mode with CH_4 is slightly less efficient than with gasoline. This is caused by the higher CR and higher intake temperature, which leads to the higher wall heat losses. To understand the source of the difference in indicated efficiency between modes, the results of high pressure cycle (HPC) indicated efficiency and of relative gas exchange (GE) loss are presented in fig. 8. The HPC indicated efficiency is defined as the work during the compression and expansion stroke over the energy of the fuel $\eta_{i,HPC} = (W_{i,HPC}/Q_1)100\%$, while the relative GE loss is defined as the absolute value of work during intake and exhaust strokes over the energy of the fuel $\eta_{i,GE} = (|W_{i,GE}|/Q_1)100\%$. The HPC indicated efficiency of the HCCI mode fuelled by gasoline, fig. 8(a) is the highest in all cases. Together with the low relative GE losses results with the highest indicated efficiency as presented in fig. 7. The GE losses, see fig. 8(b) of the SI mode are higher than the ones in HCCI mode because of the wide-open throttle operation in HCCI mode and throttling effect of the intake in SI mode. Since in SI mode the throttling is reduced at the higher engine load, the GE losses are at that load lower. In contrast to operation with gasoline, when CH_4 is used as a fuel, the HPC indicated efficiency of HCCI mode is not higher than the SI mode, and the whole difference in the overall indicated efficiency is a result of the lower relative GE loss in the HCCI mode.

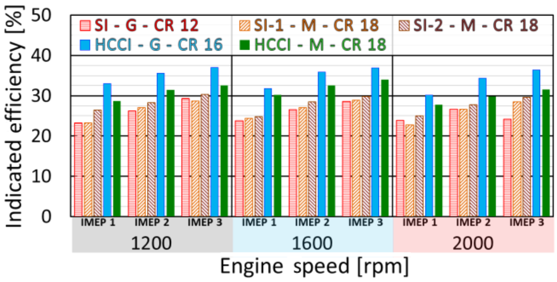

Figure 7. Indicated efficiency of the SI and HCCI mode of operation (*for colour image see journal web site*)

Since a significant formation of the NO_x in the engine cylinder starts at temperatures higher than 1800 K [24], before presenting the results of NO_x emissions, the temperatures of different cases are shown in fig. 9(a). Also, in a fig. 9(b), a comparison of the average cylinder temperature profiles as a function of the crank angle for SI and HCCI mode of operation is shown. Although the increase of temperature caused by compression and the initial temperature are higher for HCCI mode, the dilution of the mixture causes the temperature to be lower.

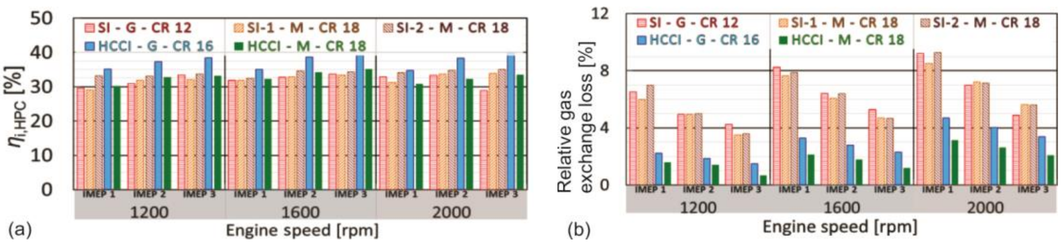

Figure 8. Comparison of the indicated HPC efficiency (a) and relative GE loss (b) for SI and HCCI modes of operation (*for colour image see journal web site*)

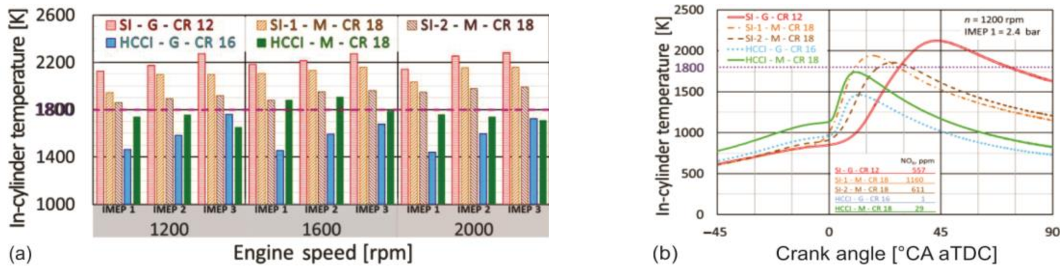

Figure 9. Peak in-cylinder temperature comparison (a) and profile of the in-cylinder temperature as a function of the crank angle (b) (*for colour image see journal web site*)

It can be observed that the average in-cylinder temperature in all SI modes crosses the threshold of 1,800 K which then results in the stronger formation of the NO_x. Opposite to SI, in the HCCI mode the average in-cylinder temperature is below 1,800 K during the whole cycle and therefore it results in the very low emission of NO_x. Figure 10 on the top presents NO_x emissions of all cases, with all the values of emissions displayed in grams per kWh [$gkW^{-1}h^{-1}$] so that it can be easier to compare the emissions of two different combustion modes in the engine cylinder and to compare them with the emission limits. The emissions of NO_x in SI mode are much higher than in HCCI mode in all of the measured cases. The reason for that is a much higher in-cylinder temperature in SI mode of operation, presented in fig. 9. Furthermore, the increase of engine load results in an increase of NO_x in SI combustion mode. In this study, both HCCI modes of operation have on average 25 times lower NO_x emissions than in all SI modes of operation. Also one has to notice that in HCCI mode the NO_x emissions are close to the limiting values for EURO VI heavy-duty engines (in some cases lower and in some slightly higher than the limit) while in SI mode the values are much higher. The emissions of HC are shown in fig. 10 in the middle and it can be noticed that as the mixture is leaner, the emissions of the HC are higher. Therefore, HCCI mode that operates with very lean mixtures (λ up to 3.2 in this study) has in some cases 3 times higher emissions of HC than in SI mode. Emissions of HC in SI mode are between 6 and 11 [$gkW^{-1}h^{-1}$], and the value slightly decreases as the engine speed and load increase. Contrary to the SI operation, the emissions of HC in the HCCI mode fuelled by CH_4 increase with the increase of the engine speed due to a decrease of the in-cylinder temperature and due to the kinetic behaviour of the mixture, *i.e*. less time for chemical reactions to occur as the engine speed increases. Also, in some cases the emissions of HC in the HCCI mode fuelled by CH_4 increase with the increase of engine load due to an overall increase of the in-cylinder temperature. The emissions of HC in HCCI mode fuelled by gasoline are higher, in some cases more than two times than in HCCI mode fuelled by CH_4 due to significantly lower in-cylinder temperature. The emission of HC in HCCI mode fuelled by gasoline decrease with the increase of IMEP due to the same reason as when was fuelled with CH_4. It has to be noticed that all operating points (SI and HCCI mode) are significantly above the limits for HC emissions of heavy-duty engines and would require the application of after-treatment system, *e.g.* oxidation catalyst, the limit for HC emissions of EURO VI heavy-duty engines is 0.13 [$gkW^{-1}h^{-1}$] [25].

The emission of CO decreases with the increase of the engine load and engine speed in the SI mode. In HCCI mode the emission of CO increases as the engine speed increases and decreases as the load increases. When the engine operates in SI mode with a lean mixture (λ = 1.2) the emission of CO is lowest and is below 5 g/kWh. When combustion modes are compared than the difference in CO emission primarily depend on the used fuel.

With gasoline, HCCI mode shows higher CO emission which is expected from the reviewed literature, but when CH_4 is used as a fuel the emission of CO is lower than in SI mode with $\lambda = 1$. The probable cause of this is a very high intake temperature required to auto-ignite CH_4 and consequently higher bulk gas temperatures during HCCI combustion which can be observed by the slight formation of NO_x. As a consequence when fuels are compared the use of CH_4 resulted in lower CO emissions in both combustion modes. The comparison of the RoHR profiles for case IMEP 1 at 1200 rpm of SI and HCCI modes is presented in fig. 11. It can be observed that for the same level of IMEP at the same engine speed the CA50 of HCCI mode is closer to the TDC and at the same time the peak value of RoHR is almost two times higher than in SI mode fuelled by CH_4 and 2.5 times higher than in SI mode fuelled by gasoline. This is a result of shorter combustion duration in HCCI mode (13.6 °CA) compared to approximately 25 °CA in SI mode fuelled by CH_4 and approximately 30 °CA in SI mode fuelled by gasoline. The interesting detail to note is that the RoHR profile of HCCI combustion of both fuels is similar, while, in SI mode the difference in fuels resulted in a significantly different combustion profile at optimal conditions.

Figure 3 showed that there is a requirement to significantly heat the intake in order to achieve HCCI combustion of both fuels, with the higher temperature requirement in the case of CH_4. Since the heat required for heating of the intake can significantly change the result of efficiency, if used from an external source, it is interesting to see if heat from the exhaust could be used for heating of the intake. Figure 12 shows the comparison of intake and exhaust temperatures of HCCI mode. It can be seen that by raising the level of IMEP the requirement for the intake air

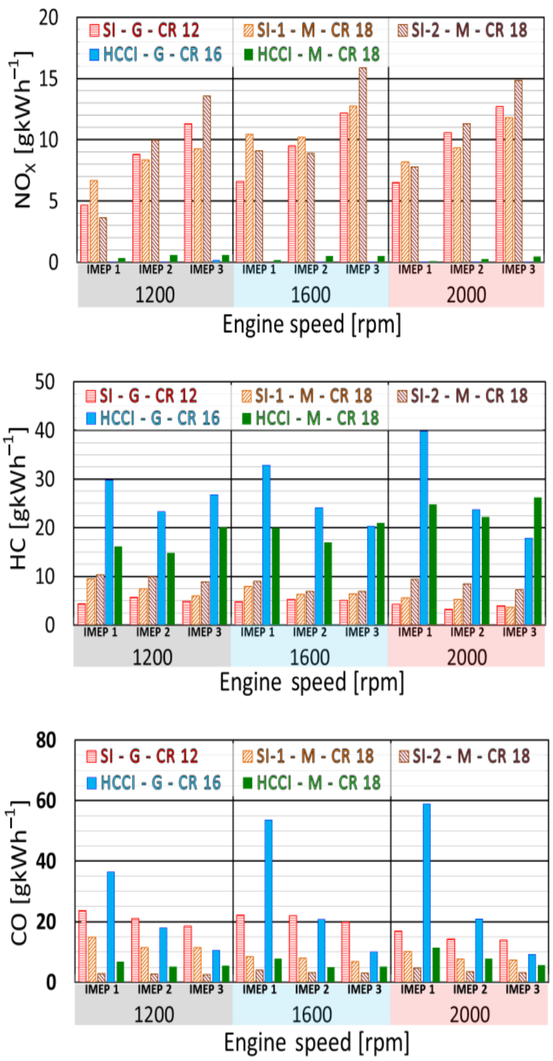

Figure 10. Emissions of NO_x, HC, and CO, displayed in $gkWh^{-1}$

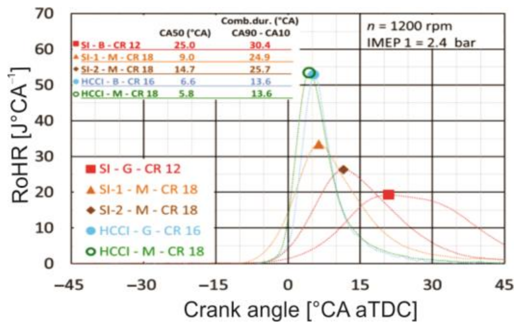

Figure 11. The RoHR for SI and HCCI modes

heating decreases. On the other hand, by increasing the level of IMEP the exhaust temperature increases. The increase in engine speed does not lead to significant changes in intake and exhaust temperatures at the same load. One of the negative effects that can be observed at low load in HCCI mode fuelled by CH_4, marked with red circles in fig. 12, is that intake temperature is higher than the exhaust temperature which means that heat from the exhaust could not be utilized to heat the intake and that external source of heating would be required, which would lower the efficiency of these cases shown in fig. 7.

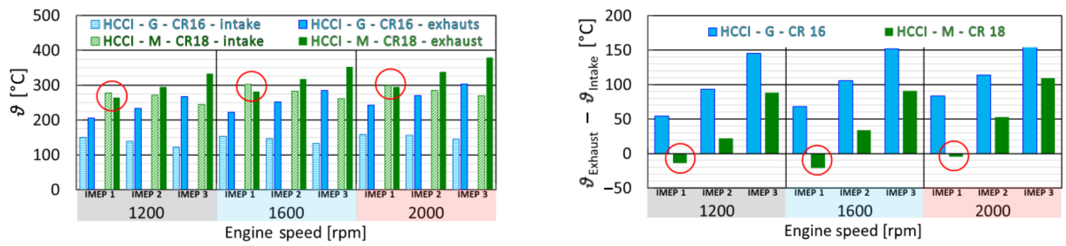

Figure 12. Comparison of the intake and exhaust temperature of HCCI mode fuelled by gasoline (CR = 16) and CH_4 (CR = 18)

Conclusions

Comparison of the SI and HCCI combustion processes in ICE is presented in this study. The SI mode of operation is performed with two different fuels, RON95 gasoline and CH_4 at CR = 12 and CR = 18, respectively. The HCCI mode of operation is performed with the same two fuels at CR = 16 and CR = 18 for gasoline and CH_4, respectively. For the purpose of comparisons, each combination of CR, fuel and combustion mode was measured at equal loads and at optimised operating points. From the results the following conclusions can be drawn:

- The HCCI mode of operation compared to the corresponding SI mode has much higher in-cylinder pressure during the entire cycle due to the higher excess air ratio and the method of the load control. Pressure rise rate in all HCCI modes is higher than in the SI mode with all fuels at all engine speeds. The indicated efficiency is higher in the HCCI mode fuelled by gasoline than in the HCCI mode fuelled by CH_4 and also the HCCI mode has higher efficiency than the corresponding SI mode due to the low relative GE losses. The HCCI mode of operation has much shorter combustion duration, and due to that fact, for the same level of IMEP at the same engine speed, the peak value of RoHR in HCCI mode is almost two times higher than in SI mode.
- The emission of NO_x is in the SI mode much higher than in the HCCI mode because of the significant difference in peak in-cylinder temperature. Due to the application of the lean mixture in HCCI mode, the emission of HC is in some cases three times higher than in SI mode, and all operating points are significantly above the regulation limits for heavy-duty engines. The CO emission of the HCCI mode is higher than in SI mode when gasoline is used as a fuel, while HCCI mode fuelled by CH_4 results in lower CO emission than in SI mode with $\lambda = 1$.
- In HCCI mode, for both fuels, the increase of the level of IMEP results with the decrease of the requirement for the air heating, while at the same time the exhaust temperature is increased. Negative effect with respect to the temperature difference between the intake and the exhaust is observed at the lowest load in HCCI mode fuelled by CH_4 and therefore prevents the use of exhaust gas for heating of the intake air.

From this comparison a general conclusion is that there is a clear benefit of the HCCI mode of combustion in the mid-load area of the standard engine map, especially with lower octane number fuel, under the assumption that the control of combustion phasing can be efficiently achieved.

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Acronyms

CA – crank angle
CNL – combustion noise level
CoV – coefficient of variation
CR – compression ratio
GE – gas exchange
HCCI – homogeneous charge compression ignition
HPC – high-pressure cycle
ICE – internal combustion engine
IMEP – indicated mean effective pressure
MAPO – maximum amplitude of pressure oscillation
MPRR – maximum pressure rise rate
RI – ringing intensity
RoHR – rate of heat release
SI – spark ignition
THC – total hydrocarbon

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