



## Performance Evaluation of Homogeneous Charge Compression Ignition Combustion Engine – A Review

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### Highlights:

- A reduced compression ratio and delayed inlet valve closure contribute to better mixing of the fuel and improve HCCI combustion.
- The inlet temperature and inlet pressure play a prominent role in HCCI combustion and emissions.
- Introduction of EGR and its proportion are very important in achieving proper combustion and obtaining better emission results.

**Abstract.** The development of HCCI combustion technology has been drawing a great deal of attention from researchers. This survey explains ongoing research methodologies and results. HCCI combustion, other than conventional combustion, is purely based on chemical kinetics. At present the automobile sector faces the problem of emissions and needs to develop clean technologies. However, HCCI operation still has issues such as ignition control, combustion phasing control, operating range control, cold start, and UHC (unburned hydrocarbon) and CO (carbon monoxide) emissions. The challenge is to overcome these problems without compromising other engine parameters and performance. For HCCI, the mixture preparation is especially important, while the compression ratio, IVC (inlet valve closure) timing, inlet pressure, inlet temperature and EGR play a very prominent role in controlling it. This paper will go through a detailed discussion of all the above conditions.

**Keywords:** *combustion; EGR; HCCI; inlet pressure; inlet temperature; IVC.*

## 1 Introduction

Homogeneous charge compression ignition (HCCI) is an improved low-temperature combustion mode, where a homogeneous mixture of air and fuel is compressed until the point of auto-ignition. The ignition and consequent combustion of the charge are mostly controlled by substance kinetics [1]. The principal combustion process in HCCI areas auto-igniting consecutively depends on the temperature in their vicinity, with the hottest regions igniting first and the coldest regions igniting later or possibly not at all [2].

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The key constraints that affect PFI (port fuel ignition), SI (spark ignition) and fuel engine are: (a) the stoichiometric task of the engine and the necessity of a three-route exhaust after the treatment framework, resulting in bad mixture properties because of the creation of high convergence of CO<sub>2</sub> and H<sub>2</sub>O; (b) large engine NO emissions because of the high combustion temperature caused by the ignition; (c) throttle load control resulting in critical issues during partial load; (d) large engine unburned hydrocarbon emissions, for the most part caused by crevice HC; (e) the inclination of the engine to knock below low-speed load conditions results in lowering the compression ratio [1].

Testing homogeneous charge compression ignition (HCCI) engines is a requirement for good combustion control design [3]. Significant advantages of HCCI compared to conventional combustion engines are lower NO<sub>x</sub> emissions and strongly reduced smoke issues. Combustion engines are broadly utilized in cars because of their high effectiveness and robustness. In a combustion engine, residues are created in fuel-rich areas and NO<sub>x</sub> in high-temperature areas. Because of this it is hard to diminish both NO<sub>x</sub> and residue creation at the same time through ignition control. To overcome these issues with fuel-rich areas and stoichiometric areas, a homogeneous charge can be utilized instead [4].

**Table 1** Parameters and their limits in HCCI combustion.

Serial no.	Method	Condition limits	Effects	References
1	Preheating of intake charge	30-140 °C	Better combustion and thermal efficiency, lower UHC and CO emission but increased NO <sub>x</sub> with temperature	[4-21]
2	Compression ratio	12-15	Good combustion efficiency but HCCI combustion effect decreases as it increases	[1,14,17, 22-33]
3	Exhaust-gas recirculation (EGR)	0-50%	Decreases NO <sub>x</sub> emission. As EGR increases thermal efficiency decreases	[1,9,12,16, 23,34-38]
4	Residuals (internal EGR)	0-3%	Affects the volumetric efficiency and decreases the HCCI effect	[6,9,12,35, 39-43]
5	Direct injection			[1,22,23, 44-51]
6	Direct injection during the NVO period	-90 to-143 CA ATDC	Better mixing, better HCCI combustion, good combustion control	[1,27,30, 33,35,43, 48,52-61]
7	Boosting the charge	40-140 MPa	Increases the load range, good combustion performance, decreases CO, UHC and soot	[23,28,29, 62-68]

HCCI combustion combines the best properties of gasoline and diesel engines to deliver diesel-like efficiency while maintaining gas-like soot-free outflow within

certain operating limits. HCCI engines are a promising option, with high thermal efficiency and low outflow attributes compared to current internal combustion engines. Table 1 gives a brief explanation of HCCI combustion, its effects and methods used to achieve HCCI.

## 2 Compression Ratio

During early injection in normal HCCI ignition, fuel is infused during the start of the pressure stroke, which permits a mixture duration that is adequate for the development of a homogeneous mixture [22,23]. Reducing the compression ratio (8-15) of the engine is a better trade-off for trouble-free HCCI ignition, considering that the combustion efficiency for different pressure ratios when using primary fuel with octane number (ON) ranging from 0 to 100, HCCI efficiency diminishes linearly with increasing compression ratio.

A decrease in ignition efficiency, which has a negative impact on the fuel utilization of the engine, causes an increase of crevice HC along with a reduction in the combustion rates of HC and CO with increasing pressure ratio [1,24]. Experiments conducted with a PCCI (premixed charge compression ignition) engine showed that under premixed combustion, NO<sub>x</sub> emissions were only slightly decreased if the compression ratio was reduced from 18.4 to 16.0 inside an SOI (start of ignition) timing range from 3 BTDC to 5 ATDC. Ash (soot) emissions, compared to NO<sub>x</sub>, were diminished more notably when the compression ratio was decreased. The decreased ash and NO<sub>x</sub> emissions can be explained by the limited ignition delay (ID), which prompts a decrease in the temperature of the gas at the TDC (top dead center) of the pressure stroke and a more complete mixture of air with energizer [23,25].

By reducing the compression ratio, the efficiency is reduced (50-80%) along with increasing the inlet temperature because of increased heat transfer [14]. The amount of heat released from the combustion HCCI compound indicates a three-stage combustion process accommodating an HTR (high-temperature reaction) and an LTR (low temperature reaction) of the pre-mixture fuel and diffusive combustion of the directly injected fuel [4,15,26,48]. With an increase in  $R_p$  (0-0.8), the highest estimations of the amount of heat release from the LTR and HTR tend to increase when the peak estimation of the amount of heat released from the dispersing ignition diminishes [26,27]. With an increase in gas volume (G30, G40, and G50) in the mixed mixture, the ignition timing of low-temperature ignition is progressively delayed along with the decrease of the peak estimation of the LTR.

This pattern is caused by the diesel fuel, which perhaps responds to the discharge of heat at low temperature ranges while the gas volume increases [26,69].

Moreover, when using a G40 mixture it can clearly be seen that along with the in-cylinder mean gas temperature and maximum in-cylinder pressure, the HCCI ignition increases compared to the other two mixes [26].

With an increased premixed proportion, the extent of diffusive combustion progressively diminishes when the extent of HCCI combustion increases. Accordingly, the ignition qualities of diesel-diesel compound (G30-G40) HCCI combustion changes gradually from DICI-based ignition to HCCI-based combustion [26,70,71]. There is a decrease in the range of HCCI activity at higher compression ratios [17]. The range of adequate SOC (start of combustion) for the mix becomes more narrow as the CR (compression ratio) is increased [17,72]. Iso-octane gives the most usable IMEP (indicated mean effective pressure) without supercharging when utilizing a compression ratio of 17. Still, iso-octane cannot be used at pressure ratio higher than 19, because of early combustion initiation [28,73-75]. To achieve the maximum advantage of HCCI mode, the compression ratio should be between 8 and 16. A mixture of different fuels will give a good output only with conditioned  $R_p$ . The compression ratio also affects the reaction temperature and efficiency. With the control of the compression ratio, HCCI mode can be achieved more effectively.

### 3 Inlet Valve Closure (IVC)

The IVC can be delayed (-90 to -143 CA ATDC) with the goal of deferring ignition and moving the pressure increase closer to the TDC. This results in more effective operation and enhanced thermal efficiency. The delay of IVC is believed to deliver a critical change in both the cool-flame phenomenon and fundamental combustion occurrence closer to the TDC. This can be supported through combustion phasing by modifying the preceding temperature of the fuel-air as a result of LIVC (late inlet valve closure) strategies [27,48]. Delayed IVC times are successful when limiting the 50% AHR (accumulated heat discharges) close to the TDC for optimal combustion phasing [27,54,76]. The residual gases undergo recompression as they accumulate in the cylinder due to the NVO (negative valve overlap) procedure at the final stage of the exhaust stroke. When a rebreathing system is introduced, part of the effectively depleted gases from the chamber are readmitted to the cylinder at the time of the intake (inlet) stroke [1,33,55]. The combustion duration is significantly shortened as the amount of delivered fuel or the equivalence ratio is increased. By differentiation, extended combustion duration happens with reduced equivalence ratio mixtures, that is with diminished adiabatic flame temperatures. The variety of the combustion term along with the variation of the equivalence ratio has a positive effect on top of the observed variation in ignition timing and this prompts a wide pattern of limiting the 50% AHR as the fuelling rate increases. Nevertheless, the increased fuelling and the greater effect of the delayed ignition time caused by the reduced specific heat of

the charge result in a temperature reduction with the increase of the 50% AHR [27,77]. As the equivalence ratio increases, in turn, the specific heat is reduced. The fuel-air mixture results in lower compression and thus increases ID (ignition delay) [56,57]. The duration of the gap between the first-stage and the second-stage combustion can be reduced by increasing the equivalence ratio. While increasing the fuelling time, the diminished specific heat proportion of the charge mixture results in extremely late ignition.

The 50% AHR point moving closer to the TDC and the change in thermal efficiency are after-effects of keeping the fuelling rate steady. As the IVC time is extended, the IMEP increases [27,58]. When the IVO delay is extended, the portion of diluent increases while the ignition rate stays the same. For shorter IVO times, deferring EVC (exhaust valve closure) has a two-stage effect. Deferring EVC, which builds up the residual fraction, first speeds up and then slows down the HCCI combustion. For longer IVO durations, increasing the exhaust leftover only lengthens the combustion occurrence [35]. An overlap strategy can be executed if: (a) an overlap method is used when the inlet pressure and/or temperature are too low to maintain combustion; (b) an IVC method is used when the inlet pressure and/or temperature are too high to phase the combustion appropriately; (c) the IVC assumes the value that results in the maximum volumetric efficiency when the engine is running in overlap-mode; (d) the EVC and the IVO assume the values that result in the most efficient fuel consumption when the engine is running in IVC mode; (e) the transition from overlap-mode to IVC mode is dictated by the saturating overlap against zero. When this occurs, and the IVC is placed at the maximum volumetric efficiency point, changes in the IVC earlier or later in the cycle will result in a lower effective compression ratio; (f) the transition from the IVC mode to the overlap mode is dictated by saturating the IVC against the position for maximum volumetric efficiency; (g) the EVO is dictated by the global engine requirements, regardless of the overlap or IVC method used [59,78]. Using a glow plug with voltage ranging from 0 V to 9.3 V has a great impact on HCCI ignition. A glow plug voltage increase results in advancement of CA50 combustion phasing. As we have seen, the combustion phasing can be controlled with a loss of efficiency of about 1.4% to 3.2% [2,79]. By using a late inlet valve closure strategy, the mixture will become more homogeneous, resulting in better combustion efficiency. An IVC strategy also helps in combustion temperature control and increasing volumetric efficiency.

#### **4 Injection Pressure**

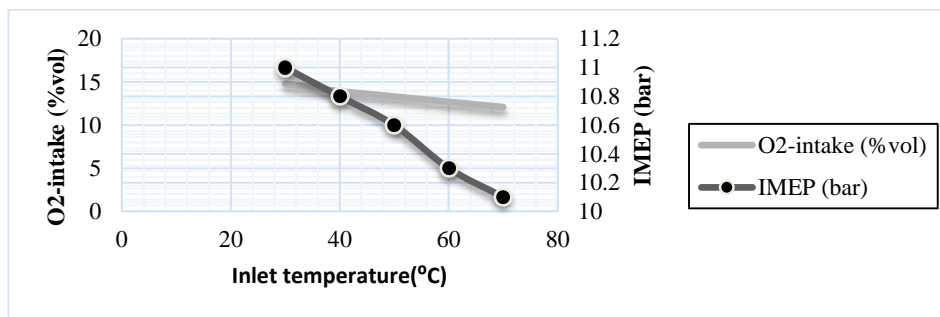
Increasing the fuel delivering pressure (40-140MPa) will improve the mixture of the in-cylinder mass, particularly when combustion is done with a smaller nozzle orifice. At high mixture delivering pressure, the injection speed increases, prompting high air entrainment and mixture, resulting in a positive splash system

and efficient combustion. Moreover, more linear sprays, which are critical for PCI (premixed charge ignition) combustion, are created when the fuel delivering pressure is increased, as demonstrated by the investigation of a pressurized vessel [64] with an optical access and common rail (CR) fuel injection framework [23]. With increased inlet pressure, the efficiency of iso-octane is decreased as a consequence of early combustion initiation [28,80,81]. By using a high EGR to control the ignition and adjusting the compensation point of NO<sub>x</sub> exchange, the amount of mixture that is injected can be controlled to a certain degree to achieve a constant charge mass to prevent engine power loss. The power yield can be increased to almost the full load of conventional diesel combustion when supercharging to 86 kPa and increasing the fuel amount for a decreased compression ratio of 12.5 [23,53]. When the supercharging pressure is two bars, the necessity of preheating is lower due to the temperature increase occurring because of increased pressure in the supercharger. This implies that no additional system would be required for preheating [28,65]. In case of supercharging, the ignition duration is longer than for normally aspirated cases. This relies upon more of the mixture being consumed in supercharged events. Hence, when the term of combustion is to be equal for a supercharged event compared to a normally aspirated case, the rate of ignition must be higher for the supercharged case for a given estimation of  $\lambda$  [28,66]. The equivalence proportion increases as the inlet valve closure time is delayed in case of supercharging. As the pressure increases, the global equivalence ratio diminishes because the fuel input amount is kept constant. Thus, supercharging can contribute to a faster ignition reaction, delivering advanced ignition times if the HCCI combustion is dynamically controlled and the global response rate specifically corresponds to the pressure. The conflicting effects of advanced ignition and extended combustion duration balance out, causing an almost imperceptible change in half consume concentration for a given inlet valve closure time with increased boost [27]. By imposing a boost on the inlet for the HCCI mode, the combustion duration is prolonged. The boosting strategy helps to achieve more load carrying capacity during HCCI mode. This strategy using a supercharger eliminates preheating at the inlet.

## 5 Inlet Temperature

With a higher inlet temperature (20-130 °C) and a lower compression ratio, the temperature is further increased for a more extended timeframe. Mathematical and test results from investigations have demonstrated that cool flames occur at temperatures many degrees below the auto-ignition temperature of the mixture. With increased inlet temperature [4,5], the cool flame phenomenon can occur at temperatures below the auto-ignition temperature, prompting two consecutive ignitions. This results in increased NO<sub>x</sub> production. The highest amount of heat discharge by cool flames increases along with the increase of the premix ratio

[6,7]. Pre-mixture diesel fuel for combustion in a direct-injected diesel engine can be more separated by further increasing the intake air temperature. The ignition delay of the direct injection mixer seems to decrease as the pre-mixture proportion increases. Consequently, the delayed start of combustion (BTDC) increases the ignition pressure in an orderly fashion [6,8]. A sudden increase of  $\text{NO}_x$  emissions at high channel heat is one of the factors that result in the operational range of diesel premixing being more constrained than gas premixing [6]. Homogeneous combustion, with its steeply rising pressure, creates large combustion disturbance. To prevent this, the charge must be weakened to a great degree, utilizing EGR [9,10]. By increasing the EGR rate (0-50%), the combustion reaction rate is lowered, the average temperature in the cylinder is decreased and the ignition reaction becomes more inadequate. The problem is that more CO cannot be oxidized totally into  $\text{CO}_2$  on account of the reduction in temperature [9,82-84].



**Figure 1** Variation in IMEP and intake oxygen with variation of inlet temperature [68].

Setting the glow plug voltage to 6.5 V means increasing the intake temperature by 6 °C. Based on the glow plug information it has longer combustion duration at higher intake temperature, so the presence of a glow plug contributes to longer combustion duration and affects the temperature distribution [2,11,13,14]. The prior combustion phase, either with higher glow plug voltage or higher intake temperature, has more a wider temperature distribution with a lower maximum distribution point [2,13,15,16], increasing the upper limit of the distribution. When the wall area is unaltered, the maximum distribution is lowered and the range is also larger when utilizing a glow plug. The unburned temperature distribution [85,86] can be controlled to some degree by effectively controlling the plug voltage and in the same way we can have some control over the heat discharge rate in HCCI [2]. This happens because of the fact that with an increase of the intake air pressure (100-150kPa), the HCCI combustion rate progressively diminishes and the mixer mixture in the cylinder is thinner. With an increase in the intake air pressure, the IMEP-net first decreases and then increases, while the

maximum mass average temperature first decreases quickly and then gradually. [26] As the charge temperature (40-130 °C) or the compression ratio is increased, the ignition delay time is shortened and the SOC is delayed. As the SOC is delayed, knock starts to become an issue and the range of HCCI operation is narrowed [17]. The lower inlet air temperature consequently increases lambda and increases the CR productivity for a consistent load. The ignition staging (CA50) is almost the same for all intake air temperatures and is marginally lower at higher loads, around 5.5° ATDC, while it is around 3°ATDC at low loads [20,24]. As a rule, the IMEP decreases with a higher stratified charge. A lower IMEP with stratification can be connected for the most part to bad combustion phasing. Another possible explanation for the decrease in IMEP could be deficient combustion, as shown by the high CO outflow. Operating conditions with high DI proportions and early injection show a lower IMEP with respect to a homogeneous charge. In any case, at low DI (direct injection) ratios, the IMEP increases over the homogeneous charge condition with stratified charge [21,87]. The average inlet temperature for a better HCCI mode is in the range of 40 °C to 110 °C. As the inlet temperature increases, in turn, the volumetric efficiency and the knock range are increased. The inlet temperature greatly affects the overall heat distribution in the cylinder during the combustion process and also helps the advancement of the SOC.

## 6 EGR

Exhaust gas dilution, which can be done by exhaust fume distribution or by catching in-cylinder leftover gases, is likely the best strategy to stimulate and control auto-ignition and to increasing the load range of HCCI-ignition mode. Reactive groups present in the leftover gases encourage auto-ignition and idle species lower the combustion rate (dilution effect) [1,88]. EGR is utilized as a method for diluting the gas mixture in HCCI engines by delaying the ignition timing. A great deal of EGR (54%) is sent to delay the ignition towards the top dead center and enhance the IMEP in a premixed charge compression ignition (PCCI) combustion framework. Large exhaust gas rates up to 68% were utilized to govern the beginning of combustion in another PCI ignition framework. Large EGR amounts can likewise be utilized to counter combustion disturbance by controlling the start of ignition. However, such a large amount of EGR also has disadvantages, including temperature-stability issues in transient reactions [16]. Changing the fuel properties or using another chemical approach can be added for advanced injection HCCI combustion with EGR. As a result of delayed injection, for example in the MK ignition framework, EGR is ordinarily used as an NO<sub>x</sub> reduction measure with common levels of around 40%. When EGR is introduced, charge dilution and the temperature limit of the cylinder are increased and NO<sub>x</sub> emissions are reduced because of the limited flame temperature [23]. The brake thermal effectiveness diminishes with the increase in EGR rate for



diesel vapor tolerance compared to when a direct injection mode of operation is applied [9,89]. Because of pressure loss in the vaporizer and problems due to residual build-up as a result of the amount of cooled EGR, the brake thermal efficiency is reduced. This may be because of an increase in HC and CO emissions [9,34]. The HCCI ignition can reduce NO<sub>x</sub> emissions up to 90-98% compared to traditional direct injection diesel combustion. At the point when colder EGR is utilized with a pre-mixed diesel vapor-air mixture, NO<sub>x</sub> emission is more reduced because of the lower combustion temperature and pressure [9,20]. There are two major impacts on the HC discharge because of EGR. The first is that the decrease of ignition heat inside the cylinder prompts an increase in HC emissions, the second is that the intake of some residual HC with depleted gases into the following cycle prompts a decline in hydrocarbon emissions [9]. As the EGR rate increases, the high-temperature reaction (HTR) peak is suppressed. This is because of the impact of the cylinder average temperature and EGR on the peak pressure increase as the EGR ratio pushes the peak pressure closer to the TDC. With the increase of the EGR rate, the maximum pressure rise rate is decreased. Increasing the EGR rate can delay both the LTR and the HTR. The CA50 decreases as the EGR rate increases. This shows that ignition happens earlier and closer to the TDC. Additionally, combustion phasing can also move it closer towards the TDC [9], so the EGR makes it possible to shorten the extremely delayed ignition by the low temperature response of the diesel-fuelled HCCI engine.

Knocking caused by too advanced ignition in fast ignition constrains the operational range of HCCI combustion [6,39,90]. The high-temperature response of pre-mixture fuel at high channel temperature is significantly better before injection of the fundamental fuel and shows a large increase in the pressure rise rate and heat discharge rate. Knocking occurs at a larger premixed proportion, corresponding to the increase of the EGR rate [6]. From the above observations, the conclusion can be drawn that the EGR range should be between 20% and 30%. EGR greatly affects the volumetric efficiency and at the same time helps controlling the combustion temperature and the emission level.

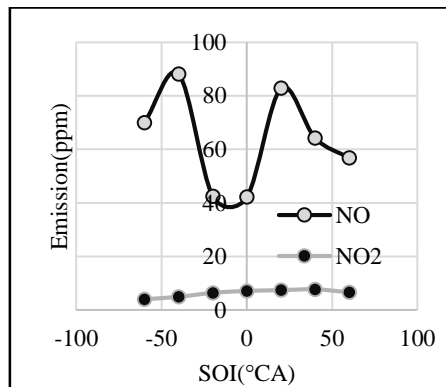
## **7 Emissions**

Close to inert conditions cause considerable combustion inefficiency because the low combustion temperature in the mass gas terminates the CO and HC combustion reactions and due to the significant increase in mixer utilization and CO and HC emissions [1,53]. In premixed lean diesel combustion (PREDIC), because of overheating of the air-fuel mixture, NO<sub>x</sub> formation is considerably lower compared to ordinary diesel-combustion, but it is accompanied by increased HC and CO emissions.

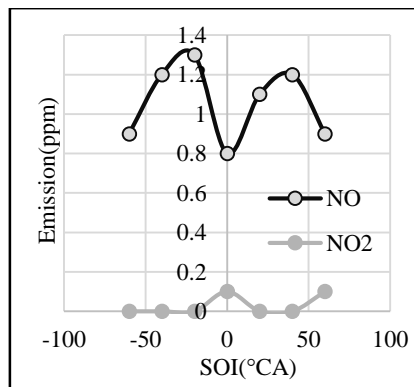
The disadvantages of the PREDIC combustion framework include restricted halfway load task and absence of ignition timing control [1,47]. With advanced injection in the range of 75-40 CA BTDC and an equivalence ratio of 0.38, nozzle intake velocity is high, with fuel sticking to the ignition chamber divider. Compared with conventional diesel ignition, NO<sub>x</sub> emissions decrease to a low level but HC and smoke emissions both increase. Utilizing a high-pressure wide-angle nozzle permits high fuel scattering and low splash infiltration, which causes low outflow of both NO<sub>x</sub> and smoke [23,91]. Multiple injection systems have been developed according to this principle for advanced single-injection for diesel combustion in HCCI mode. Frameworks that feature multiple injection include multiple stage diesel combustion (MULDIC), where the first phase of ignition is premixed combustion, which helps decrease the NO<sub>x</sub> outflow, while the next stage is dispersion/ignition, which occurs under high temperature and thin air conditions. This enables the MULDIC operating range to be extended to increased load conditions, with a great decrease in NO<sub>x</sub> formation at higher load constant compared to the PREDIC delayed injection system [23]. By delaying the injection start timing until after BTDC, the gas temperature and thickness decrease due to cylinder expansion, which prompts longer ignition delay (ID) and enhanced mixture formation [23,92].

If higher amounts of EGR are used, the peak heat release ratio diminishes, prompting less NO<sub>x</sub> development. The ID is extended, permitting sufficient time for improved air-fuel mixture, which decreases net residue creation. Highly premixed late injection (HPLI) and homogeneous charge late injection (HCLI) are the most recent HCCI diesel-combustion approaches that include delayed injection timing [23]. With homogeneous ignition of a premixed mixture, the temperature is required to be the same in the whole ignition chamber, except close to the walls [4,17,93,94]. NO<sub>x</sub> emissions are increased with the increase of the fuel octane number since a large amount of heat is required to get auto-ignition.

With a constant octane number, NO<sub>x</sub> emissions are reduced by increasing the compression ratio and lowering the channel temperature. Due to poor atomization of diesel fuel and more inhomogeneous mixture mixing, the temperature of combustion will be higher in the higher mixture zones and thus more NO<sub>x</sub> is generated [4,95]. A low ignition temperature causes high emissions of unburned hydrocarbons [4]. The ignition temperature close to the cylinder wall will be even lower, because of heat loss [4,94,96]. When the engine is operated with early combustion timing (by increasing the compression ratio), the residue emissions increase strongly [4]. NO<sub>x</sub> formation diminishes with an increase of the premixed ratio. Increased NO<sub>x</sub> formation lower premixed ratios possibly results from the increased gas temperature caused by the delayed ignition preceding TDC [6].



**Figure 2** Measured mole fractions of the exhaust port sample [52].



**Figure 3** Measured mole fractions of the intake port sample [52].

The crevice effect and flame quenching near the walls result in an increase of HC. The increase of the HC emission rate is expected to be more extreme with diesel premixing due to its poor atomization [6,94]. At the highest voltage setting of the glow plug,  $\text{NO}_x$  generation was reduced by 15% [2]. With an increased premixed proportion, the CO, HC and soot emissions first increased and then decreased, while the  $\text{NO}_x$  discharges diminished monotonically for each tested fuel [26,97]. When the premixed proportion increases, premixed mixture fixation before auto-ignition increases correspondingly. At that point the general ignition is overwhelmed by HCCI combustion. Thus, the CO and HC emissions start to increase while the  $\text{NO}_x$  emissions decrease. The CO emissions are higher with the utilization of intake air boost [26,98]. It is implied that  $\text{NO}_x$  emissions are extremely sensitive to local temperature. HCCI ignition is known to produce near zero  $\text{NO}_x$  discharge because of the low combustion temperature [21,88,99]. The available information suggests that a more heterogeneous charge, accomplished by the delay of the injection timing and an increase of the DI ratio, prompts high

NO<sub>x</sub> emissions. At the most delayed injection permitted by the highest pressure rise rate, the NO<sub>x</sub> emissions are practically identical to that of the low load states of a diesel engine [21,54]. The NO<sub>x</sub> discharges seem to diminish at higher engine speed. This suggests that there are fewer local high-temperature zones at higher engine velocity. Increasing the DI ratio and delaying the injection timing, the NO<sub>x</sub> emissions increase ever more quickly as the charge (temperature) moves closer to the engine knock limit [21,100]. Charge stratification can significantly enhance UHC (unburned hydrocarbon) emissions. Increasing the DI ratio greatly reduces the discharged UHC [21,101]. High levels of UHC and CO are emitted, which are assumed to originate from insufficient combustion, extinguishing, and crevice effects [21,94,102]. CO increases with stratification, so less CO-CO<sub>2</sub> oxidation occurs [21]. Less NO<sub>x</sub> is produced at lower piston cavity temperature. This demonstrates that NO is sensitive to temperature even with a high NO temperature limit. A result of this is reduced NO<sub>x</sub> formation in case of supercharging. As there is no flame lead in HCCI, less fuel should be stuck in the crevices. When the cylinder pressure increases, a larger portion of the gas is forced into the crevices [13,93], which is officially consumed and thus not added to the HC outflow. With an increase in intake pressure, and consequently in IMEP, specifically the HC emissions are reduced for ethanol and iso-octane. When the engine is normally suctioned, flammable gas gives the lowest amount of HC. With supercharging, petroleum gas and ethanol create similar levels of HC. With HCCI, CO is exceptionally reliant on load and preheating. Little CO is generated under conditions of high load and hot inlet air [28,103].

It is important to control emissions in HCCI mode without compromising the performance of the engine too much. By implementing all of the above-mentioned strategies, NO<sub>x</sub> emissions can be reduced with increased CO, UHC and soot emissions as trade-off.

## 8 Conclusions

HCCI combustion works better at lower pressure ratios (12-18), but a higher compression ratio results in better emission reduction of UHC, CO, and soot. Results with higher compression ratio were not considered. With respect to delayed inlet valve closure timing there are indications of improved mixture properties, because it provides sufficient time for enhanced mixing. Changing the timing for various loads needs to be contemplated for continuous engine operation. The experiments and investigations in all studies were done in single load condition (low load). More investigation must be done for higher load conditions. There is very little knowledge about the performance of HCCI combustion with continuously varying loads.

Increasing the inlet temperature results in better performance and has the added advantage of decreased NO<sub>x</sub> emissions. The equivalence ratio is the most critical parameter for HCCI. By increasing the inlet pressure, HCCI can be operated at slightly higher load, but the HCCI combustion rate diminishes. From all the results we can conclude that by regulating the EGR, NO<sub>x</sub> emissions can be reduced with the drawback of increased UHC, CO and soot emissions. Furthermore, efficiency decreases at low temperature. In terms of NO<sub>x</sub>, CO, UHC, and soot emissions, the ideal operating range has not been found.

### Nomenclature

CA50	=	Crank angle at half of the fuel mass burnt
CA_ATDC	=	Crank angle after TDC
CA_BTDC	=	Crank angle before TDC
DICI	=	Direct injection compression ignition
EGR	=	Exhaust gas recirculation
EVO	=	Exhaust valve opening
NVO	=	Negative valve overlap
G30, G40, G50	=	Percentage of gasoline volume in the mixture
ID	=	Ignition delay
IMEP	=	Indicated mean effective pressure
IVC	=	Inlet valve closing
IVO	=	Inlet valve opening
NVO	=	Negative valve overlap
PCCI	=	Premixed charge compression ignition
R <sub>p</sub>	=	Compression ratio
SOC	=	Start of combustion
SOI	=	Start of injection
TDC	=	Top dead center

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