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EFFECT OF MIXTURE PREPARATION IN A DIESEL HCCI ENGINE USING EARLY IN-CYLINDER INJECTION DURING THE SUCTION STROKE

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ABSTRACT–It is becoming increasingly difficult for engines using conventional fuels and combustion techniques to meet stringent emission norms. The homogeneous charge compression ignition (HCCI) concept is being evaluated on account of its potential to control both smoke and NOx emissions. However, HCCI engines face problems of combustion control. In this work, a single cylinder water-cooled diesel engine was operated in the HCCI mode. Diesel was injected during the suction stroke (0° to 20° degrees aTDC) using a special injection system in order to prepare a nearly homogeneous charge. The engine was able to develop a BMEP (brake mean effective pressure) in the range of 2.15 to 4.32 bar. Extremely low levels of NOx emissions were observed. Though the engine operation was steady, poor brake thermal efficiency (30% lower) and high HC, CO and smoke were problems. The heat release showed two distinct portions: cool flame followed by the main heat release. The low heat release rates were found to result in poor brake thermal efficiency at light loads. At high brake power outputs, improper combustion phasing was the problem. Fuel deposited on the walls was responsible for increased HC and smoke emissions. On the whole, proper combustion phasing and a need for a well-matched injection system were identified as the important needs.

KEY WORDS : HCCI, PCCI, Homogeneous charge mixture preparation, Low NOx, Early injection

NOMENCLATURE

aTDC : after top dead center
ATAC : active thermo-atmospheric combustion
BMEP : break mean effective pressure
BSU : bosch smoke units
bTDC : before top dead center
CA : crank angle
CI : compression ignited
CO : carbon monoxide
DME : di-methyl ether
DTBP : di-tertiary butyl peroxide
EGR : exhaust gas recirculation
EGT : exhaust gas temperature
FID : flame ionization detector
HC : hydrocarbon
HCCI : homogeneous charge compression ignition
HRR : heat release rate
HTR : high temperature reaction
NOx : nitrogen oxide
PM : particulate matter
ppm : parts per million
RON : research octane number
SOC : start of combustion

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1. INTRODUCTION

The main advantages of the diesel engine are its high fuel efficiency and torque. Thus, they are the most preferred prime movers for mass transportation and power generation. However, they are bulky and noisier than their gasoline counterparts. Recently, increasing fuel prices have made cars with lower fuel consumption more attractive. In spite of this and with the recent advances in diesel combustion technology, they have not been able to significantly enter the passenger car sector. Still its higher fuel efficiency has not outweighed its disadvantages significantly. The major emissions from the diesel engines are CO, HC, NOx and particulate matter. HC and CO emissions are typically very low. However, soot and NOx emissions are significant and it is difficult to simultaneously reduce both. The main reason for this is that the formation of both emissions is strongly affected by factors that work on both these emissions in opposite directions. Increase in cylinder gas temperature reduces soot oxidation, but increases NOx formation rapidly.

Therefore, the diesel engine combustion system has to compromise to both limit the emissions and keep fuel efficiency high.

In the development of new emission-reduction engine technologies for passenger cars, the focus is mainly on part-load operation. Unlike the heavy-duty diesel engines used in commercial vehicles, which spend much of their time at higher loads, the engines of modern passenger cars are only subjected to relatively light loads in typical traffic conditions. To make progress in the task of achieving simultaneous reduction of NOx and soot emissions with good part-load efficiency, the focus is on the search for alternate combustion processes. An alternate combustion process has been demonstrated by a number of independent workers in which a homogeneous fuel-air mixture is ignited spontaneously throughout the entire mixture. Thus, it avoids the formation of soot that occurs in mixing controlled combustion. This combustion system was called by different names by different researchers. Generally, this combustion system is referred to as the Homogeneous Charge Compression Ignition (HCCI) combustion system.

In an HCCI engine, a homogeneous mixture of air and fuel is compressed and ignited as the temperature rises on compression. HCCI combustion can be considered as a hybrid form of the diesel and Otto engine combustion processes as it combines the homogeneous mixture preparation of an Otto engine with the compression ignition of the diesel engine. However, the combustion process is different. When the temperature and pressure of the mixture are high enough, the compressed homogeneous charge ignites simultaneously at multiple spots in the combustion chamber so there is neither a diffusion flame, as in a diesel engine nor a flame front traveling through a premixed charge, as in a spark ignition engine. The disadvantages of an HCCI engine are controlling the combustion process and the limited operational band of load since the timing of the start of combustion is determined by the ambient conditions in the cylinder and can not be controlled nearly as precisely as with a spark ignition or a diesel injection system, especially under transient conditions. Whether the HCCI engine is based on an Otto or a diesel engine does not really affect the combustion process and the main difference lies in the type of fuel used. For high-octane fuels, the heat of compression for any realistic compression ratio is usually insufficient and the temperature of the charge typically needs to be increased for autoignition to occur. This can be done by preheating the air in the intake system or by internal EGR. When diesel fuel is used instead, the difficulties are exactly the opposite, as the temperature during compression must be reduced considerably to prevent premature ignition. Preparing a homogeneous mixture is also a challenge with most fuels.

Several research groups around the world are involved in the development of the HCCI combustion process with varying degrees of success. It is still not at a level that an engine can be made to run in this mode at all operating conditions. The concept has been demonstrated in two stroke engines by Onishi *et al.* (1979). They termed this combustion mode "Active Thermo-Atmospheric Combustion" or ATAC. Cycle-by-cycle repeatability was improved when a two-stroke engine was run in the ATAC combustion mode and this combustion system was extended to four stroke engines by Najit and Foster (1983). They conducted experiments with primary reference fuel (PRF) to understand the physics of this combustion process and concluded that auto ignition combustion was controlled by the low temperature chemistry and there was little possibility of active radicals being recycled into cylinder that would initiate combustion. Thring (1989) continued the work of Najit and Foster in four stroke engines by examining the performance of this autoignition combustion system and gave it the name "Homogeneous Charge Compression Ignition" or HCCI. He operated with blended gasoline and mapped the HCCI operation regime as a function of the Air/Fuel equivalence ratio and external EGR rates. He suggested the strategy of using the HCCI combustion system in SI engines for partial load conditions.

Development of diesel-fueled HCCI only started after the mid nineties only and research work has progressed in two different fueling routes: 1) Premixed HCCI, in which the fuel is inducted or injected into the intake manifold; 2) In-cylinder HCCI, in which fuel is injected directly into cylinder either before TDC or after TDC so that mixing can be accomplished prior to autoignition. In the pioneering work of the premixed diesel HCCI, Ryan and Callahan (1996) used a port fuel injector to supply diesel fuel into the intake air stream and an air pre-heater was also used to initiate combustion. He studied this combustion system with a range of compression ratios from 7.5 to 17:1 with EGR (Ryan and Callahan, 1996). Further study in the same engine was done by Gray and Ryan (1997). They pointed out the issues of premature ignition, knocking and accumulation of liquid fuel on surfaces in the intake system. They conveyed that a compression ratio of 9 to 11:1 is the best for HCCI combustion. They experienced higher smoke levels at low intake air temperatures (below 130°C), and revealed that this was due to combustion occurring in scattered diffusion flames around large droplets. The fuel consumption was higher by 28% over the normal CI mode due to non-optimal combustion phasing. Subsequent studies by other investigators have also applied premixed fueling to HCCI. Christensen *et al.* (1998) investigated diesel-fueled HCCI with port fueling as part of an investigation of variable compression ratio for controlling HCCI with various fuel types. Their results had an agreement with Gray and Ryan (1997). Smoke emissions were significant at some conditions. These trends were thought to be due to poor vaporization of the diesel fuel creating an inhomogeneous mixture. Kaneko *et al.* (2002) achieved premixed diesel-fueled HCCI with the help of a pintle-nozzle injection at the manifold (120 bar injection pressure). They experienced higher HC emissions, lubricant dilution and notable smoke emissions. All the problems were eliminated when they switched from diesel to light naptha fuel. The limitations have largely been due to the inability to control initiation and rate of combustion over the entire speed-load range. Choi *et al.* (2004) have concluded that the trend of SOC is sensitive to intake temperature, EGR rate and cylinder-to-cylinder variations. Sato *et al.* (2006) and Lu *et al.* (2007) have used the combination of high octane fuel (methane, RON90) and high-Cetane fuels (DME & DTBP) to control the high temperature reaction. Iijima *et al.* (2007) have studied the effect of high-Octane fuel and internal EGR on ignition initiation timing and they have concluded that high-Octane fuels are very sensitive to SOC than the internal EGR.

Diesel-fueled HCCI with in-cylinder injection has the advantages such as fuel accumulation in the intake manifold system and less chance of lubricant dilution when compared to premix diesel-fueled HCCI. However, the main disadvantage is that little time is available for fuel-air mixing. NOx and smoke emissions can be significant without proper mixing (Yokato *et al.*, 1998). Hino Motors had also done experiments with in-cylinder diesel-fueled HCCI. They used the strategy of split injection: early injection followed by injection near TDC. They faced the problem of high HC emissions, so a special injector nozzle tip (0.1 mm hole diameter, 30 holes at different angles) was used. They concluded that two injector tips were needed for early injection and near TDC injection (Akira *et al.*, 1998). Toyota Motor Corp (UNIBUS: Uniform Bulky Combustion System) (Yoshinori *et al.*, 1999) and IFP (NADI: Narrow Angle Direct Injection) (Bruno Walter *et al.*, 2002) also followed this strategy to achieve in-cylinder diesel-fueled HCCI with a common rail injection system. Nissan Motors achieved in-cylinder HCCI with the late injection strategy. The principles of this late-injection HCCI process are described by Kimura *et al.* (1999). The injection timings are 7° bTDC to 5° aTDC with high levels of EGR. Rapid mixing was achieved by combining high swirl with the toroidal combustion-bowl geometry.

In the internal combustion engines laboratory at the Indian Institute of Technology Madras, India; the authors have recently been involved in the study of HCCI engines with a view to develop combustion systems that can combine HCCI with conventional diesel or dual fuel combustion. The results presented here are from the first phase of this work with diesel as the fuel. The fuel was injected directly into the cylinder at a very early stage, i.e. at the start of suction, by modifying the injection system in order to allow sufficient time for as homogeneous a mixture as possible to be produced. Injection early in the suction stroke means that the fuel will enter the hot exhaust gas trapped in the clearance volume, which will aid vaporization. Five injection timings from TDC to 20°CA aTDC (during suction stroke) have been applied to study the combustion, performance and emission characteristics of the engine.

2. EXPERIMENTAL SETUP

The engine specifications are given in Table 1. The base engine used in this work is a single cylinder, water-cooled, DI, diesel engine which was modified to operate on HCCI mode. There were no changes in the combustion chamber geometry to convert it from CI to HCCI mode. However, a separate inline injection system run by the cam shaft, which could be set at any injection timing desired, was developed and used.

The required injection timings were set with the help of an external mechanical arrangement. This system will

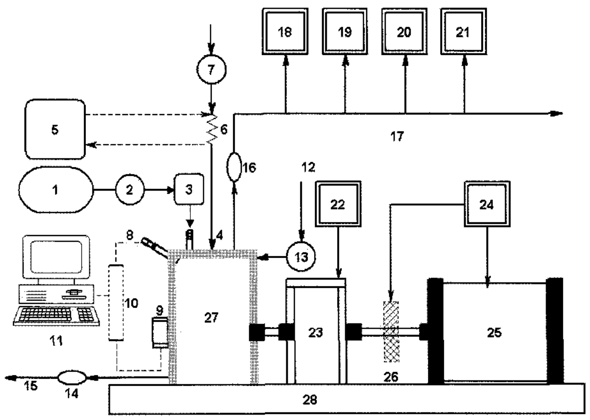

Figure 1. Schematic of experimental set up.

1. Diesel fuel tank
2. Fuel measurement system
3. In-cylinder injection system
4. Intake manifold
5-6: Heater system
7. Air flow measurement
8. Pressure transducer
9. Crank angle encoder
10. Data acquisition system
11. Personal computer
12. Cooling water inlet
13. Water flow measurement
14. Cooling water temperature measurement
15. Cooling water outlet
16. Exhaust gas temperature measurement
17. Exhaust manifold
18-21: Exhaust gas analyzers
22. Dynamometer controller
23. Eddy current dynamometer
24. Clutch controller
25. Electric motor
26. Electromagnetic clutch
27. Engine
28. Engine bed

Table 1. Engine specifications.

Bore × Stroke	80 × 110 mm
Connecting rod length	231 mm
Compression ratio	16:1
Rated output	3.7 kW @ 1500 rpm
Displacement volume	0.000553 m^3
Injector N.O.P	220 bar

be changed to a common rail system in due course. The fuel injection system was driven by the engine camshaft by a toothed gear arrangement. A needle-lift sensor was used to find the actual start of the injection. An electric heater was installed in the intake manifold to pre-heat the intake air if needed. The pre-heating system was only used to start and warm up the engine in HCCI mode operation. An electrical rheostat was used to control the supply of electricity to the heater to attain different heating coil temperature. All the results presented in this paper are with the heater switched off. An eddy current dynamometer was connected to the engine and an electric motor was connected to dynamometer at the other end through an electromagnetic clutch to engage and disengage it. This was used to start as well as to motor the engine.

Intake air and exhaust gas temperatures were measured by thermocouples and the coolant water outlet temperature was measured by using RTD. Provisions were made to measure the flow rates of the fuel, air and cooling water. An optical-encoder was used to record the crank angle signals and a flush-mounted, water cooled, piezo-electric pressure transducer was used to measure in-cylinder pressure history. During experiments, in-cylinder pressure data was acquired by using a PC-based high-speed data acquisition system for 100 consecutive cycles. The heat release rate was calculated from the cylinder gas pressure history (Brunt *et al.*, 1998). FID for HC, NDIR analyzer for CO, Chemilumiscent analyzer for NOx and Bosch smoke meter for smoke emissions were used for the measurement.

2.1. Experiment Procedure

First, the engine was started in HCCI mode by adjusting intake air temperature to 100°C. Once the engine was started, the load was increased to 2.15 bar BMEP and the intake air temperature was reduced to 35°C. All of the experiments were done at a constant engine speed of 1500 rpm and constant intake air temperature of 35°C. In these operating conditions, the engine was operating in the HCCI mode in the range of 2.15 to 4.32 bar BMEP. For different operating loads in HCCI mode, different injection timings were tried. Once the operating condition at a particular BMEP and injection timing was

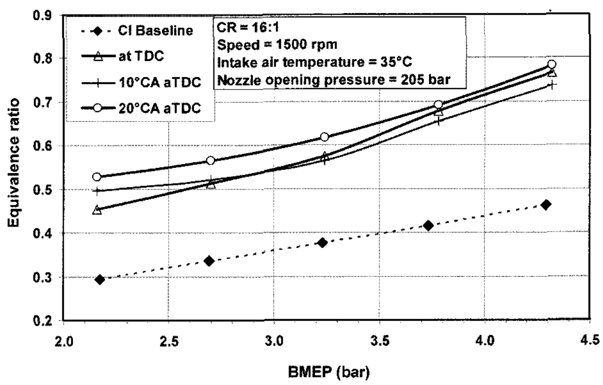

Figure 2. Variation of equivalence ratio with BMEP.

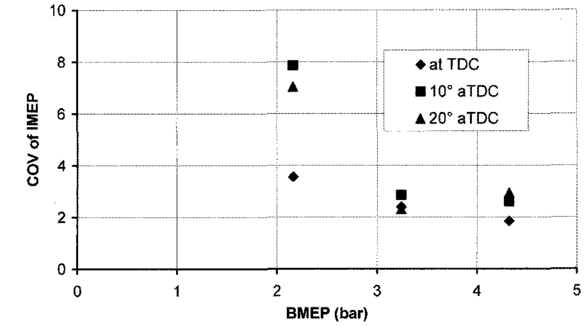

Figure 3. Coefficient of variation of IMEP with BMEP.

steady, different readings were noted down and performance, emission and combustion characteristics were evaluated.

3. RESULTS AND DISCUSSION

3.1. Equivalence Ratio and BMEP

The effect of BMEP and injection timing on the equivalence ratio is shown in Figure 2. Different injection timings ranging from 5°CA bTDC to 20°CA aTDC were tried. Since the operation was poor at 5°CA bTDC, those results are not presented. It is seen that only in the BMEP range of 2.15 to 4.32 bar, the engine could be operated in the HCCI mode without significant cycle-by-cycle variations, which is shown in Figure 3. This range corresponds to equivalence ratios varying from 0.45 to 0.78. Below an equivalence ratio of 0.45, the engine could not be run in the HCCI mode due to the leanness and low heat release rates. Similarly, beyond an equivalence ratio of 0.78, rapid pressure rise leading to abnormal combustion or knocking limits the operation in the HCCI mode. Also from Figure 2, it can be observed that the equivalence ratio at any given BMEP, in the case of the HCCI mode is much higher than that of the CI engine. Therefore, the thermal efficiencies are far lower. In order to produce a given brake power output, the present HCCI configuration needs more fuel compared to that of the CI mode.

3.2. Brake Thermal Efficiency

Figure 4 shows the variation of brake thermal efficiency with BMEP at different injection timings. In the HCCI mode with direct in-cylinder injection using a conventional injector, the brake thermal efficiency is much lower than that of the CI engine. This is mainly because in the HCCI mode with direct injection, the ranges of injection timings suitable were all close to TDC. This means that the fuel was injected during the beginning of the suction stroke where the gas pressure was low. Hence, the penetration of the fuel was very high and it hit the combustion chamber walls. During the injection period, which typically lasts for more than 30 degrees of CA, the fuel was also likely to hit the cylinder walls and the piston crown as the piston descends. Due to this, evaporation of the fuel was incomplete and delayed. Hence, only a part of the fuel was probably prepared to burn in the premixed mode and a considerable part was participating in the diffusion phase of the combustion later. This is evident from the heat release rate shown later.

Thus, in the early injection mode used in this work, which is called as HCCI mode; all the fuel was not completely prepared to form a truly homogeneous mixture. From Figure 4, we find that at low BMEP in particular, earlier injection is better. This is because it will lead to injection of the fuel into the trapped hot exhaust before it is cooled by the incoming fresh air. Almost all the injection timings considered show a similar trend and are in a band of brake thermal efficiency of 2% from low to high BMEP. Injection timing before TDC was poor probably because some of the fuel was likely to escape with the exhaust gas.

3.3. Cylinder Gas Pressures

The HCCI process is chemically controlled. The homogeneous mixture is compressed until ignition occurs simultaneously at multiple spots across the combustion chamber. HCCI combustion can be divided into low temperature and high temperature reaction phases. Low temperature reactions start at approximately 700 K and were described by Semenov (Glassman, 1996). The low temperature reactions are clearly distinguished in Figure 5. The high temperature reactions start at around a temperature of 800 K. It should be noted that the temperature curve represented in the Figure 5 is the average estimated gas temperature rather than the local temperature.

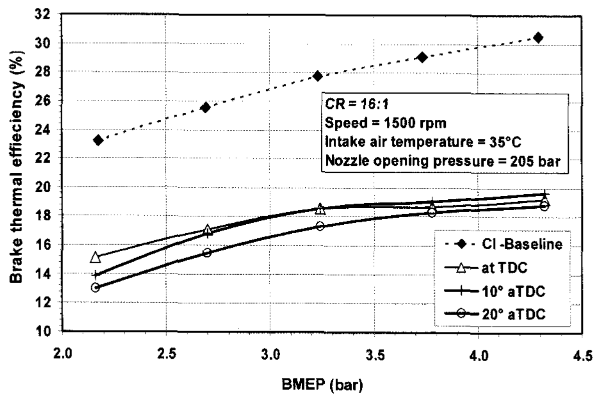

Figure 4. Variation of brake thermal efficiency with BMEP.

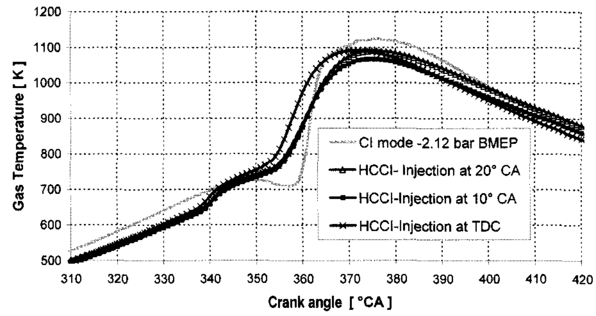

Figure 5. Variation of cylinder gas temperature for HCCI and CI mode.

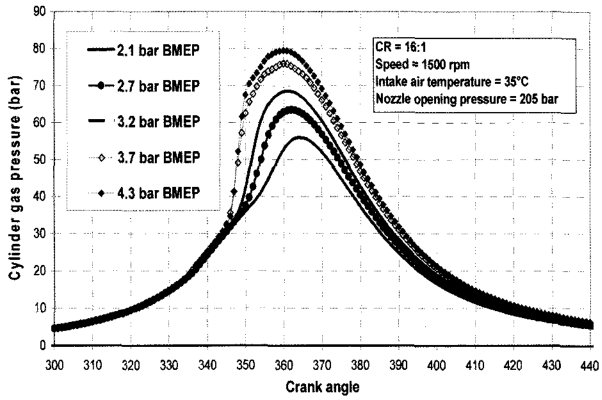

Figure 6. Variation of cylinder gas pressure with BMEP for injection at TDC (HCCI).

A close examination of the cylinder pressure indicates a difference between the HCCI and diesel combustion modes from crank angles of about 335 degrees. This is firstly due to the initiation of the cool flame at this crank angle. The cylinder pressure curve becomes steeper for the case of HCCI. The main heat release also starts earlier.

The cylinder pressure versus crank angle variation at different BMEP is shown in Figure 6 and variation of peak cylinder gas pressure is shown in Figure 7.

From Figure 6, it can be observed that, as the BMEP increases the peak pressure and rate of pressure rise,

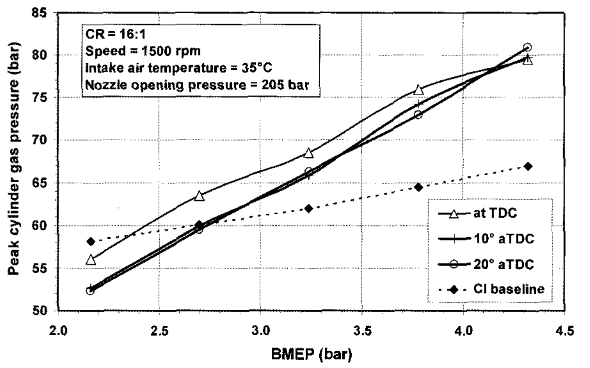

Figure 7. Variation of peak cylinder gas pressure with BMEP.

increase and angle of occurrence of peak pressure becomes more advanced. This is because with increase in BMEP, the cylinder wall and retained gas temperatures increase, causing the rise of charge temperature during compression.

From Figure 7, it is observed that, except at very low BMEP, the peak pressure is higher in the HCCI mode as compared to the diesel mode. In addition, because of early occurrence of the peak pressure, which is close to or even before TDC, the work output produced by a HCCI engine is less than that of the CI engine even with higher equivalence ratios (higher fuel input). This is because, in HCCI mode, the early rise in cylinder pressure leads to increased compression work. This problem can be overcome by properly phasing the combustion process. This can be done by varying the intake charge temperature or its composition by adding inert gases etc. These are being investigated and do not form the part of the present investigations.

The variation of maximum rate of pressure rise with BMEP for different injection timings is shown in Figures 8. Higher rate of pressure rise resulted in knocking or abnormal combustion at high BMEP. We also found that at low BMEP, where combustion in the HCCI mode was

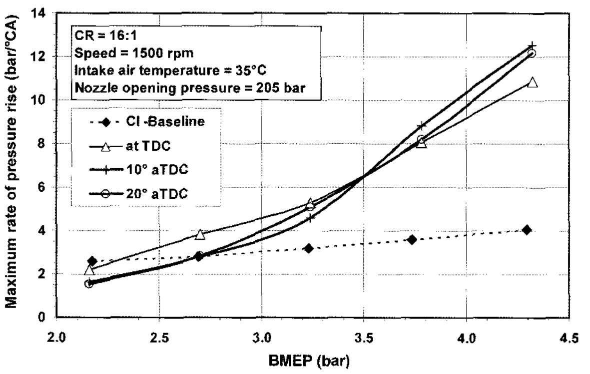

Figure 8. Variation of maximum rate of pressure rise with BMEP.

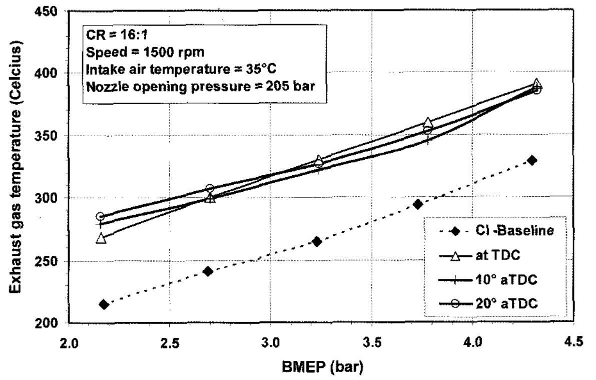

Figure 9. Variation of exhaust gas temperature with BMEP.

not proper due to low cylinder gas temperature; injection at TDC led to the highest cylinder pressure and rate of pressure rise. It can be noted that this injection timing was found to be better than the others at low outputs as far as brake thermal efficiency is concerned.

3.4. Exhaust Gas Temperature

In general, it is observed that in the HCCI mode, exhaust gas temperature (EGT) is higher than that of the CI engine for the entire range of BMEP and injection timings considered. As will be shown later, the heat release rate occurs until late in the expansion stroke in the present engine configuration. This happens even though the initial rate of heat release is quite high. This is because a considerable amount of diesel is deposited on the combustion chamber walls and burns later. Thus, not all the injected fuel was not premixed. Injection timing at 10° aTDC for which brake thermal efficiency was better than other timings generally resulted in the lowest EGT as shown in Figure 9.

3.5. Smoke Emission

Figure 10 shows the variation of the smoke emissions with BMEP and injection timing. It can be seen that smoke emissions are very high in the HCCI mode compared to that of the CI mode for the entire range of BMEP. This is mainly due to the high equivalence ratio on account of the low thermal efficiency. Fuel that is deposited on the walls also does not mix well with the air and leads to increased smoke emission.

Thus, in this configuration, though the fuel is injected early in the intake stroke, the mixture is not completely homogeneous. In fact, if the charge is mixed homogeneously, the smoke level will be negligible. Thus, in the present configuration, the lack of proper mixture preparation leads to this problem. Smoke values are lowest with the injection timing of 10° aTDC as expected as it results in the best thermal efficiency. Injection at

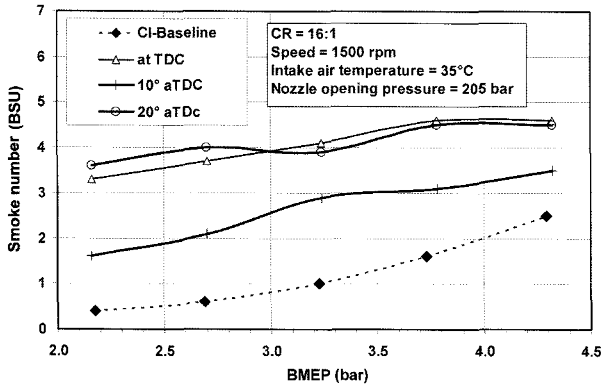

Figure 10. Variation of smoke emission with BMEP.

TDC will lead to a lot of fuel being deposited in the bowl of the piston. Late injection timings will mean that the fuel is introduced into air at a low temperature that will affect vaporization. It may be noted that when the injection starts late, a higher amount of air would have entered the cylinder and mixed with the retained high temperature exhaust. Improving mixture formation with high pressure injection, proper air movement or using gaseous fuels that can readily form mixtures will be beneficial.

3.6. NOx Emissions

From Figure 11 it is observed that for the entire range of BMEP at all injection timings, NOx emissions for HCCI mode are generally lower than that of the CI mode. As compared to the CI mode, the gas temperatures reached will be lower in the HCCI mode as the mixture is more homogeneous and thus lean everywhere. In the case of the CI mode, the gas temperature in certain locations will be much higher than in the case of the HCCI mode due to stratification of the mixture. The level of NO is in the range of 10 to 180 ppm between BMEP of 2.15 and 3.8 bar whereas in CI mode, it is in between 260 and 480 ppm. As will be shown later, combustion gets too much advanced in the HCCI mode and its rate also becomes

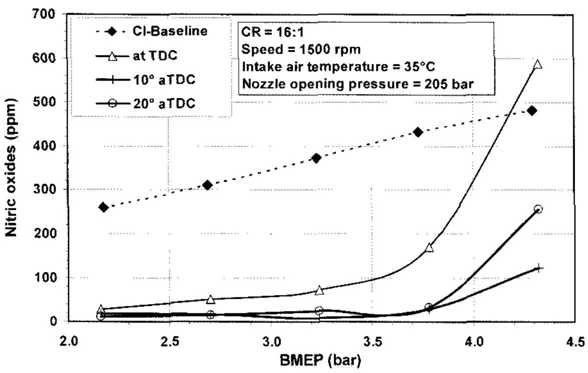

Figure 11. Variation of NOx emission with BMEP.

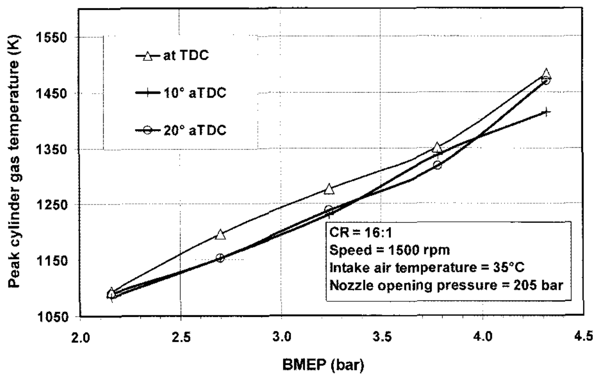

Figure 12. Variation of peak cylinder gas temperature with BMEP.

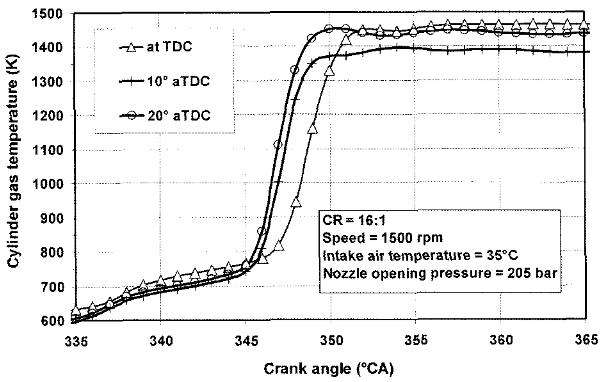

Figure 13. Variation of cylinder gas temperature at 4.32 bar BMEP.

very high at high BMEP. This is the reason for the high NO emissions at these conditions. The higher gas temperatures at the time of injection which lead to better vaporization of the injected fuel may be the reason for the higher NO emissions in the case of injection at 0° aTDC. The cylinder gas temperatures calculated from the pressure crank angle data indicate this trend, which is clearly seen in Figures 12 and 13. The peak gas temperature was calculated by assuming that, at the start of suction stroke, the cylinder contents were exhaust gas at the temperature of the exhaust manifold, which then mixed with the air that was inducted adiabatically. The temperature during the closed period of the cycle was determined using the ideal gas law. It was seen that the peak cylinder gas temperature was higher for injection at TDC. This certainly influences NO emissions.

3.7. HC and CO Emissions

Figure 14 shows the variation of CO emission with BMEP and injection timings considered. From Figure 14, it is observed that CO emissions are higher in the case of the HCCI mode than that of CI mode for the entire range of the BMEP and injection timings. Normally in the

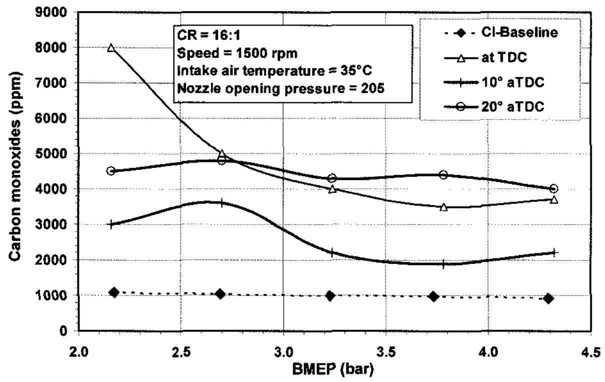

Figure 14. Variation of CO emission with BMEP.

conventional diesel engines, CO emissions are very low due to the overall lean mixture operation. In the present case, the equivalence ratios are higher than the CI mode and could lead to increased CO emissions. Furthermore, the CO level is higher at low loads in contrast to what happens in the case of standard diesel engines. This may be because at low BMEP, the overall wall temperatures are low and hence the fuel deposited on the walls does not vaporize well and leads to poor mixture formation and CO emissions. At high loads, overall wall temperatures are high and this assists evaporation and enhances fuel distribution in the combustion chamber, resulting in better oxidation of the fuel. In the HCCI mode, the injection timing of 10° aTDC gives lowest emission of CO than other injection timings.

Figure 15 shows the variation of HC emissions. Here again, the HC emission levels are higher for HCCI mode than that of CI mode. The trend of HC emission follows that of equivalence ratio. In this case also, poor mixture formation seems to be the main reason. Too early injection may also cause some of the injected fuel to escape through the exhaust valve during the overlap period.

3.8. Heat Release Rate

Figures 16 to 18 show the heat release rate at different

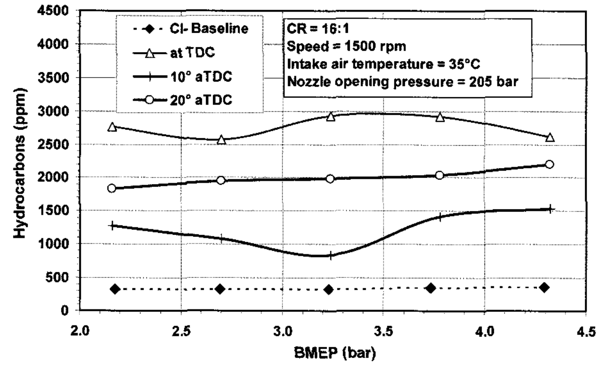

Figure 15. Variation of HC emission with BMEP.

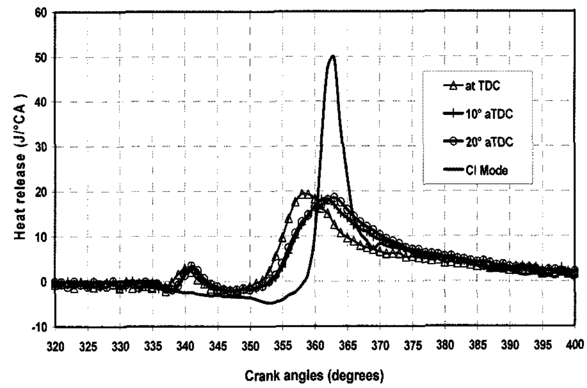

Figure 16. Variation of HRR at 2.15 bar BMEP.

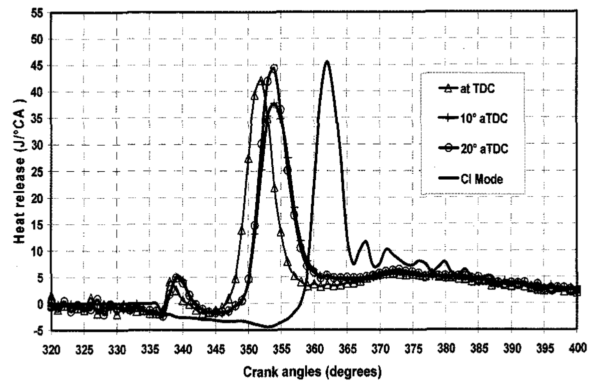

Figure 17. Variation of HRR at 3.24 bar BMEP.

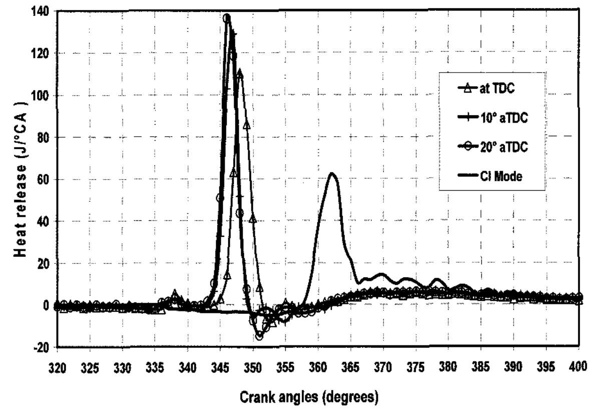

Figure 18. Variation of HRR at 4.32 bar BMEP.

BMEP and injection timings. Figure 19 shows the variation of heat release rate with BMEP for the case of injection at TDC. In the HCCI mode, it is seen that there are three heat release phases. This is particularly clear at the high BMEP. The first phase is due to the cool flame. The second phase is the main combustion in the homogeneous mixture. The third phase may be due to the slow burning of the fuel deposited on the piston and cylinder head surfaces during later stages. It was observed that at

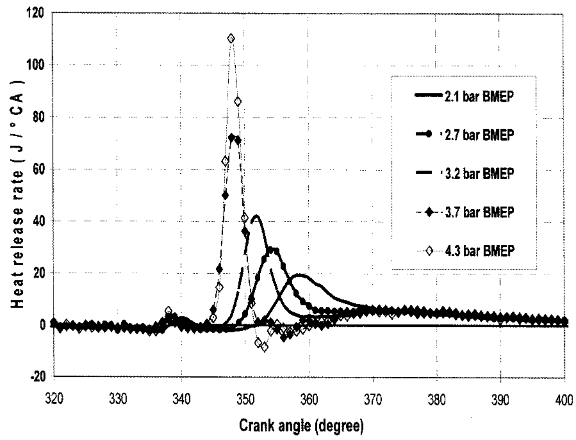

Figure 19. Variation of heat release rate with BMEP for injection at TDC (HCCI).

the BMEP of 2.15 bar, the heat release rates are much lower than that of CI mode due to the low gas temperatures encountered. Here the injection timing of 0° aTDC leads to very early combustion as the gas temperature at the time of injection is high. In addition, the same trend continues at a BMEP of 3.24 bar.

At a BMEP of 4.32 bar, the heat release rate in the HCCI mode is much higher than that of CI mode indicating knocking combustion. When the comparison was made for heat release rates at the three different conditions of BMEP, the reason for the 30% penalty in brake thermal efficiency in HCCI mode was evident.

At all BMEP, the heat release rate in the HCCI mode occurs too early. At 4.32 bar BMEP, in HCCI mode, even though the amount of energy released and heat release rate was high compared to the CI mode, the combustion was too much advanced to produce useful work.

For a BMEP of 2.15 and 3.24 bar, cool flame reactions are starting earlier for injection at TDC than that of other two injection timings. This leads to an advance in the start of main combustion. This may be due to the trapped exhaust gases in the cylinder as explained earlier.

At a BMEP of 4.32 bar, cool flame reactions are starting early with the injection timings of 20° and 10°CA aTDC, leading to early start of main combustion. This is expected because the injected fuel droplets are exposed to the hot top surface of the piston due to delayed injection at higher engine output. In the case of late injection timings, injection of fuel occurs as the piston descends down considerably from TDC. This can lead to fuel impingement on the cylinder walls and top of the piston, particularly when the injection durations are high which occurs at high out puts.

3.9. Start of Combustion

Crank angle position at the start of cool flame and main

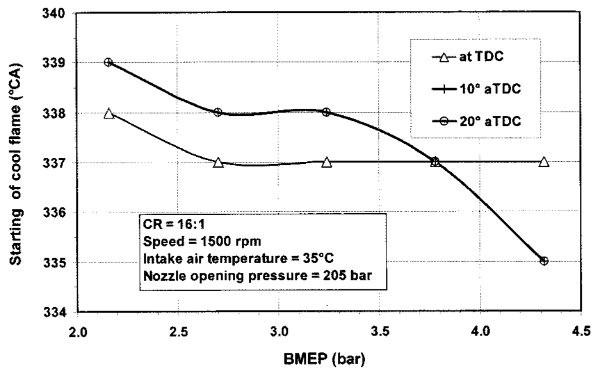

Figure 20. Variation of start of cool flame with BMEP.

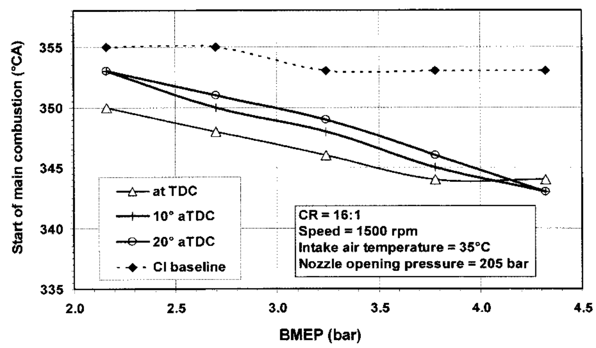

Figure 21. Variation of start of combustion with BMEP.

combustion are shown in Figures 20 and 21. Start of the main combustion (SOC) is defined as the crank angle at which the heat release curve just after the cool flame crosses the zero-line and reaches a positive value. It is seen that the SOC is early when the cool flame starts early. The ignition delay is also lowest for injection at TDC as shown in Figure 22. The heat release shown earlier has two regions after the cool flame. The first one is significant and may be because of the combustion of the premixed fuel and air. The second one is thought to be due to the slow burning of the deposited fuel on the combustion chamber walls. The duration of the premixed

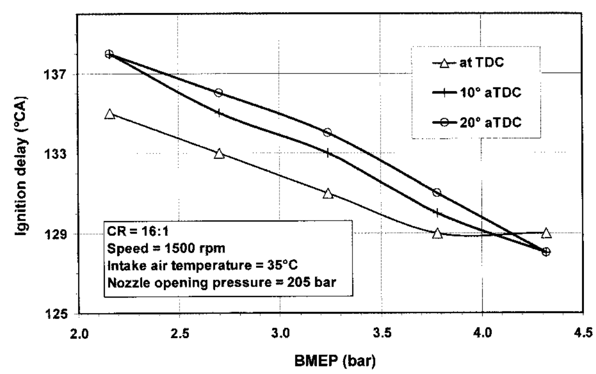

Figure 22. Variation of ignition delay with BMEP.

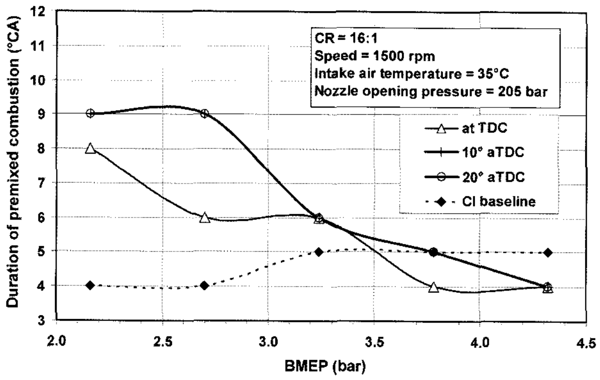

Figure 23. Variation of duration of premixed combustion phase with BMEP.

combustion phase is defined as the crank angle in degrees from start of combustion to the peak value of the heat release rate.

Figure 23 shows the variation of duration of premixed combustion phase with BMEP and injection timings for HCCI mode. It can be observed that the duration of the

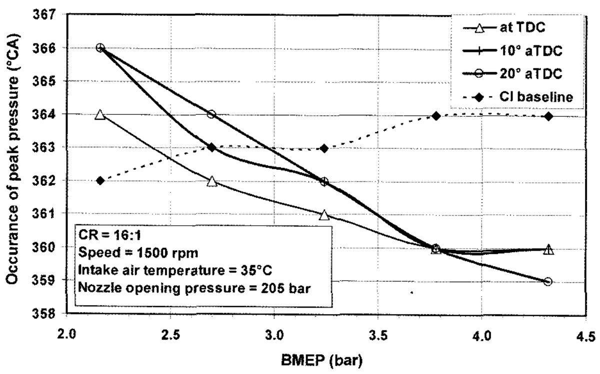

Figure 24. Variation of occurrence of peak pressure with BMEP.

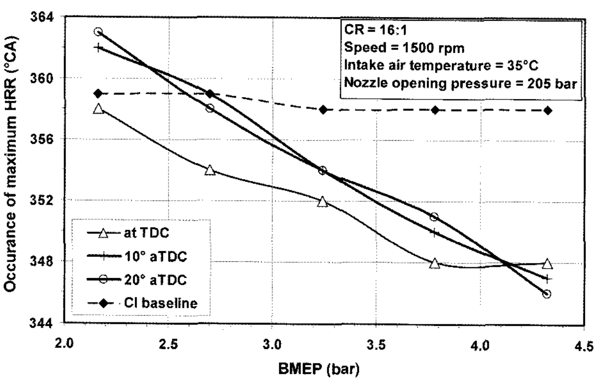

Figure 25. Variation of occurrence of maximum heat release rate with BMEP.

premixed combustion phase is high at low BMEP and low at the high BMEP conditions. This may be due to the increase in the temperature of the trapped charge and combustion chamber walls.

Figures 24 and 25 show the variation of occurrence of the peak pressure and maximum heat release rate. It is observed that for HCCI mode at low BMEP, the occurrence of peak pressure takes place late. At high BMEP, it is early due to reduced ignition delay. In addition, delayed injection delays the occurrence of the peak pressure. The trends of peak heat release rate are also similar. Good vaporization with injection at TDC leads to early occurrence of peak pressure and peak heat release rate.

4. CONCLUSIONS

Based on this preliminary work on running a diesel engine in HCCI mode with very early injection into the cylinder the following conclusions are drawn:

Though it is possible to run the engine in early injection HCCI mode, the thermal efficiency is lower by about 30% and the HC emissions are also considerably higher than the CI mode. Brake thermal efficiency is affected by slow combustion at low outputs and early combustion phasing at high outputs. A sizable quantity of the injected fuel is deposited on the bowl and on the piston depending on the injection timing, which leads to poor vaporization and inhibits homogeneous mixture formation. This also is the reason for higher levels of HC and smoke emissions. The injection timing of 10° aTDC was found to be the most suitable.

The range of BMEP at which the engine could be operated is 2.7 to 3.7 bar. At high BMEP, the problem was too rapid combustion where as at low BMEP, the heat release rates were too slow.

NOx emissions are very low in the HCCI mode. At lower output and medium output conditions, the NO level is about 20 to 30 ppm, whereas in CI mode, it is about 250 to 380 ppm. At higher outputs, rapid and too advanced combustion elevates NO levels.

The heat release rate clearly indicates the occurrence of a zone of cool flame followed by the zone of main heat release. The start of main combustion was influenced by the start of the cool flame.

On the whole, it is seen that avoiding wall wetting and proper combustion phasing are the important requirements to be met for HCCI engine operation. A proper injection spray profile and suitable methods to retard the combustion process at high outputs are needed.

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