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Applied Energy 88 (2011) 1169–1180

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Experimental study of combustion and emission characteristics of ethanol fuelled port injected homogeneous charge compression ignition (HCCI) combustion engine

Rakesh Kumar Maurya, Avinash Kumar Agarwal *

Engine Research Laboratory, Department of Mechanical Engineering, Indian Institute of Technology Kanpur, Kanpur 208 016, India

A R T I C L E I N F O

Article history:
Received 14 April 2010
Received in revised form 3 August 2010
Accepted 19 September 2010
Available online 29 October 2010

Keywords:
Combustion
Emission
Homogeneous charge compression ignition (HCCI)
Controlled auto-ignition
Ethanol
Port fuel injection (PFI)

A B S T R A C T

The homogeneous charge compression ignition (HCCI) is an alternative combustion concept for in reciprocating engines. The HCCI combustion engine offers significant benefits in terms of its high efficiency and ultra low emissions. In this investigation, port injection technique is used for preparing homogeneous charge. The combustion and emission characteristics of a HCCI engine fuelled with ethanol were investigated on a modified two-cylinder, four-stroke engine. The experiment is conducted with varying intake air temperature (120–150 °C) and at different air–fuel ratios, for which stable HCCI combustion is achieved. In-cylinder pressure, heat release analysis and exhaust emission measurements were employed for combustion diagnostics. In this study, effect of intake air temperature on combustion parameters, thermal efficiency, combustion efficiency and emissions in HCCI combustion engine is analyzed and discussed in detail. The experimental results indicate that the air–fuel ratio and intake air temperature have significant effect on the maximum in-cylinder pressure and its position, gas exchange efficiency, thermal efficiency, combustion efficiency, maximum rate of pressure rise and the heat release rate. Results show that for all stable operation points, NO_x emissions are lower than 10 ppm however HC and CO emissions are higher.

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1. Introduction

Automotive industry has been strongly required to develop clean technologies of lower fuel consumption for ambient air quality improvement, green house gas reduction and energy security. As a result, fuels and engines used in transportation have to face two main challenges of improving fuel economy and reducing emissions in a highly competitive economy. Considering continuously stringent emission regulations, as well as increasing shortage of primary energy resources, the development of new highly efficient and environment friendly combustion systems, associated with alternative fuels becomes increasingly important and hence research need to be carried out in this domain. Many research programs are currently being undertaken in the area of developing alternative fuels and new combustion concepts.

To reduce the green house gases (mainly CO_2), renewable sources of engine fuels are explored for research. Currently for the selection of alternate fuels, overall process is taken into account right from the fuel production to the vehicle emissions and a "well-to-wheel" balance has to be analyzed for each technology. Ethanol seems to be emerging as a strong candidate fuel to comply with these new constraints: it is extracted from biomass and consequently has a good "well-to-wheel" CO_2 emission balance [1]. Bio-ethanol is net negative in CO_2 cycle and there are no net energy losses [2]. Ethanol is of interest as a partial future replacement of gasoline [3]. Using alternative fuels, such as ethanol and methanol in internal combustion engines has the potential to reduce the dependency on petroleum fuels. Ethanol is mainly produced from sugar products and methanol is mainly produced from natural gas and coal stocks [4–6].

Both, reducing exhaust emissions and increasing thermal efficiency in combustion engines are of equally great importance. There are basically three modes of combusting fuels in reciprocating engines. First, gasoline engine depend on flame propagation through a homogeneous fuel–air mixture. Secondly, diesel engines depend upon diffusion combustion. In a modern diesel engine, fuel is injected at a controlled rate to accomplish diffusion burning rate equal to injection rate [7]. The potential to significantly improve the fuel economy of a conventional spark ignition or compression ignition engine seems limited. Hence, third combustion concept is being developed, which is called homogeneous charge compression ignition (HCCI). HCCI means that the fuel and air should be mixed homogeneously before combustion starts and the mixture is auto-ignited due to increase in temperature at the end of the compression stroke [8]. Thus HCCI is similar to SI in the sense that

* Corresponding author. Tel.: +91 512 259 7982; fax: +91 512 259 7408.
E-mail address: akag@iitk.ac.in (A.K. Agarwal).

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doi:10.1016/j.apenergy.2010.09.015

Nomenclature

CAD	crank angle degree	θ	crank angle
CAI	controlled auto-ignition	$T(\theta)$	temperature at crank angle θ
CI	compression ignition	w.r.t.	with respect to
COV	coefficient of variation	COV_{IMEP}	coefficient of variation of IMEP
EGR	exhaust gas recirculation	$dP/d\theta_{max}$	maximum rate of pressure rise
EVO	exhaust valve opening	T_{max}	maximum mean gas temperature
HCCI	homogeneous charge compression ignition	Q_{in}	total heat values of introduced fuel per cycle
IMEP	indicated mean effective pressure	m_f	fuel mass per cycle
ISFC	indicated specific fuel consumption	q_{LHV}	lower heating value of the fuel
IVC	intake valve closing	λ	relative air/fuel ratio
PFI	port fuel injection	η_{ge}	gas exchange efficiency
ROHR	rate of heat release	η_{com}	combustion efficiency
SI	spark ignition	$\eta_{i,g}$	gross indicated thermal efficiency
SOC	start of combustion	σ	standard deviation
TDC	top dead center		

both engines use a premixed charge and HCCI is similar to CI as both rely on auto-ignition for combustion initiation.

The first study on HCCI was conducted on a two-stroke engine by Onishi et al. in 1979 [9]. They named the combustion process Active-Thermo Atmosphere Combustion (ATAC). It was observed that there is no flame propagation, like the one in a conventional SI engine. Instead, the whole mixture burns at the same time. Just after the presentation of Onishi, Noguchi et al. showed the same combustion process in an opposed piston two-stroke engine [10]. Noguchi also conducted measurements of radical concentration during combustion. In 1994, Iida showed that the possible operating conditions for stable two-stroke HCCI combustion could be significantly expanded by using methanol as fuel [11]. Najt and Foster showed in 1983 that it was possible to achieve HCCI in a four-stroke engine as well [12]. They used an engine with variable compression ratio which was operated on a mixture of isooctane and *n*-heptane.

HCCI was first tested on a real production engine by Stockinger et al. in 1992 [13]. Researchers used a standard 1.6 l engine, which was converted to HCCI operation with preheated intake air. The part load efficiency increased from 14% to 34%. Interesting research on HCCI was made by Aoyama et al. on HCCI and they named combustion process Premixed Charge Compression Ignition (PCCI) [14]. Researchers made comparisons between PCCI, diesel and gasoline direct injection (GDI). At optimum air–fuel ratio (λ), PCCI had the lowest fuel consumption. NO_x emissions were also much lower with PCCI, but HC emissions were higher.

The HCCI combustion process has been studied with certain success in two-stroke [9,15] and four-stroke engines [3,7,8,12,16–19], and with liquid [3,7–18] and gaseous [3,19] fuels. The HCCI family can be distinguished according to the fuel introduction strategy employed [21]. This distinction include port injection [12,16,20], early in-cylinder injection [22,23], late in-cylinder injection [24], and dual fuel introduction (both in-cylinder and port injection) [21].

The HCCI engine is being investigated worldwide because of its higher efficiency and ultra-low NO_x and PM emissions [25]. However, problems still need to be resolved, such as control of the ignition time and combustion over a wide range of engine speeds and loads, control of the HC and CO emissions under low loads and so on [26]. The main problem with the HCCI is that the ignition is completely controlled by chemical kinetics, and is therefore affected by the fuel composition, equivalence ratio, and thermodynamic state of the mixture [27]. There is no external control such as the fuel injection or spark timing that are used on diesel or SI engines. Achieving the required level of control during transient engine operation is even more challenging since charge temperatures have to be correctly matched to the operating condition during rapid transients with a high repeatability since the speed and load are changing.

A HCCI engine takes advantage of high compression ratio, similar to that of a diesel engine, un-throttling and lean operation. Engine heat transfer losses and the losses due to gas dissociation at the time around TDC can be smaller than those in either diesel or spark ignition engines. Meanwhile, gas temperature after HCCI combustion can be lower than 1800–1900 K due to high dilution of the charge by air or residuals. Thus, formation of NO_x in the combustion chamber is suppressed. Also, because soot formation during combustion in a premixed lean mixture is extremely low, HCCI technology reduces both NO_x and PM emissions simultaneously [28].

The most significant challenge of homogeneous charge compression ignition (HCCI) engines is the development of control strategies that allow effective HCCI operation over the largest possible operating regime of the engine. Extending the operating limits of HCCI is critical to realize the considerable efficiency and emission benefits that are hallmark of HCCI. Researchers are exploring several approaches including spark-assist [29–31], variable valve timing [28] and variable compression ratio [32] methods to achieve the goal of improved HCCI performance.

HCCI engines are inherently fuel flexible and can run on low or high grade fuels as long as the fuel can be heated to the point of auto-ignition [33]. In particular, HCCI engines can run on ethanol. Ethanol has high octane number indicating good anti-knock performance, high latent heat of vaporization allowing a denser fuel–air charge, and excellent lean-burn properties. But ethanol also has disadvantages such as low energy density and low cetane number, which make it difficult to use it in conventional CI engines. Ethanol has been widely used as fuel, mainly in Brazil, or as a gasoline additive for octane improver and better combustion in USA and Canada [34]. Ethanol has also been used as oxygen additive in diesel in order to depress the soot and NO_x emissions [35].

Alcohols (methanol and ethanol) have often been suggested as promising alternative fuels, and they are identified as having the potential to improve air quality when replacing the gasoline/diesel in the conventional IC engine. This potential is primarily due to the different organic species that are emitted by alcohol-fuelled engines, demonstrated by Jeffrey et al. [36]. Guerrieri et al. [37] investigated the effect of high percentage ethanol blends on vehicle's exhaust emissions. The results showed that HC and CO emissions decreased while NO_x and acetaldehyde emissions and fuel consumption increased as the ethanol content in the test fuel increased. Formaldehyde and CO_2 emissions were relatively unaffected by the addition of ethanol. Both ethanol and methanol exhibit good combustion characteristics in HCCI engine. Iida et al. [11] noted that methanol sustained much larger operating range of air/fuel ratio than gasoline in a two-stroke engine. Oakley et al. [38] also noted that alcohols have significantly higher tolerance to dilution than hydrocarbon fuels tested. Pure Ethanol HCCI combustion studies are performed on a single cylinder engine with port fuel injection and combustion emission analyzed by Zhang et al. and Hui et al. [39,40]. Following the same path, the authors performed similar experiments with higher compression ratio engine in order to improve further the engine efficiency.

It can be seen that HCCI combustion process is very attractive for its benefits in terms of thermal efficiency and emissions. HCCI combustion offers intrinsic fuel flexibility. To decrease the dependency on fossil fuels, it is necessary to explore the alternative fuels for new generation engines. This study is performed keeping in mind the development of new and highly efficient and environment friendly combustion systems using alternative fuels. In this paper, experimental investigations of HCCI combustion is done on modified engine running at 1500 rpm, using ethanol as fuel at different inlet air temperatures (120–150 °C).

2. Experimental setup

A two cylinder, four-stroke, air-cooled, naturally aspirated, bowl-shaped combustion chamber; direct injection diesel engine was modified for the present experiments. The specifications of unmodified engine are given in Table 1. For achieving the HCCI combustion, it is required to operate the engine cylinder at a particular speed and generate suitable conditions for HCCI combustion to happen. Most of the time, an AC dynamometer is required for motoring the engine at the desired speed. In the present study, AC dynamometer was not available therefore a new technique of using one cylinder of a two cylinder engine in conventional CI mode has been used so that HCCI can be achieved in the second cylinder. Second of the two cylinders of the engine is modified to operate in HCCI mode, while the first cylinder operates like an ordinary diesel engine, thus generating enough power to motor the second cylinder for achieving the HCCI combustion. A schematic diagram of the experimental setup is shown in Fig. 1.

A fuel premixing system was installed in the intake manifold. This system consists of electronic port fuel injector, an injection start timing (w.r.t. TDC) and injection duration controlling electronic circuit. Controlling circuit developed for this purpose is used to control the pulse width, which in turn triggers the fuel injector. Electronic fuel injector has four holes and the fuel is injected in the manifold at 3 bars fuel injection pressure. The fuel injection timing for the most port-injection SI engines is set to 'end of injection' before the intake valve opens so as to avoid the liquid fuel from entering into cylinder directly. The HCCI engine with port injection has a similar effect. Therefore the injection timing at TDC of compression stroke was chosen for the experiments. Fresh air entering the engine is heated by an air pre-heater positioned upstream of the intake manifold. The intake air heater is operated by a closed loop controller, which maintains constant intake air temperature as set by user for feed-back control. A thermocouple in conjunction with a digital temperature indicator was used in measuring the intake and exhaust gas temperature. An orifice-meter and a U-tube manometer were used to measure air consumption of the engine. A surge tank fixed on the inlet side of the engine maintains a constant air flow through the orifice-meter and eliminates cyclic fluctuations.

The in-cylinder pressure was measured using a water-cooled piezo-electric pressure transducer (6061B, Kistler) which is mounted flush with the cylinder head. The pressure transducer minimizes thermal shock error by using a double walled diaphragm and an integral water cooling system. To measure the crank angle degrees (CAD), an optical shaft encoder (ENC58/6-720ABZ/5-24V, Encoders India) is coupled with the crankshaft using a helical coupling. The cylinder pressure history data acquisition and combustion analysis is done using a LabVIEW based program.

In-cylinder pressure was recorded for 100 cycles at 0.5°CA (degrees crank angle) resolution and averaged to calculate the indicated mean effective pressure (IMEP), rate of heat release (ROHR), mean gas temperature, and other combustion related parameters.

Table 1
Detailed engine specifications.

Engine characteristics	Specification
Make/model	Indec/PH 2
Injection type	Direct injection
Number of cylinders	Two
Bore/stroke	87.3/110 mm
Power per cylinder	4.85 kW at 1500 rpm
Compression ratio	16.5
Total displacement	1318 cc
Fuel injection timing	24° before TDC
Fuel injection pressure	210 kg/cm^2 at 1500 rpm

2.1. Rate of heat release (ROHR)

ROHR is calculated from the acquired data using the zero dimensional heat release model [41]. Consequently, the main combustion parameters were extracted from the heat release and in-cylinder pressure curves. ROHR was calculated as

$$\frac{dQ(\theta)}{d\theta} = \left(\frac{1}{\gamma - 1}\right)V(\theta)\frac{dP(\theta)}{d\theta} + \left(\frac{\gamma}{\gamma - 1}\right)P(\theta)\frac{dV(\theta)}{d\theta}$$

The following assumptions were made in this calculation:

- Cylinder charge was considered to behave as an ideal gas.
- Distribution of thermodynamic properties inside the combustion chamber was considered to be uniform.
- Dissociation of combustion products was neglected.
- No variation of cylinder mass due to blow-by was considered.
- Heat transfer from the cylinder is neglected in this model.

2.2. Mean gas temperature

Mean gas temperature is calculated by assuming uniform temperature within the engine cylinder using ideal gas law [41]. The results are valid between intake valve closing (IVC) and exhaust valve opening (EVO). Mean gas temperature was calculated as:

$$T(\theta) = \frac{P(\theta)V(\theta)n(\theta)}{P_{\mathrm{IVC}}V_{\mathrm{IVC}}n_{\mathrm{IVC}}}T_{\mathrm{IVC}}$$

In this calculation, molar ratio is assumed to be unity.

2.3. Gas exchange efficiency (η_{ge})

Gas exchange efficiency is defined as the ratio between the indicated work during the complete cycle and the closed part of the cycle [42]. Gas exchange efficiency was calculated as:

$$\eta_{ge} = \frac{\mathrm{IMEP}_n}{\mathrm{IMEP}_g}$$

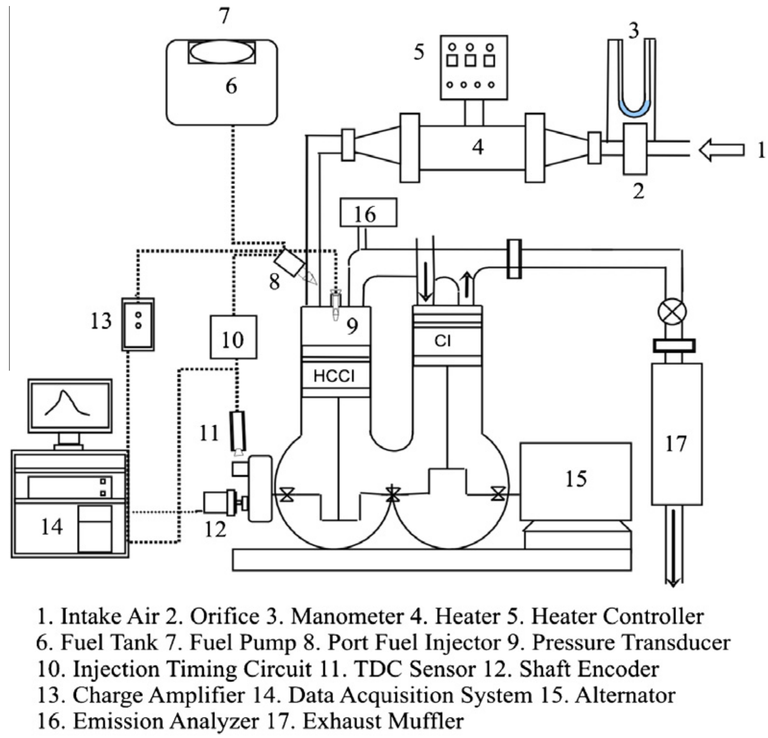


Fig. 1. Schematic diagram of experimental setup.

Combustion efficiency (η_{com}) was calculated from the following equation [43]:

$$\eta_{com} = \frac{\sum ROHR}{Q_{in}} \times 100$$

where $\sum ROHR$ is integrated value of heat release rate; Q_{in} is total heat values of introduced fuel.

The cooling losses by the convective, radiative and conductive heat transfer through the wall of combustion chamber were not considered.

Gross Indicated thermal efficiency is defined as the ratio between the work on the piston during the compression and expansion stroke ($W_{i,g}$) to the input fuel energy [32].

$\eta_{i,g} = \frac{W_{i,g}}{m_f q_{\mathrm{LHV}}}$ where m_f is fuel mass per cycle and q_{LHV} is the lower heating value of the fuel.

2.4. Combustion reaction speed

Defined as ratio of maximum rate of heat release and mass of fuel supplied [44].

$$v_c = \left(\frac{dQ}{dt}\right)_{\max} / m_{\mathrm{fuel}}$$

where v_c is combustion reaction speed J/ms/g; dQ/dt is the rate of heat release J/ms

Relative air–fuel ratio (λ) is the ratio of the actual air/fuel ratio to the stoichiometric air/fuel ratio. The results of this investigation are presented with respect to different relative air/fuel ratios (λ) present in the HCCI operating region. Experiments were conducted on the modified engine at constant speed of 1500 rpm and different intake air temperatures.

3. Results and discussion

In this section, the experimental results at different engine operating conditions at constant engine speed of 1500 rpm are presented with ethanol as fuel for intake air temperature in the range 120–150 °C. This temperature range is selected based on the engine performance and usable air–fuel ratio. At higher intake air temperature, richer mixture of fuel–air has advanced ignition timing and rate of pressure rise is also very high, which leads to knocking combustion. To avoid the knocking combustion, engine is operated at leaner mixture thus IMEP is lower. Considering these constraints, experiments are performed for intake air temperatures in the range of 120–150 °C.

3.1. Operating region

There are two operating limits that are relevant to HCCI engine operation; the so-called 'knock limit' when the pressure rise rate is

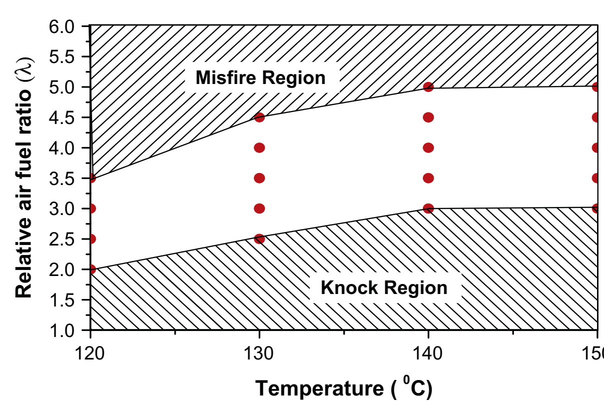
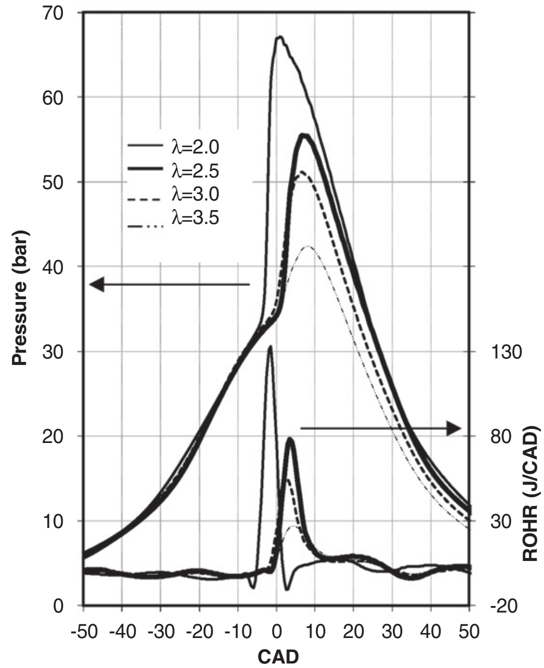
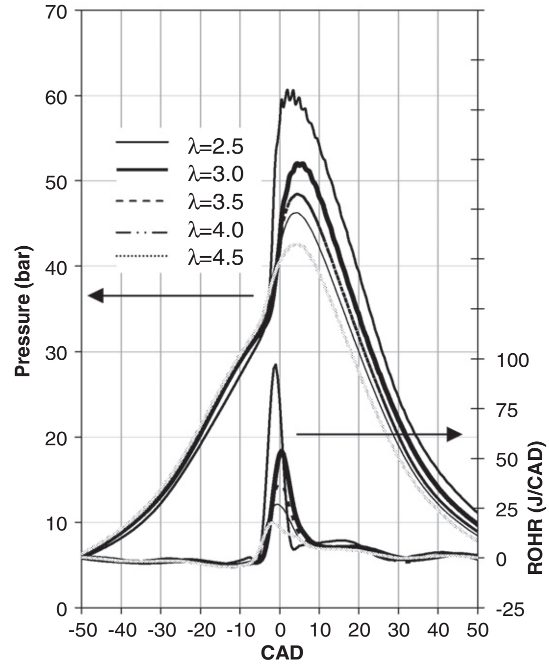


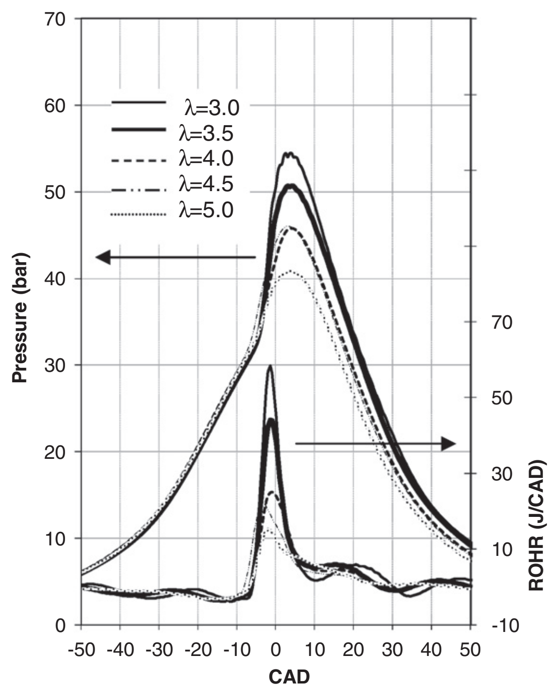
Fig. 2. HCCI stable operating range for ethanol.



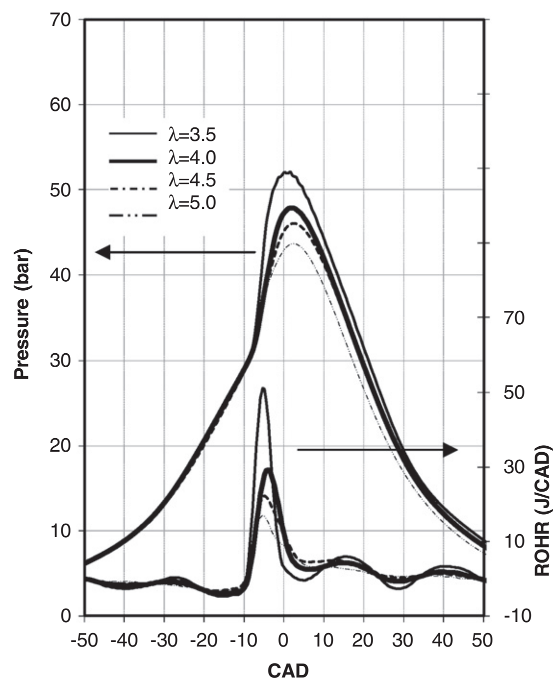
a. P-θ and rate of heat release for different relative air fuel ratios (λ) at intake air temperature of 120 °C



b. P-θ and rate of heat release for different relative air fuel ratios (λ) at intake air temperature of 130 °C



c. P-θ and rate of heat release for different relative air fuel ratios (λ) at intake air temperature of 140 °C



d. P-θ and rate of heat release for different relative air fuel ratios (λ) at intake air temperature of 150 °C

Fig. 3. *P–θ* and rate of heat release for ethanol fuelled HCCI combustion.

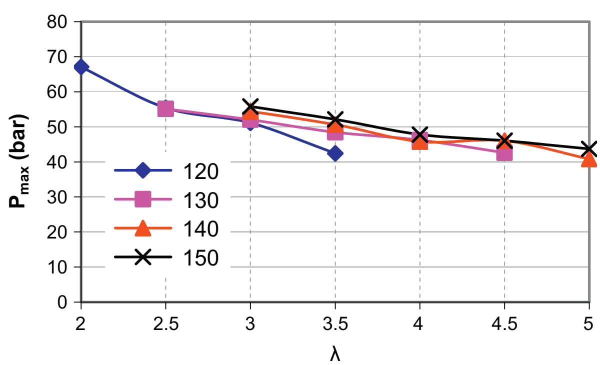
unacceptably high and the 'instability limit' or the 'misfire limit' when the cyclic variation is unacceptably high. Indeed the cause of excessive cyclic variation might be the failure of auto-ignition in some cycles [45]. To study the HCCI combustion, criteria as to what constitutes HCCI combustion must be defined. The first boundary defines the lower limit for the HCCI combustion. At low loads, fuel flow rate decreases hence the net heat release also decreases. It is believed that the resulting gradual reduction of average combustion temperature results in more unburned charge that is characterized by high CO and THC emissions and by increase in cycle-to-cycle variation. Cycle-to-cycle variation of the combustion process in an engine can be monitored by cylinder pressure transducer. Fluctuation of the indicated mean effective pressure (IMEP) is used as a measure of cycle-to-cycle variations and expressed as COV_{IMEP}. The Coefficient of Variation (COV) of IMEP for 100 consecutive engine cycles was calculated as standard deviation (σ) divided by mean value (IMEP) as a percentage [46].

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP} \times 100$$

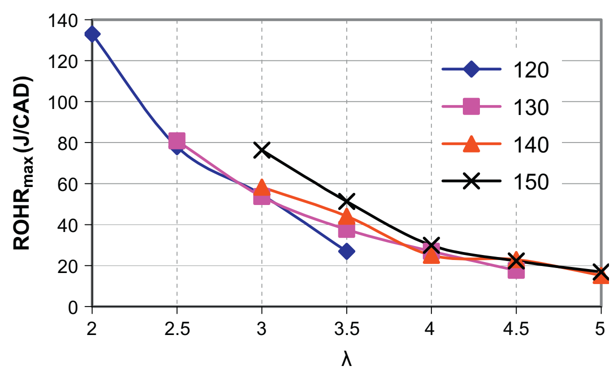
Since the drivability problems in automobiles normally arise when COV_{IMEP} exceeds 10% [41], so in this investigation this value is used for the misfire boundary condition for HCCI combustion criteria. At higher loads, when the fuelling rate is increased (i.e. lower λ), the HCCI combustion rate increases in intensity, and gradually causes unacceptable noise and may potentially cause engine damage. Therefore knocking combustion can be defined as being at the upper limit of HCCI combustion. In this investigation, the upper limit of HCCI combustion was defined as being when the rate of pressure rise in a cylinder exceed 1.0 MPa per crank angle degree ($dP/d\theta_{max}$ = 1.0 MPa/CAD) for each individual cycle [47].

Values of both COV_{IMEP} and $dP/d\theta_{max}$ are determined from the recorded combustion chamber pressure with respect to crank angle. By using the above HCCI operating region criteria, successful HCCI combustion region is found and shown in Fig. 2.

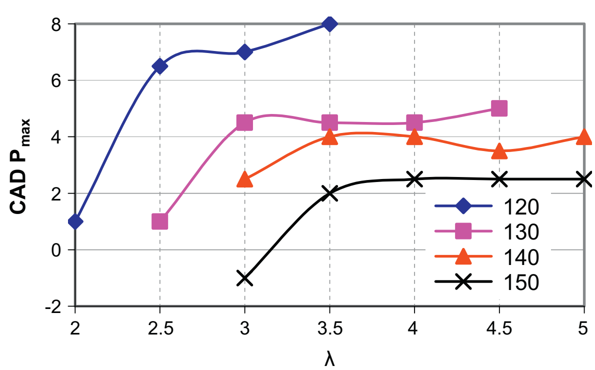
It was observed that for stable HCCI operating conditions, engine runs at richest mixture for low intake air temperature and at higher inlet air temperature, leaner mixtures can also be successfully auto-ignited. Fig. 2 shows that HCCI operating region was found for ethanol in the range of λ = 2.0–5.0 for different intake air temperatures used in the present investigations.



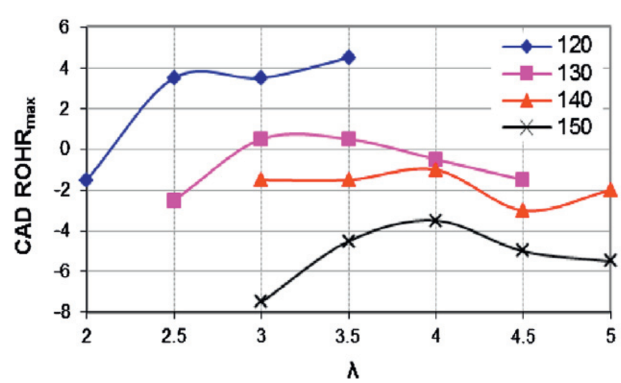
a. Maximum cylinder pressure for HCCI mode at different relative air fuel ratios (λ)



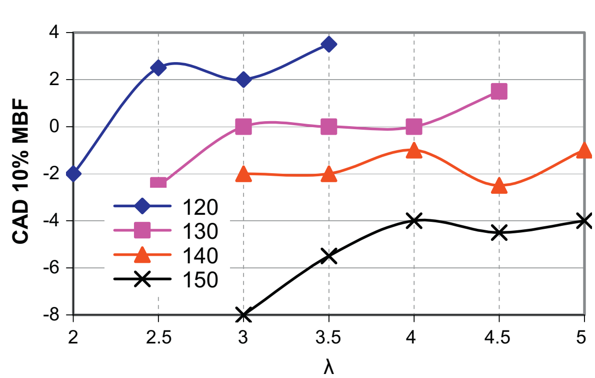
b. Maximum ROHR for HCCI mode at different relative air fuel ratios (λ)



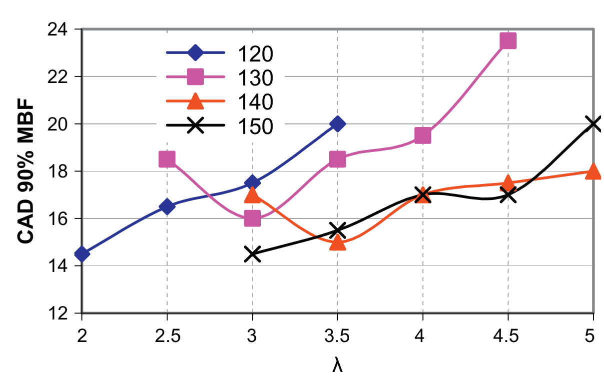
c. CAD Maximum cylinder pressure for HCCI mode at different relative air fuel ratios (λ)



d. CAD Maximum ROHR for HCCI mode at different relative air fuel ratios (λ)



e. CAD 10% MBF for HCCI mode at different relative air fuel ratios (λ)



f. CAD 90% MBF for HCCI mode at different relative air fuel ratios (λ)

Fig. 4. Cylinder pressure and heat release parameters with CA positions for stable HCCI combustion.

3.2. Cylinder pressure and rate of heat release

The cylinder pressure was measured using piezo-electric pressure transducer for all operating conditions and average of 100 cycles with a resolution of 0.5°CA is plotted for HCCI combustion of ethanol. Fig. 3 shows the in-cylinder pressure traces and rate of heat release for ethanol HCCI combustion at different relative air–fuel ratios (λ) and inlet air temperatures. For all plots, the trace with the highest maximum pressure corresponds to the operating condition with the richest mixture, as given by Fig. 3a–d, and the lowest maximum pressure corresponds to the leanest mixture at any inlet air temperature. It can be noticed from Fig. 3a–d that maximum pressure decreases with increase in λ as lesser amount of fuel is available for auto-ignition and as a result, it develops lower maximum pressure.

The cylinder pressure was analyzed using a single zone heat release model, which gives the rate of heat release. Details concerning the model are given in Section 2. ROHR is measure of how fast chemical energy of fuel is converted into the thermal energy by combustion. This directly affects rate of pressure rise and accordingly the power produced. The rate of heat release profile is plotted on the secondary axis of Fig. 3a–d for different engine operating conditions of stable HCCI combustion of ethanol. It is observed from Fig. 3a–d that for all plots, the trace with the highest maximum rate of heat release corresponds to the operating condition with the richest mixture and the lowest rate of heat release corresponds to the leanest mixture for each inlet air temperature. For richer mixture, large amount of fuel is auto-ignited at several locations in combustion chamber as compared to leaner mixture so combustion rate or rate of heat release is higher for the richest mixture at any intake air temperature.

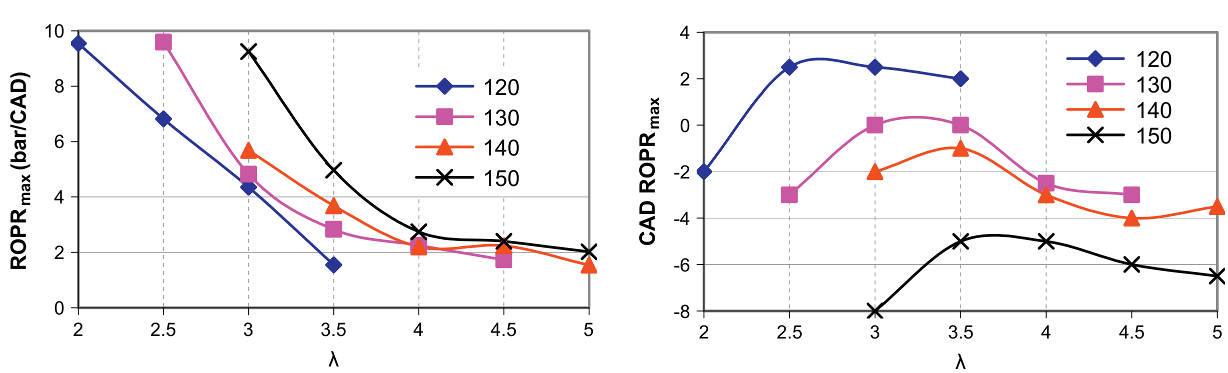
Fig. 4 shows detailed analysis of cylinder pressure and heat release parameters and their crank angle positions of ethanol HCCI combustion in stable engine operating conditions. It can be noticed from Fig. 4a that maximum pressure decreases with increase in λ as lesser amount of fuel is available for auto-ignition and as a result, it develops lower maximum pressure. It can also be observed that P_{max} values at different air temperatures are almost same for various relative air–fuel ratios. The maximum pressure is dependent on two factors namely, pressure rise due to combustion and change in the volume due to piston movement. It can be observed from Fig. 4c that maximum pressure position occurs after TDC for almost all cases. Since volume is increasing, the pressure have opposite effect due to change in volume. Due to combined effect of these two factors, maximum pressure is almost constant.

Fig. 4c shows the CAD corresponding to maximum cylinder pressure for ethanol HCCI combustion. It can be observed that crank angle position of maximum pressure increases after TDC because the fuel–air mixture becomes leaner and the rate of heat release is lower. Fig. 4b shows maximum rate of heat release at different engine operating conditions for ethanol HCCI combustion. It is noticed from these figures that maximum rate of heat release is highest corresponding to richest mixture and lowest for leaner mixture at constant intake air temperature since lower amount of fuel is ignited in the cylinder. Due to advanced ignition timing of the rich fuel/air mixture, the peak values of the heat release rate are very high for richer fuel–air mixture. It can be noticed from Fig. 4d that CAD corresponding of maximum ROHR is moving towards TDC as engine was operated with leaner mixtures at a constant intake air temperature.

Fig. 4e shows the crank angle position for 10% mass burn fraction of ethanol fuelled engine running in stable HCCI combustion mode at different engine operating conditions. This can be noted as start of combustion in the combustion chamber. It can be noticed from the figure that on increasing inlet air temperature, the combustion starts earlier at any relative air–fuel ratio as combustion chamber temperature increases. In HCCI combustion, the start of combustion depends on chemical kinetics, which is dependent on the pressure and temperature history inside the combustion chamber. The variation of crank angle position for 90% mass burn fraction at different engine operating conditions in HCCI mode is shown in Fig. 4f. The 90% mass burn fraction can be regarded as close to the end of combustion in the cylinder. It is observed that as mixture becomes leaner, the crank angle for 90% MBF moves away farther away after TDC.

3.3. Rate of pressure rise

Rate of pressure rise is an important parameter for the investigation in HCCI combustion as it is used to define the upper boundary of HCCI combustion. When the fuelling rates are increased (i.e. lower λ), the HCCI combustion rates increase and intensify, and gradually cause unacceptable noise and potentially cause engine damage, which may eventually lead to unacceptably high levels of NO_x emission. Therefore knocking combustion is often used to define the upper limit of HCCI. Maximum rate of pressure rise is related to combustion noise generated in the engine [48–49]. Fig. 5 shows the maximum rate of pressure rise and its crank angle position for ethanol HCCI combustion at different engine operating conditions. It can be seen from the figure that the maximum rate of pressure rise is very high for richer fuel–air mixtures and is rather low for leaner fuel–air mixtures. For richer mixtures, the rate of combustion is very high for every intake air temperature. Engine becomes very noisy when running on richer mixture. This



a. Maximum rate of pressure rise for HCCI mode at different relative air fuel ratios (λ)

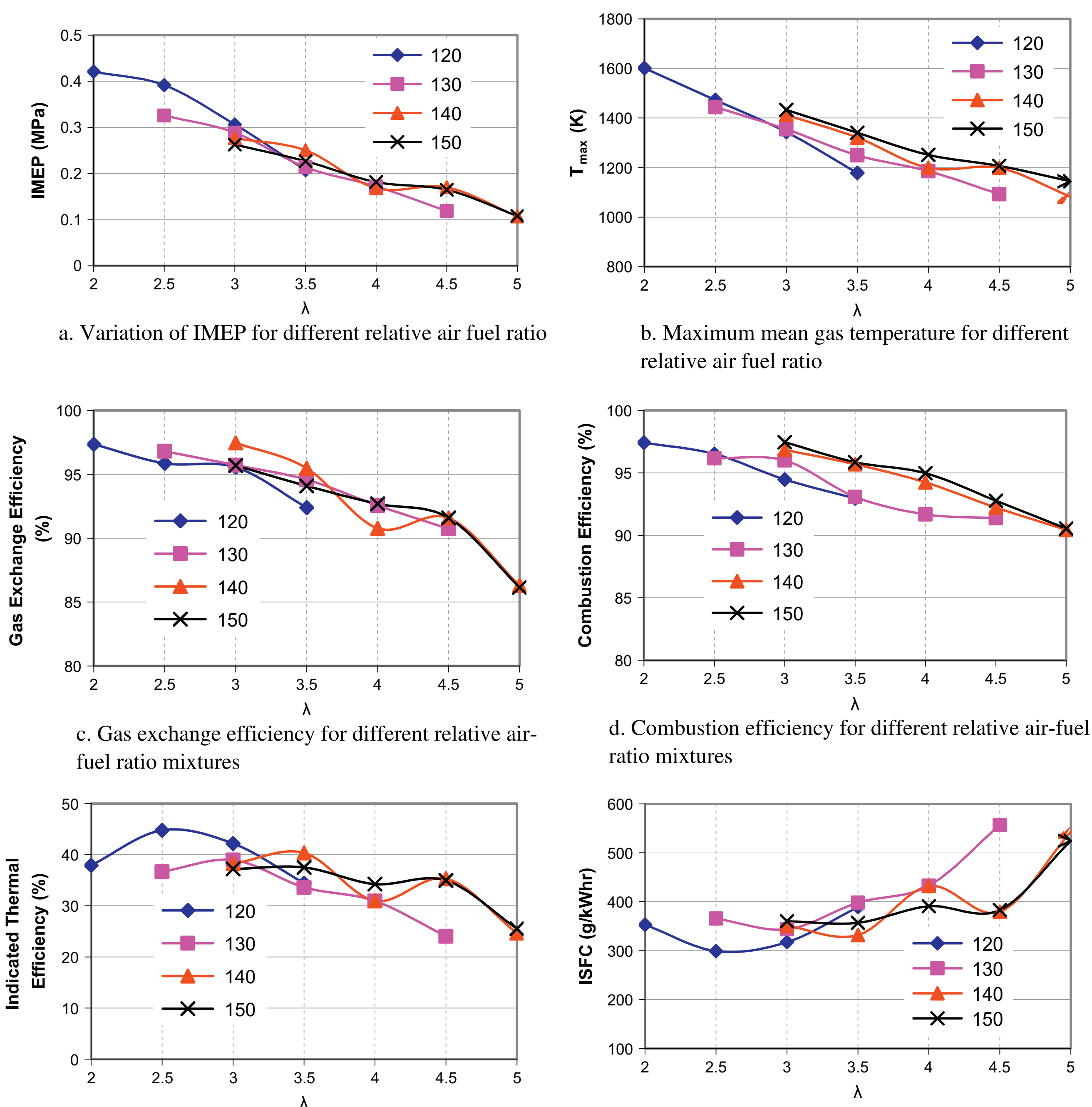
b. CAD Maximum rate of pressure rise for HCCI mode at different relative air fuel ratios (λ)

Fig. 5. Maximum rate of pressure rise and its crank angle position for stable HCCI combustion.

is explained by high rate of pressure rise. It is also observed from the figure that maximum rate of pressure rise decreases with mixtures becoming leaner because smaller amount of fuel is burned. It can be noticed from the figure that on increasing inlet air temperature, crank angle position for maximum rate of pressure rise shifts earlier for constant air–fuel ratio.

3.4. Performance parameters

Fig. 6 shows the performance parameters of stable ethanol HCCI combustion at different engine operating conditions. It is well known that HCCI combustion can only operate at part loads. At higher loads, the peak rate of pressure rise is too high and engine noise increases to unacceptable levels due to multipoint combustion with high burning rates. Due to this limitation, HCCI combustion requires highly diluted mixture in order to slow down the speed of the chemical reactions sufficiently and lead to slower combustion. The rich side limit for IMEP is limited by the rate of combustion and hence that of rate of pressure rise. IMEP is an important indication of the usable power per cycle produced by the engine. The variation of IMEP at different engine operating conditions in HCCI mode is shown in Fig. 6a. The maximum IMEP encountered in this investigation is 4.3 bars with operating boundary conditions of COV_{IMEP} is less than 10% and maximum rate of



a. Variation of IMEP for different relative air fuel ratio

b. Maximum mean gas temperature for different relative air fuel ratio

c. Gas exchange efficiency for different relative air-fuel ratio mixtures

d. Combustion efficiency for different relative air-fuel ratio mixtures

e. Indicated thermal efficiency for different relative air-fuel ratio mixtures

f. ISFC for different relative air-fuel ratio mixtures

Fig. 6. Performance parameters of stable ethanol HCCI combustion.

pressure rise is less than 10 bar per crank angle degree. It can be noticed from the figure that IMEP is decreasing as engine operates on leaner mixtures.

Maximum gas temperature inside the combustion chamber is related to emissions from the engine for HCCI combustion. So, it is worth examining the variation of maximum average gas temperature inside the combustion chamber. With homogeneous combustion of a premixed charge, the temperature is expected to be same throughout the combustion chamber, except near the walls. This, in combination with very lean fuel–air mixtures gives low maximum temperature during the cycle. NO_x formation is very sensitive to peak temperature encountered during the combustion. The variation of maximum mean gas temperature in the combustion chamber is shown in Fig. 6b at all test conditions. It can be noticed from the figure that the maximum mean gas temperature in the combustion chamber is highest for the richest mixture. As mixture becomes lean, the maximum average gas temperature decreases.

The HCCI engine operates un-throttled, which reduces the pumping losses at part load compared to conventional spark ignition engine. Gas exchange efficiency shows the engine losses due to pumping work. Fig. 6c shows gas exchange efficiency of ethanol fuelled engine running in HCCI combustion mode at different engine operating conditions. It is observed from the figure that gas exchange efficiency decreases as mixture becomes leaner at any inlet air temperature. Maximum gas exchange efficiency for ethanol is 97.47%.

Fig. 6d shows the combustion efficiency at different engine operating conditions for ethanol HCCI combustion. Formula for calculation of combustion efficiency is given in Section 2. Combustion efficiency is an indicator of how well the engine is burning the fuel. 100% combustion efficiency is not realistically achievable in HCCI, however very good combustion efficiency should be around 95% [42]. HCCI operation is very sensitive to combustion timing, as it influences the combustion temperature. Late combustion timing means decreased temperature so that the combustion is inferior and the combustion efficiency is lowered. Maximum combustion efficiency is found to be 97.45% among all test points in this investigation. It is observed from these figures that combustion efficiency decreases with the mixture becoming leaner because of lower combustion temperature with leaner mixtures.

Fig. 6e shows the measured gross indicated thermal efficiency. Since the experiments were carried out with a two cylinder engine, which was converted to operate in HCCI mode on only one cylinder, brake thermal efficiencies can be measured with relatively lower confidence. The engine friction of the first cylinder working in HCCI mode becomes high compared to power produced from the same cylinder. Therefore only indicated efficiency calculations are reported in the present research. It is observed from Fig. 6e that the indicated thermal efficiency is lower for leaner mixtures at all intake air temperatures. In the present study, the maximum indicated thermal efficiency (44.78%) is achieved for $\lambda = 2.5$ at intake air temperature of 120 °C for HCCI combustion mode.

Indicated specific fuel consumption (ISFC) is the ratio of fuel consumed and the indicated power. The indicated power is calculated from the pressure–volume curve. Fig. 6f shows the ISFC for ethanol at different engine operating conditions in HCCI combustion mode. It can be observed from the figure that the ISFC decreases with increasingly leaner mixtures. This observation is justified since the indicated thermal efficiency has opposite trend in the HCCI operating region.

Fig. 7 shows the combustion reaction speed of ethanol HCCI combustion at different air–fuel ratio and intake air temperatures. The combustion reaction speed is defined to estimate the rapidity of the oxidation reactions in HCCI combustion. The combustion reaction speed is dependent on the maximum gas temperature

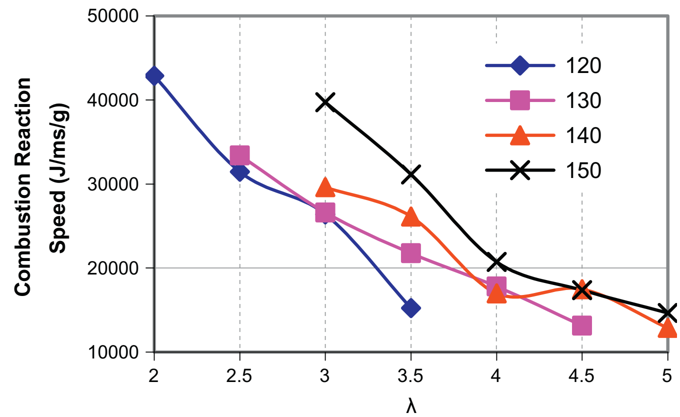


Fig. 7. Variation of combustion reaction speed of ethanol HCCI combustion.

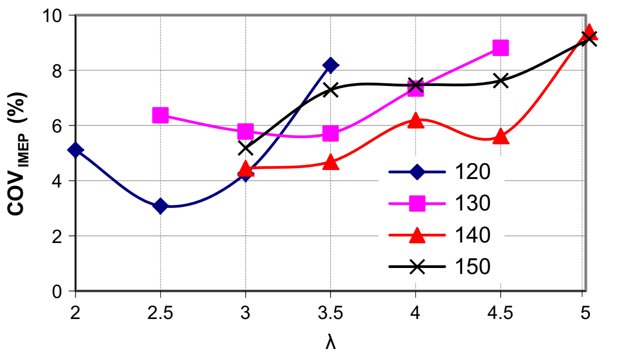
[44]. It cab be observed from the figure that combustion reaction speed decreases as mixture becomes leaner.

3.5. Cycle-to-cycle variation

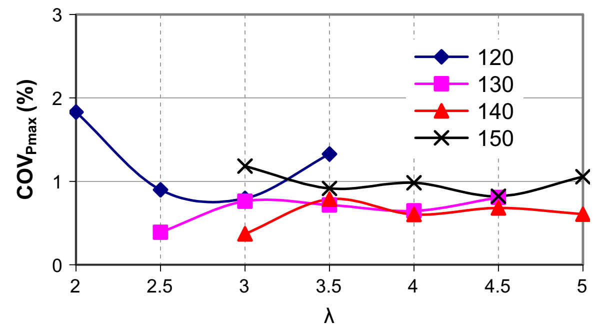
In the HCCI combustion, ignition is completely controlled by chemical kinetics and is therefore affected by fuel composition, equivalence ratio, and the thermodynamic state of air–fuel mixture. There is no external control such as fuel injection or spark timing that are used in CI or SI engines. Cyclic variations occur as the in-cylinder pressure varies from cycle-to-cycle due to the changing rate and incompleteness of the burning generated by asymmetric distribution of temperature and fuel concentration. As a result of the temperature sensitive nature of HCCI combustion, unstable operation has been observed. The measures of combustion stability are coefficient of variation in indicated mean effective pressure (COV_{IMEP}) and $COV_{P\max,}$ which are used in this investigation. These cyclic variations and combustion instability lead to closed loop control of combustion phasing. Fig. 8 shows the cycle-to-cycle variation of different parameters in the combustion chamber.

Fig. 8a shows COV of IMEP for 100 consecutive cycles at each test condition in HCCI combustion mode. It can be noticed from Fig. 8a that COV of IMEP increases with increase in λ (i.e. leaner mixture) for all intake air temperatures. Fig. 8b shows COV of P_{max} for 100 consecutive cycles at each test conditions in HCCI combustion mode. For all engine test conditions P_{max} deviated in narrow range. It can be observed from the figure that for all test points, the variation in maximum in-cylinder pressure is rather small (COV < 2%).

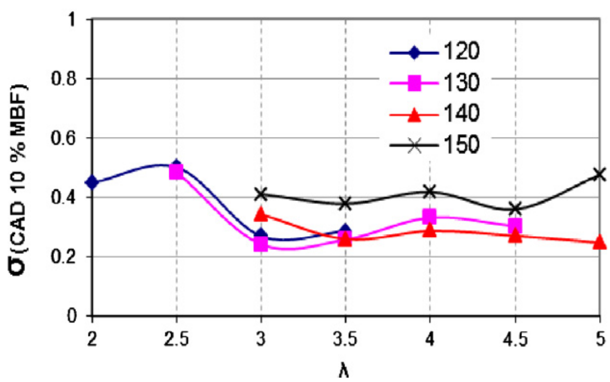
Fig. 8c shows the variation of standard deviation of crank angle corresponding to 10% MBF for all HCCI operating condition. This parameter is often regarded as marker for start of combustion. It can be noticed from figure that the crank angle position of 10% MBF is concentrated in a close range and its spread has standard deviation in the range of 0.24–0.50 CAD. In HCCI, start of combustion depends on chemical kinetics of the fuel. The chemical kinetics is controlled by the pressure–temperature history of the combustion chamber. The COV of P_{max} is very small (Fig. 8b) therefore the cyclic variation in the start of combustion is also very small (Fig. 8c). Fig. 8d shows variation of standard deviation of crank angle corresponding to 90% MBF for all HCCI operating conditions. It can be noticed from this figure that the crank angle position of 90% MBF has large variation in standard deviation and its spread has standard deviation in the range of 0.89–4.52 CAD. The standard deviation is larger for the richer mixtures as compared to leaner mixture as richer mixtures have higher tendency to knock, which



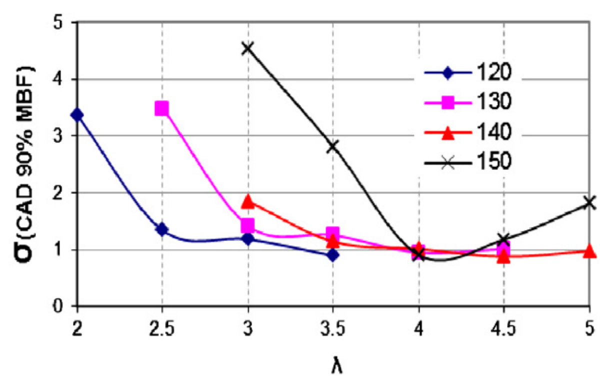
a. Variation of COV $_{IMEP}$ for different relative air fuel ratio



b. Variation of COV$_{Pmax}$ for different relative air fuel ratio



c.Variation of Standard deviation of CAD 10 % MBF



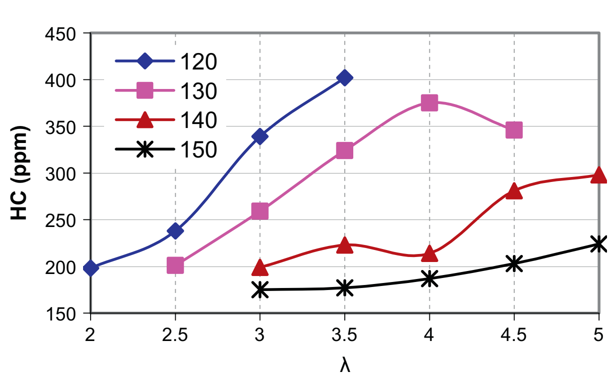
d. Variation of Standard deviation of CAD 90 % MBF

Fig. 8. cycle-to-cycle variation of different parameters for stable HCCI combustion.

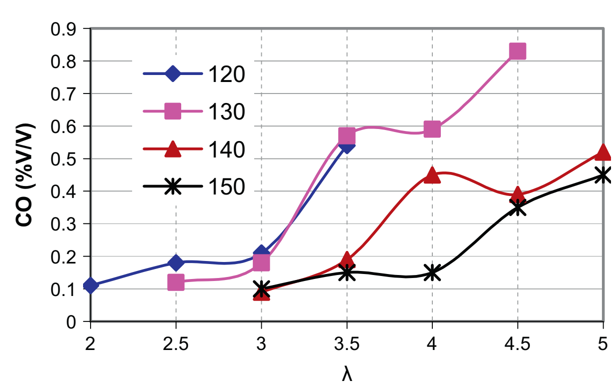
leads to higher cylinder pressure oscillations. Therefore crank angle position corresponding to 90% MBF has larger variation for richer mixtures.

3.6. Exhaust emissions

HCCI features low temperature combustion, but this low combustion temperature results in higher emissions of unburned hydrocarbons. The combustion temperature near the walls will be even lower, due to heat losses. Combustion may be quenched or does not occur at all close to the walls. Unburnt hydrocarbons form as a consequence of incomplete combustion of the hydrocarbon fuels. The level of unburned hydrocarbons in the exhaust gas is specified in terms of the total hydrocarbon concentration expressed in ppm. Engine exhaust gases contain a wide variety of hydrocarbon compounds. Lower combustion chamber temperatures prevent NO_x formation, but at times, this becomes too low to fully oxidize the fuel, resulting in high-unburned hydrocarbons emission. Large parts of the HC emissions probably originate from the wall regions [50]. Fig. 9a shows the HC emissions from ethanol HCCI combustion using intake air temperature in the range of 120–150 °C. The trend shows that HC emissions increase when the



a. HC Emissions at different relative air fuel ratios (λ)



b. CO Emissions at different relative air fuel ratios (λ)

Fig. 9. HC and CO emissions for stable HCCI combustion.

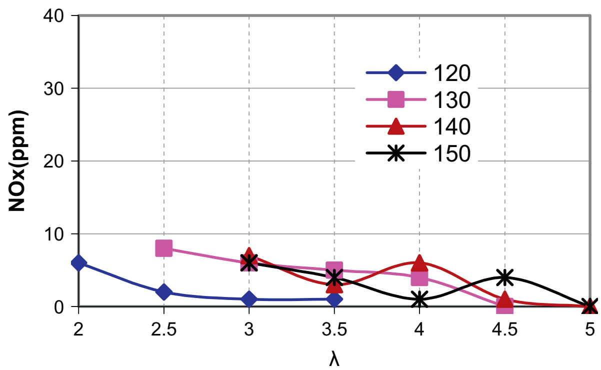


Fig. 10. NO_x emissions for stable HCCI combustion.

mixture becomes leaner. Leaner mixture lowers the temperature of the combustion chamber, thus emitting higher hydrocarbons. It can also be noticed that for constant air–fuel ratio, on increasing intake air temperature, unburnt hydrocarbon emission decreases.

With HCCI combustion, CO is dependent on the combustion temperature. In general, close to the rich limit for HCCI and/or with early combustion phasing, very little CO is generated. But close to the lean limit and/or with late combustion phasing, lot of CO can be formed. Fig. 9b shows the CO emissions from ethanol HCCI combustion using intake air temperature in the range of 120–150 °C. It is observed from figure that on increasing relative air–fuel ratio, CO increases drastically. This depends mainly on the lower combustion temperature and the later combustion phasing. At the end of combustion, the temperature becomes too low for complete oxidation and high amount of CO is generated.

Fig. 10 shows the NO_x emissions from ethanol HCCI combustion using intake air temperature in the range of 120–150 °C.

Nitric oxide (NO) and nitrogen dioxide (NO_2) are usually grouped together as NO_x emissions. Nitric oxide is the predominant oxide of nitrogen produced inside the engine cylinder. The principal source of NO is the oxidation of atmospheric (molecular) nitrogen. NO_x formation is very sensitive to the temperature history during the cycle. At temperatures over 1800 K, the NO_x formation rate increases rapidly with increased temperature. The NO formation rate is governed by Zeldovich mechanism [46]. It is observed from Fig. 10 that NO_x emissions of HCCI combustion are very low for the lean fuel/air mixture and low temperature combustion. For all stable operation points, NO_x emissions are lower than 10 ppm.

4. Conclusions

The combustion and emission characteristic of HCCI engine were investigated in a modified two cylinder engine. The inlet air was supplied in range of 120–150 °C temperatures and the engine was operated at a constant engine speed of 1500 rpm, fuelled with ethanol. Stable HCCI combustion is achieved in range λ (2.0–5.0) for ethanol. HCCI mode operation of the engine is within a narrow load range of the engine. The maximum IMEP obtained during the experiment was 4.3 bars. The coefficient of variation of IMEP is lower for richer mixture of air and fuel and increases as mixture becomes leaner. The coefficient of variation in maximum in-cylinder pressure is small (COV < 2%). Maximum gas exchange efficiency obtained for ethanol HCCI combustion is 97.47%. Maximum combustion efficiency for ethanol is 97.45% for engine operating conditions explored in this investigation. The maximum indicated thermal efficiency was found 44.78% at relative air–fuel ratio of 2.5 for intake air temperature of 120 °C. Extremely low NO_x (<10 ppm) was emitted from all the stable HCCI operating conditions. The HC and CO emission is however higher. HC emissions decrease and CO emissions increase with increase in relative air–fuel ratio (λ).

References

[1] Jeuland N, Montagne X, Gautrot X. Potentiality of ethanol as a fuel for dedicated engine. In: IFP international conference, which fuels for low CO_2 engines? Paris; 2004. p. 07–19.
[2] Owen K et al. Automotive fuels reference book. 2nd ed. Warrendale: Society of Automotive Engineers; 1995.
[3] Christensen M, Einewall P, Johansson B. Homogeneous charge compression ignition (HCCI) using iso-octane, ethanol and natural gas – a comparison with spark ignition operation. SAE paper 972874; 1997.
[4] Joshi S, Lave L, Lester M, Lankey R. A life cycle comparison of alternative transportation fuels. SAE paper 2000-01-1516; 2000.
[5] Dhaliwal B, Yi N, Checkel D. Emissions effects of alternative fuels in light-duty and heavy-duty vehicles. SAE paper 2000-01-0692; 2000.
[6] Nabi M, Minami M, Ogawa H, Noboru M. Ultra-low emission and high performance diesel combustion with highly oxygenated fuel. SAE paper 2000-01-0231; 2000.
[7] Ryan III TW, Mehta D, Calllahan TJ. HCCI: fuel and engine interaction. In: IFP international conference, which fuels for low CO_2 engines; 2004. p. 59–67.
[8] Maurya RK, Agarwal AK. Experimental investigation of the effect of the intake air temperature and mixture quality on the combustion of a methanol- and gasoline-fuelled homogeneous charge compression ignition engine. J Automobile Eng Proc IMechE Part D 2009;223:2041–991.
[9] Onishi S, Hong J, Shoda K, Do Jo P, Kato S. Active thermo-atmosphere combustion (ATAC) – a new combustion process for internal combustion engines. SAE paper 790501; 1979.
[10] Noguchi M, Tanaka Y, Tanaka T, Takeuchi Y. A study on gasoline engine combustion by observation of intermediate reactive products during combustion. SAE paper 790840; 1979.
[11] Iida N. Combustion analysis of methanol-fueled active thermo-atmosphere combustion (ATAC) engine using a spectroscopic observation. SAE paper 940684; 1994.
[12] Najt PM, Foster DE. Compression-ignited homogeneous charge combustion. SAE paper 830264; 1983.
[13] Stockinger M, Schäpertöns H, Kuhlmann P. Versuche an einem gemischansugenden Verbrennungsmotor mit Selbstzündnung MTZ. Motertech Z 1992;53:80–5.
[14] Aoyama T, Hattori Y, Mizuta J, Sato Y. An experimental study on premixed-charge compression ignition gasoline engine. SAE paper 960081; 1996.
[15] Ishibashi Y, Asai M. Improving the exhaust emissions of two-stroke engines by applying the activated radical combustion. SAE paper 960742; 1996.
[16] Machrafi Hatim, Cavadias Simeon, Amouroux Jacques. A parametric study on the emissions from an HCCI alternative combustion engine resulting from the auto-ignition of primary reference fuels. Appl Energy 2008;8:755–64.
[17] Komninos NP. Investigating the importance of mass transfer on the formation of HCCI engine emissions using a multi-zone model. Appl Energy 2009;86:1335–43.
[18] Yokota H, Kudo Y, Nakajima H, Kakegawa T, Susuki T. A new concept for low emission diesel combustion. SAE paper 970891; 1997.
[19] Hou J, Qiao X, Wang Z, Liu W, Huang Z. Characterization of knocking combustion in HCCI DME engine using wavelet packet transform. Appl Energy 2010;87:1239–46.
[20] Maurya RK, Agarwal, AK. Experimental Investigation of cycle-by-cycle variations in CAI/HCCI combustion of gasoline and methanol fuelled engine. SAE paper 2009-01-1345; 2009.
[21] Walter B, Gatellier B. Near zero NO_x emissions and high fuel efficiency diesel engine: the NADITM concept using dual mode combustion. Oil Gas Sci Technol, Rev IFP 2003;58:101–14.
[22] Takeda Y, Keiichi N, Keiichi N. Emission characteristics of premixed lean diesel combustion with extremely early staged fuel injection. SAE paper 961163; 1996.
[23] Iwabuchi Y, Kawai K, Shoji T, Takeda Y. Trial of new concept diesel combustion system-premixed compression-ignited combustion. SAE paper 1999-01-0185; 1999.
[24] Kimura S, Aoki O, Ogawa H, Muranaka S, Enomoto Y. New combustion concept for ultra-clean and high-efficiency small di diesel engines. SAE paper 1999-01-3681; 1999.
[25] Shi S-S. Recent progress in combustion technologies for automotive engines. J Combust Sci Technol 2001;7:1–15.
[26] Tiegang H, Shenghua L, Longbao Z, Chi Z. Combustion and emission characteristics of a homogeneous charge compression ignition engine. J Automobile Eng Proc IMechE Part D 2005;219:1133–9.
[27] Kelly-Zion Peter L, Dec JE. A computational study of the effect of the fuel-type on the ignition time in HCCI engines. Proc Combust Inst 2000;28.
[28] Yang J, Culp T, Kenney T. Development of a gasoline engine system using HCCI technology – the concept and the test results. SAE paper 2002-01-2832; 2002.
[29] Hyvonen J, Haraldsson G, Johansson B. Operating conditions using spark assisted HCCI combustion during combustion mode transfer to SI in a multi-cylinder VCR-HCCI engine. SAE paper 2005-01-0109; 2005.
[30] Wagner R, Edwards K, Daw C, Green J, Bunting B. On the nature of cyclic dispersion in spark assisted HCCI combustion. SAE paper 2006-01-0418; 2006.

[31] Wang Z, Wang JX, Shuai SJ , Ma QJ. Effects of spark ignition and stratified charge on gasoline HCCI combustion with direct injection. SAE paper 2005-01-0137; 2005.
[32] Christensen M, Hultqvist A, Johansson B. Demonstrating the multi fuel capability of a homogeneous charge compression ignition engine with variable compression ratio. SAE 1999-01-3679; 1999.
[33] Mack JH, Aceves SM, Dibble RW. Demonstrating direct use of wet ethanol in a homogeneous charge compression ignition (HCCI) engine. Fuel 2006;85:2622–31.
[34] Poulopoulos SG, Samaras DP, Philippopoulos J. Regulated and unregulated emissions from an internal combustion engine operating on ethanol-containing fuels. Atmos Environ 2001;35:4399–406.
[35] Xing-cai Lu, Yang JG, Zhang W, Huang Z. Effect of cetane number improver on heat release rate and emissions of high speed diesel engine fueled with ethanol–diesel blend fuel. Fuel 2004;80:2013–20.
[36] Jeffrey WH, Irick DK, Whalen MV. Improving the cold start performance of alcohol fueled engines using a rich combustor device. SAE paper 981359; 1998.
[37] Guerrieri DA, Caffrey PJ, Rao V. Investigation into the vehicle exhaust emissions of high percentage ethanol blends. SAE paper 950777; 1995.
[38] Oakley A, Zhao H, Ladommatos N, Ma T. Dilution Effects on the controlled auto ignition (CA) combustion of hydrocarbon and alcohol fuels. SAE paper 2001-01-3606; 2001.
[39] Hui X, Zhipeng W, Bangquan H, Zhao H. Comparison of HCCI combustion respectively fueled with gasoline, ethanol and methanol through the trapped residual gas strategy. SAE paper 2006-01-0635; 2006.
[40] Zhang Y, He BQ, Xie H, Zhao H. The combustion and emission characteristics of ethanol on a port injection HCCI engine. SAE paper 2006-01-0631; 2006.
[41] REVelation operator reference manual. Madison (USA): Hi-Techniques Inc. 2004. p. 95–6.
[42] Muñoz RV. The performance of a dual fuel HCCI engine. Master thesis, ISRN LUTMDN/TMHP-06/5098-SE; 2006.
[43] Iida N. Natural gas HCCI engines. In: Zhao H, editor. CAI and HCCI engines for the automotive industry. Woodhead Publishing Limited; 2007.
[44] Iida N. HCCI engines with other fuels. In: Zhao H, editor. CAI and HCCI engines for the automotive industry. Woodhead Publishing Limited; 2007.
[45] Kalghatgi GT. Fuel effects in CAI gasoline engines. In: Zhao H, editor. CAI and HCCI engines for the automotive industry. Woodhead Publishing Limited; 2007.
[46] Heywood JB. Internal Combustion engine fundamentals. NewYork: McGraw-Hill Book Company; 1988.
[47] Iida M, Hayashi M, Foster DE, Martin JK. Characteristics of homogeneous charge compression ignition (HCCI) engine operation for variations in compression ratio, speed, and intake temperature while using *n*-butane as a fuel.. J Eng Gas Turbines Power 2003;125:472–8.
[48] Eng JA. Characterization of pressure waves in HCCI combustion. SAE paper 2002-01-2859; 2002.
[49] Jeuland N, Montagne X, Duret P. Engine and fuel related issues of gasoline CAI (controlled auto-ignition) combustion. JSAE 20030349, SAE paper 2003-01-1856; 2003.
[50] Christensen, M, Johansson. B. Homogenous charge compression ignition with water injection. SAE paper 1999-01-0182; 1999.

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