AN HCCI ENGINE FUELLED WITH IS-OCTANE, ETHANOL AND THEIR BLENDS

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Abstract

This paper investigates Homogeneous Charge Compression Ignition (HCCI) combustion on an engine that is fuelled with ethanol, iso-octane, and ethanol/iso-octane. The engine is a four-stroke three cylinder in-direct injection type diesel engine converted to a single cylinder HCCI operation. In order to clarify the effects of fuel chemistry on HCCI combustion, the trials were done at a constant engine speed, a fixed initial charge temperature and engine coolant temperature. The HCCI engine was fuelled with a lean mixture of air and fuel (ethanol, iso-octane or mixture of ethanol/iso-octane). The engine performance parameters studied here include indicated mean effective pressure (IMEP) and thermal efficiency. Heat-release rate (HRR) analysis was done to determine the effect of fuels on combustion on-set. The experimental results demonstrate that the addition of iso-octane to ethanol retards the on-set of combustion and subsequently leads to a reduction of the IMEP and thermal efficiency. For a particular fuel, the on-set of combustion depends closely on the intake charge temperature (as reported by several other researchers) and any increase in the initial charge temperature leads to advances in the on-set of combustion. Furthermore, the experimental results demonstrate that operating the engine on a lean charge reduces engine-out NO_x emissions significantly.

Keywords: *HCCI engine, combustion on-set control, alternative fuels*

1. Introduction

The automotive industry has been forced to respond to the increasing environmental concerns of consumers and the rising cost of fossil fuels. Furthermore, the prevailing mode of operation for premixed charge engines, such as gasoline engines, is mostly in partial loading, or with the throttle valve partially closed. This results in poor fuel economy and increased engine emissions. The industry is aimed at developing an alternatively-fuelled engine that can improve engine operation at partial loads. The homogenous charge compression ignition (HCCI) engine combines the use of premixed air and fuel, usually associated with spark ignition (SI) engines, with self-ignition induced by high compression ratio, usually encountered in compression ignition (CI) engines. There are numerous consequences of such an organization of the engine combustion process, such as increased thermal efficiency, lower cycle temperatures and reduced NO_x emissions. To ensure desired ignition timing and to moderate reaction rates during combustion, the air/fuel mixtures need to be lean or even ultra-lean. While there are numerous advantages in exploiting the HCCI mode of operation, there are also a number of issues that need to be addressed. HCCI combustion lacks a means of combustion on-set and subsequent pressure rise control, since both the spark ignition timing (SI engine) and the injection timing (CI engine) are absent. Also, the HCCI engine operation is limited to part-load only [1].

Ethanol is used in this study since it is an alternative fuel that can be produced domestically. It has been an attractive fuel of choice in IC engines due to the potential reduction of CO_2 (a greenhouse gas) that occurs during the production of crops used for manufacturing ethanol. Oakley et al. [2] experimented with a variety of fuels, including ethanol (unblended) to determine the in-cylinder conditions necessary to obtain HCCI combustion in a 4-stroke engine. They found alcohols to be much more tolerant to air and recycled exhaust gas (EGR) dilution. The study done by Guerrieri et al. [3] showed that addition of ethanol decreased the CO and HC emissions, while there was increase in fuel consumption, NO_x and acetaldehyde emissions. However, the CO_2 and formaldehyde emissions did not vary much with the addition of ethanol. HCCI engines and various strategies of HCCI combustion control, which include the use of EGR, preheating of combustion air, and steam injection, are discussed in a comprehensive review compiled by Zhao et al. [4]. A method of enabling auto-ignition timing control was proposed by Furutani et al. [5] in which two different fuels with different octane numbers were combined. The study found that an optimal combination of these two fuels could be determined for each speed and load condition. Christensen et al. [6] showed that HCCI combustion is possible with different types of the fuel and regardless of fuel type used; increasing compression ratio had a strong influence and helped in decreasing the intake temperature**.** Recently, a number of researchers are looking into internal EGR using negative valve overlap to achieve HCCI combustion with conventional compression ratio and less intake air preheating [7,8].

Study done by Mack et al. [9] investigated the HCCI combustion of diethyl ether and ethanol mixtures using carbon 14 tracing both experimentally and numerically. They showed that in HCCI combustion of a mixture of diethyl ether in ethanol, the diethyl ether is more reactive than ethanol. Xingcai et al. [10] experimentally investigated the auto-ignition and combustion characteristics of HCCI combustion for various blends of ethanol/n-heptane mixtures. Their results showed that, with addition of ethanol in n-heptane, the indicated thermal efficiency can be increased up to 50% at large engine loads, but the thermal efficiency deteriorated at light engine load. In terms of operation stability of HCCI combustion, for a constant energy input, n-heptane showed an excellent repeatability and light cycle-to-cycle variation, while the cycle-to-cycle variation of the maximum combustion pressure and its corresponding crank angle, and ignition timing deteriorated with the increase of ethanol addition.

Operation of an HCCI engine is deprived of a well-defined reference event (such as spark timing in SI engine or injection timing in CI engine), which present a special challenge for the engine control. Future engine control strategies have been proposed that rely on cylinder pressure as a fundamental operating principle. The main advantage of such a system is that feedback can be given immediately from the moment of engine firing for use in closed-loop control. There is no warm-up period inherent to this system. However, the challenge remains how to resolve the pressure changes that occur within the combustion chamber. In this study, the engine performance was evaluated based on cycle-resolved in-cylinder measured pressures with the use of a Kistler piezoelectric transducer in terms of rate of heat release, IMEP and thermal efficiency. Furthermore, regulated engine-out emissions are also presented in this paper.

2. Experimental Set-up

Figure 1 shows the experimental setup, which consists of a four stroke, three-cylinder compression-ignition engine (Kubota D905) converted to a single cylinder HCCI operation. The stock engine was an indirect injection type with a pre-chamber.

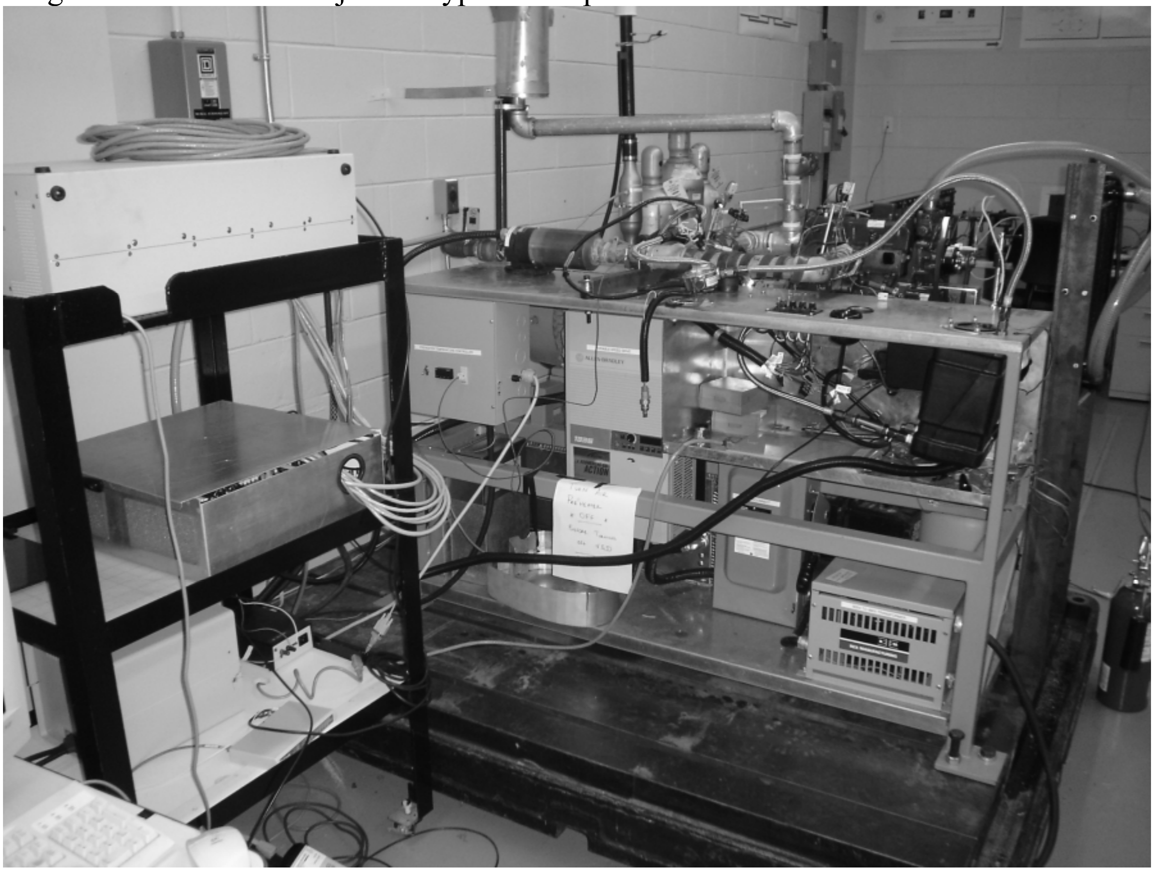


Fig. 1. Experimental set-up and data acquisition system

The engine is motored using an AC motor with a variable speed drive (VSD). The VSD system is used to maintain the engine speed at a desired RPM during both motoring and firing scenarios. The VSD has an additional resistor brake that dissipates the additional power generated by the engine. The experimental set-up also consists of an intake air pre-heater and a temperature controller that maintains the intake charge temperature with an accuracy of ± 1.5 °C from the set value during steady state operation. A laminar flow element in conjunction with a differential pressure transmitter is used to measure the airflow rate in the intake system of the engine. Also, a number of K-Type thermocouples were installed along the intake and exhaust pipes to record the intake and exhaust gas temperatures. The fuel system is designed to be compatible for operation with alcohols and the fuel injector used for the experiments has a dynamic flow rate of 2.18 g/s at 400 kPa fuel rail pressure [11]. The HCCI engine has a modified head with a secondary piston and spacer arrangement that plunges into the engine pre-chamber, shown in Figure 2. By adjusting the secondary piston plunge (spacer width), the compression ratio of the cylinder can be varied between 15:1 and 22:1. It should also be noted from Figure 2 that the passage between the main chamber and pre-chamber has been enlarged to ensure free flow of the gas between the chambers. A Kistler piezoelectric 5mm wall-mounted transducer (6052A1) is mounted in the glow plug hole to acquire in-cylinder pressure measurements. An optical encoder with a 0.1 CA resolution was used for engine speed and crank angle measurements.

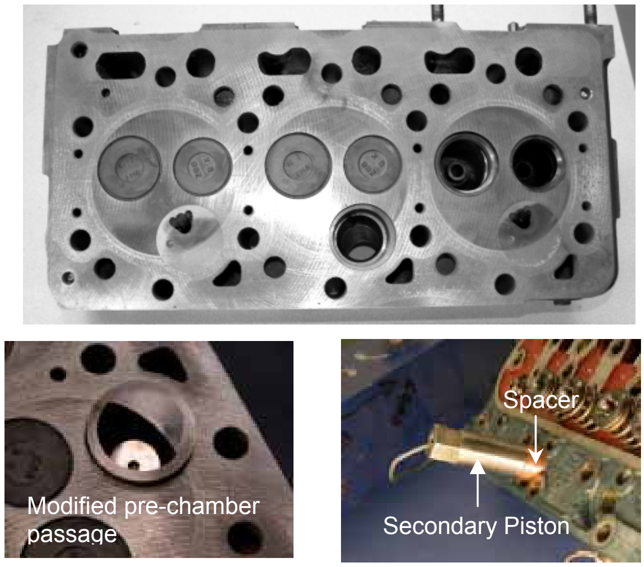


Fig. 2. Engine head with secondary piston and spacer arrangement, [11].
The modified pre-chamber cap before pressure fit is also shown here.

A Combustion Analysis System (CAS) was used for acquiring the pressure and crank angle data. The CAS system has a resolution of 0.1 CA for both pressure and encoder measurements at the maximum rated speed. The data acquisition system has a real-time processor for analyzing engine parameters such as IMEP, maximum pressure rise rate, peak pressure, etc. The engine-out emissions were evaluated using a portable MicroGasTM 5-gas analyzer.

3. Test Conditions

The fuels used in this study are 100% ethanol, 100% iso-octane and 50% ethanol & 50% iso-octane blend by volume. The volumetric efficiency of the engine is between 80% and 72% for the range of engine speed used in this study. The volumetric efficiency decreases with increasing engine speed and intake air temperature. The overall decrease in the volumetric efficiency is due to the lengthening of the stock intake pipes to accommodate the air pre-heater, airflow sensors, and EGR section. The engine specifications and conditions used in the experimental study are as shown in Table1.

The results reported in the next section are averaged data based on sampled 125 individual cycles. The HCCI combustion is initiated by maintaining a high compression ratio (r_c = 20) and adjusting the intake charge temperature from 120 to 150 °C. The discussion of the results includes the engine performance in terms of the HRR, IMEP, and coefficient of variation of IMEP, thermal efficiency, and regulated engine-out emissions.

Tab. 1. Engine specifications and test conditions

Engine Type	Vertical 4-stroke liquid-cooled diesel
Number of cylinders	1 operating (other 2 cylinder valves disabled)
Compression ratio, r_c	20:1
Bore & Stroke (mm)	72.0 x 73.6
Equivalence ratio, ϕ	0.3
Engine Speed (RPM)	805, 1035, 1275 & 1520
Coolant Temperature, T_c	75 °C (± 2 °C)
Intake charge temperature, T_i	120 – 150 °C (± 1.5 °C)

4. Results and Discussion

Figures 3 through 8 depict the in-cylinder pressure and the rate of heat release for the fuels tested over different intake charge temperatures and engine speed (results for 1035 and 1275 RPM are not included). The intake charge temperature for ethanol is varied between 120 °C and 140 °C. For iso-octane and ethanol and iso-octane blend the charge temperature is varied between 130 °C and 150 °C. Data in Figures 3 through 14 indicate that for a particular fuel and engine speed, the intake charge temperature has a strong impact on the on-set of combustion. For example, in Figure 3 the on-set of combustion (the CA corresponding to 1% of HRR) advances from 1.3 CA BTDC to 3.8 CA BTDC when the intake charge is increased from 126.7 °C to 147.2 °C. This influence has been reported by several other researchers earlier, [6, 12-17].

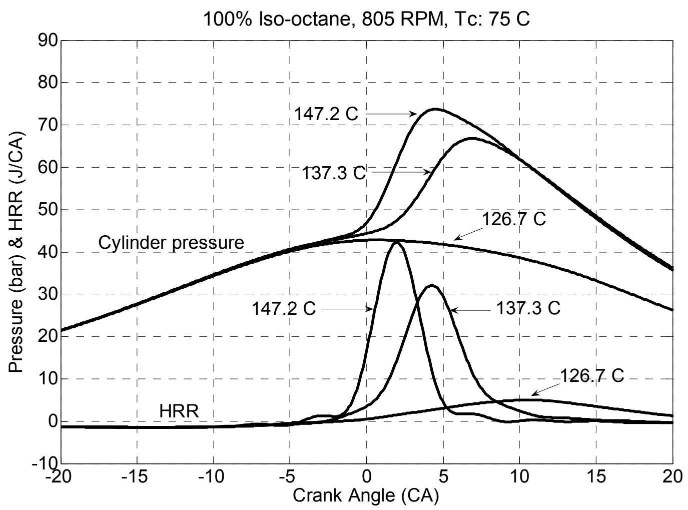


Fig. 3. Cylinder pressure and rate of heat release at different intake charge temperatures (805 RPM, 100% Iso-octane)

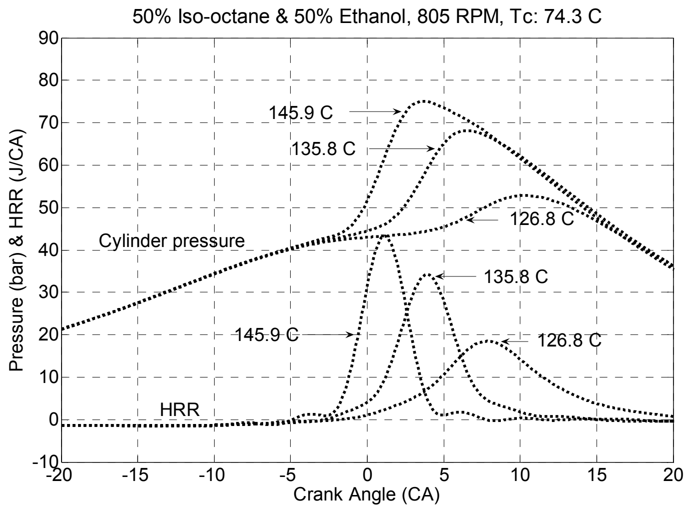


Fig. 4. Cylinder pressure and rate of heat release at different intake charge temperatures (805 RPM, 50% Iso-octane & 50% Ethanol)

Also, for a particular engine speed and intake charge temperature, it can be noted that ethanol advances the on-set of combustion when compared with iso-octane. For example, the on-set of combustion in the case of pure ethanol and pure iso-octane for the 1035 RPM and 136 °C are 5.6 and 2.9 CA BTDC. This trend is confirmed when fuelling with the iso-octane/ethanol blend where the on-set of combustion for the 1035 RPM and 136 °C is 3.8 CA BTDC, which lies between the values of pure ethanol and pure iso-octane. Similar results have been reported in [17, 18]. The advancement in on-set of combustion suggests that alcohols might be advantageous fuels for HCCI combustion. Concurrently, it should also be noted that ethanol has a higher enthalpy of vaporization (0.84 MJ/kg) compared to that of iso-octane (0.351 MJ/kg).

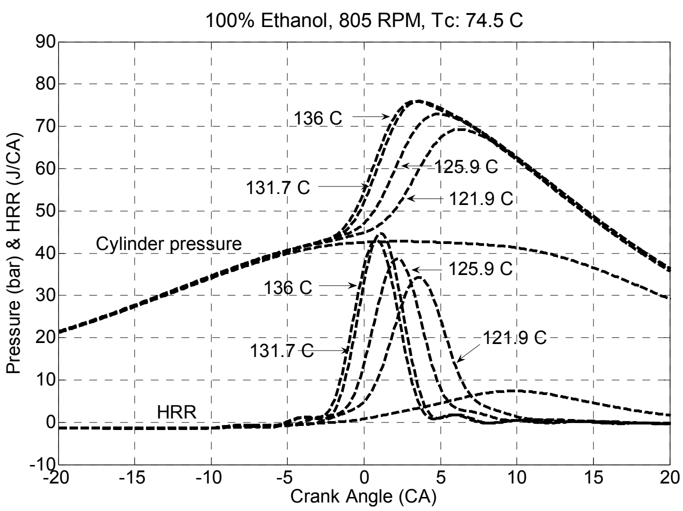


Fig. 5. Cylinder pressure and rate of heat release at different intake charge temperatures (805 RPM, 100% Ethanol)

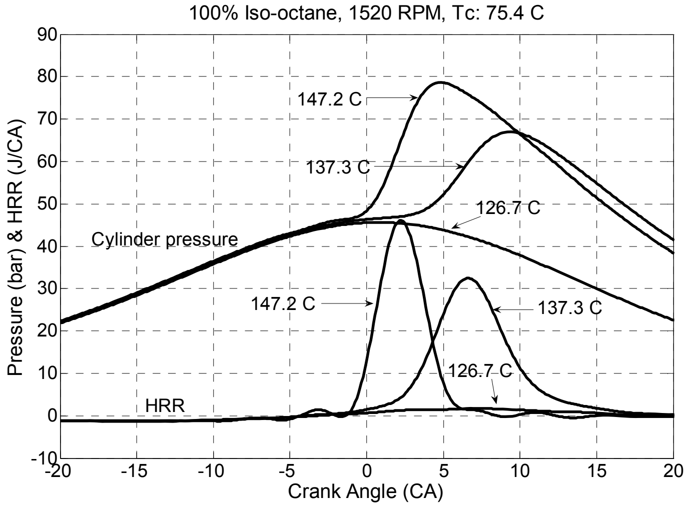


Fig. 6. Cylinder pressure and rate of heat release at different intake charge temperatures (1520 RPM, 100% Iso-octane)

There is an interesting trend that separates the lower speed conditions (805 and 1035) from the higher speed conditions (1275 and 1520) when the intake charge temperatures are low. From the rate of heat release it can be noted that for the lower speeds (805 and 1035) the ethanol and the ethanol/iso-octane blend show consistent ignition without misfire. Whereas, for the same fuels and operating conditions, the higher speeds (1275 and 1520) show either total or partial misfire. This behavior is also observed for pure iso-octane with total misfire at higher speeds. While at lower speeds there is either partial or consistent ignition. This particular effect of either partial or total misfire at higher speeds for the lower intake charge temperature conditions can be linked to two factors: heat losses due to heat transfer and quality of charge mixing.

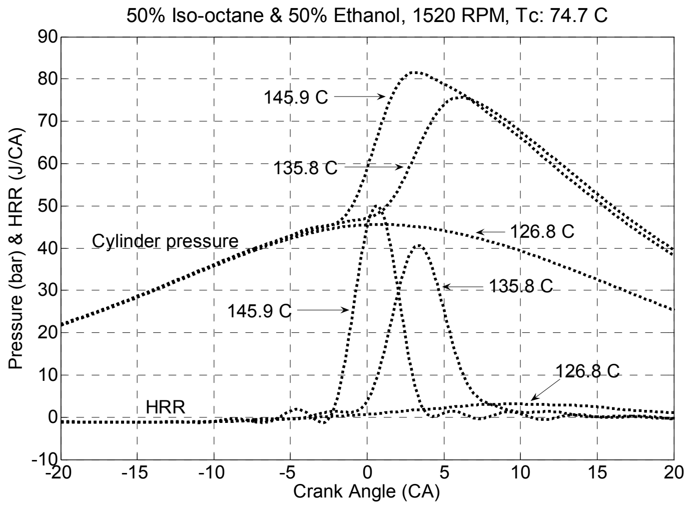


Fig. 7. Cylinder pressure and rate of heat release at different intake charge temperatures (1520 RPM, 50% Iso-octane and 50% Ethanol)

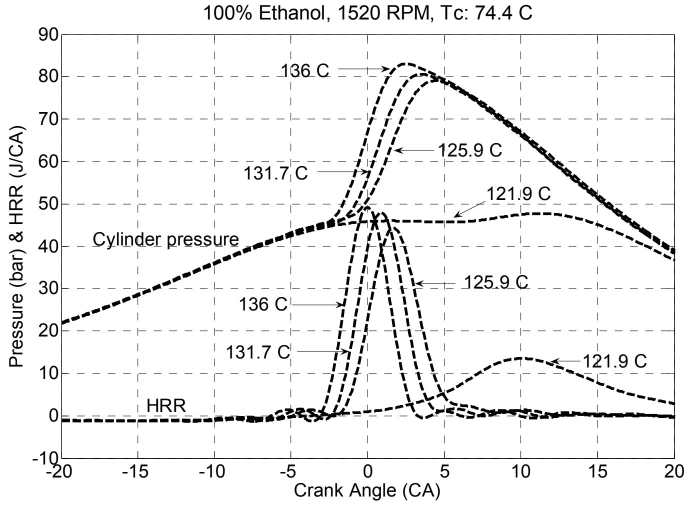


Fig. 8. Cylinder pressure and rate of heat release at different intake charge temperatures (1520 RPM, 100% Ethanol)

The engine used in this study is an In-Direct Injection (IDI) type and the surface to volume ratio for the cylinder is high and this contributes to an increased heat transfer loses. In addition, at higher engine speed the in-cylinder gas velocity increases and hence, the convective heat transfer rate. The overall increase in heat transfer rate can reduce the overall temperature of the charge at the end of compression stroke leading to misfire at higher speeds. However, it should also be noted that the residence time of the charge in the cylinder decreases with increasing engine speed. The shorter residence time will decrease the overall energy transferred as heat during the compression stroke and act in favor of attaining higher charge temperature at the end of compression stroke. Later in this section, looking at the thermal efficiency, it shows that the shorter residence time plays a more important role than the increase in in-cylinder gas velocity and contributes to lower heat transfer losses at higher speeds.

Another important factor that influences the on-set of combustion at lower speeds is inhomogeneities in the charge that are due to poor quality of fuel and air mixing. For the tested engine speeds, the intake pipe flow is a transition flow from laminar to turbulent regimes (Reynolds number, Re: 2500 – 5000). That state of the flow does not promote and ensure homogeneity of the charge. This creates richer regions within the cylinder that ignite at these low intake temperatures. However, again the time available for the charge to mix before compression is longer at lower engine speeds, which will favor the homogeneity of the mixture. From the results obtained, it appears that there is inadequate mixing at lower speeds that leads to richer regions within the engine cylinder and subsequent ignition at the lower intake charge temperature conditions. Furthermore, it should be noted that due to presence of the pre-chamber in this IDI type engine, there is trapped residual mass that acts as an ignition source for subsequent cycles once the initial firing (combustion) cycles occur. The amount of trapped residual mass is not measured here. The pre-chamber occupies 5% of the total cylinder volume at BDC and it is expected that this percentage of trapped residual mass will increase at higher engine speeds.

It should also be said here that at lower speeds the 1035 RPM showed an overall faster pressure rise rate and higher peak pressure compared to that of 805 RPM. Also, at higher speeds the 1520 RPM showed a better trend than that of 1275 RPM. This behavior was consistent during the experimental trials and might be the particular characteristic of this engine.

Figure 9 and Figure 10 show the IMEP and the coefficient of variation in the IMEP for the two intermediate engine speeds. The IMEP reported here are the net values which include pumping losses. From figures it can be noted that the COV_{IMEP} has very high levels at the low initial charge temperature conditions where there is partial or complete misfire.

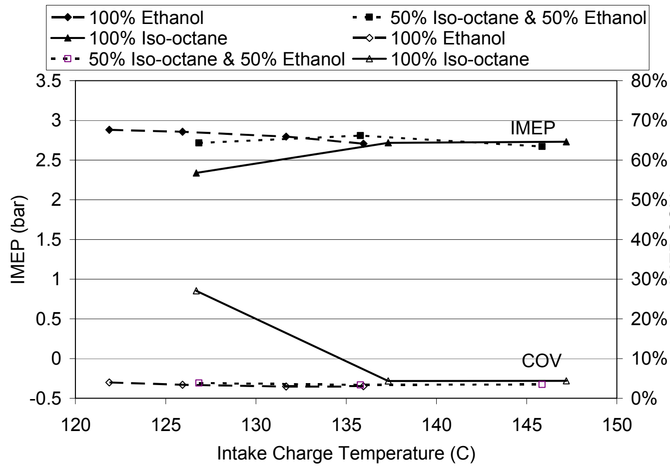
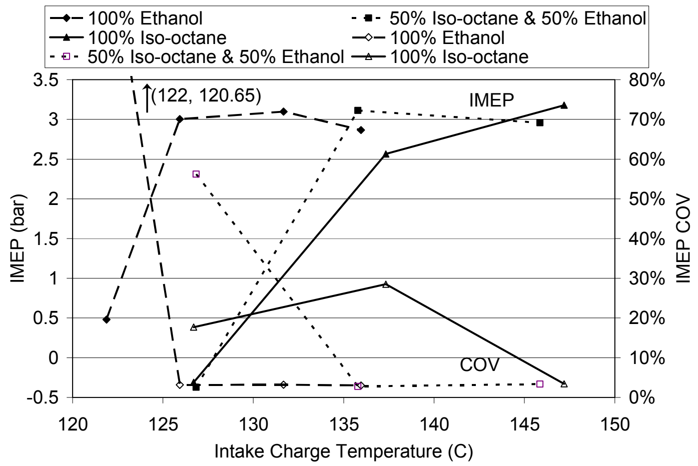


Fig. 9. IMEP and COV $_{IMEP}$ at 1035 RPM



Figu.10. IMEP and COV $_{IMEP}$ at 1275 RPM

Also, looking at pure iso-octane for different operating conditions, there is a specific charge temperature that gives optimal combustion timing. This trend can be also seen clearly for the ethanol and ethanol/iso-octane blend at higher engine speeds (Figure 10). For pure iso-octane, the 147.2 °C gives the best IMEP value for 1035 and 1275 RPM. At 1275 RPM it is evident from the HRR and pressure profile that this operating point features better combustion characteristics than at other temperatures. Whereas, at 1035 RPM 147.2 °C and 137.3 °C intake temperatures very similar IMEP values are attained even though there is significant difference in their pressure and HRR profiles. At 147.2 °C only a little improvement in IMEP is gained. A closer look at the pressure profile will reveal that the137.3 °C pressure curve at first stays below the 147.2 °C pressure curve till 15 crank angle ATDC. At this point, the 137.3 °C pressure curve shifts ahead of the 147.2 °C. It should be noted that the cylinder volume change is minimum when the piston is close to the TDC and significant volume change occurs only during the 30 to 150 CA region of the expansion stroke. Hence, the above noted shift in the pressure curve plays a role in the work done during the expansion stroke. Also, the peak pressures and temperatures for the 137.3 °C case are lower compared to the 147.2 °C case. This in-effect reduces the heat transfer losses for the 137.3 °C condition, especially when the piston is close to the TDC. This combined effect due to the shift in the pressure curve and difference in the heat transfer losses results in similar IMEP values for both the 147.2 °C and 137.3 °C even though they show different pressure and HRR profile. Similar trend is seen for pure iso-octane at 805 and 1520 RPM, where the 137.3 °C shows a better IMEP value compared to that of 147.2 °C condition.

For the ethanol/iso-octane blend the 135.8 °C initial charge temperature gives the best IMEP value for all the tested speeds even though the 145.9 °C shows a pressure profile that is closer to TDC. This is again due to lower heat transfer rates resulting from lower peak temperatures compared to 145.9 °C case and also, due to the shift in the pressure curve between these two temperatures during the 30 to 150 CA region of the expansion stroke where the changes in cylinder volume are maximum. In the case of ethanol, there is not much difference in the IMEP values at lower 805 and 1035 RPM's, with 125.9 °C and 121.9 °C giving a slightly higher IMEP values compared to other initial charge temperatures at 805 and 1035 RPM. Whereas in the case of 1275 RPM the optimum temperature is 131.7 °C and for 1520 RPM the 125.9 °C gives a better IMEP value than the other higher initial charge temperature conditions.

For all the test conditions that produce the highest IMEP values and also for the conditions that are closer to the highest IMEP values irrelevant of the fuel type used, the COV_{IMEP} values remain less than 5%. The only exception from this is the pure iso-octane for the 1520 RPM condition where the COV_{IMEP} is around 6%. The COV_{IMEP} values in this work are within the stable operating region (less than 5%). Similar values where seen for richer mixtures (ϕ - 0.8 to 1) with very high amounts of trapped residual gas at 1500 RPM in [8]. In the same paper, the authors also reported a sharp increase in the COV_{IMEP} with leanest mixture studied (ϕ = 0.8). In our experiments the mixture is very lean compared to that of the one used in [8], and still the COV_{IMEP} stays below 5%. One of the reasons for stable operation can be due to presence of the pre-chamber in this IDI type engine, where there is trapped residual mass that acts as an ignition source for subsequent cycles once the initial cycle with combustion occurs.

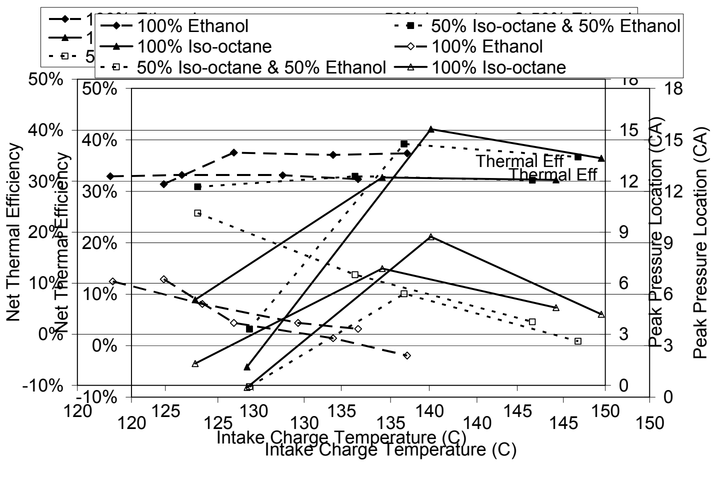


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Fig. 12. Thermal efficiency and peak pressure location ATDC (1520 RPM)

Figure 11 and Figure 12 depict the thermal efficiency at the two extreme engine speeds. The efficiency is calculated as the ratio of net indicated work to product of fuel inducted per cycle and lower heating value of fuel. In addition, the figures show the peak pressure location for different operating conditions. The trends in thermal efficiency are similar to the IMEP since the amount of input energy remains constant for a fixed equivalence ratio. The thermal efficiency of the engine is between 30% and 45% and increases with the engine speed. The net indicated thermal efficiency reported in [19] for similar lean charge of ethanol and iso-octane is about 38% and 40% (compression ratio used in that study was 18 and the equivalence ratio used is ϕ = 0.35 for iso-octane and ϕ = 0.33 for ethanol). The efficiency values reported here are lower than that of [19] because of the IDI type engine used here.

As mentioned earlier, the thermal efficiency increases with engine speed irrelevant of the type of fuel used. It should be noted that the volumetric efficiency for a given intake temperature condition drops by 6% when the engine speed is increased from 805 to 1520 RPM. Hence, the amount of intake charge per cycle does not change significantly with increasing engine speed. From engine-out emission results in Figure 14, the combustion quality does not improve significantly with increased thermal efficiency. These results allow to conclude that the residence time of the in-cylinder charge and the heat transfer losses decreases with increasing engine speed are responsible for improved thermal efficiency of the engine.

Figure 13 shows the maximum pressure rise rate for the various fuels at engine speed of 1035 RPM. The maximum pressure rise rate for all the tested conditions are within 10 bar/CA. The maximum stress limit for IDI engines is about 5-6 bar/CA, [18]. However, in [19] the authors reported that their modified diesel engine (normal stress limit: 6-10 bar/CA) for HCCI operation has been running at 15-20 bar/CA for extended time periods without significant wear. Hence, the pressure rise rates shown here are within the acceptable range. The lowest values for the maximum pressure rise rate were when the engine started to misfire. In general, the maximum pressure rise rate increases with intake charge temperature for a given fuel irrespective of the engine speed. The only exception is the ethanol/iso-octane blend at 805 RPM condition where there is no major difference in the maximum pressure rise rate for different intake charge temperatures. There is also no direct correlation between the maximum pressure rise rate and the best IMEP values. However, there is a direct correlation between the maximum pressure rise rate and the on-set of combustion.

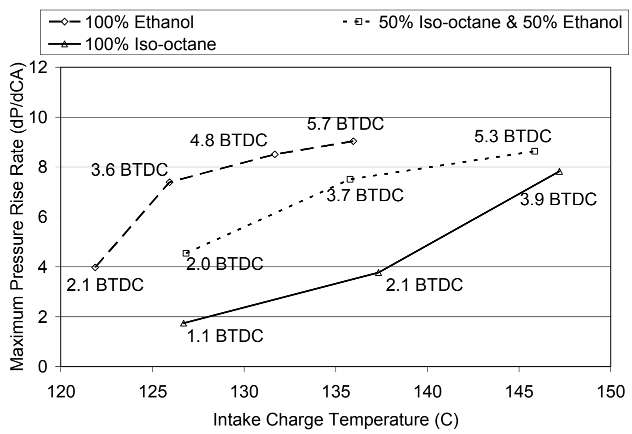

Fig.13. Maximum pressure rise rate with corresponding combustion on-set CA values at 1035 RPM

Figure 13 shows the on-set of combustion values displayed next to the plotted maximum pressure rise rate for each operating condition. From the values it is clear that the combustion on-set has a strong influence on the subsequent pressure rise rate. For most of the operating conditions the best IMEP values where obtained for the maximum pressure rise rate of about 7 to 8 bar/CA. However, in the case of pure ethanol for 1035 RPM, the best IMEP is obtained when the maximum pressure rise rate is only 4 bar/CA. Similarly, for the ethanol/iso-octane blend for the 805 and 1520 RPM conditions, the maximum pressure rise rates are about 5 bar/CA.

Figure 14 shows the carbon monoxide (CO) and non-methane unburned hydrocarbon (UHC) engine-out emissions that corresponds to the best IMEP values for each engine speed and fuel type. Hence, the intake charge temperature may not be the same for a given speed and fuel type. The oxides of nitrogen (NO_x) are not shown in the figure since the maximum value read is less than 10 ppm for the entire test conditions. This shows that near-zero NO_x emissions can be obtained using a lean charge. Similar results for NO_x for the equivalence ratio of 0.321 were reported in [20] for iso-octane.

It can be seen that the UHC emissions are in the order of 750 to 1150 ppm and the CO levels are between 0.15 and 0.26 %.

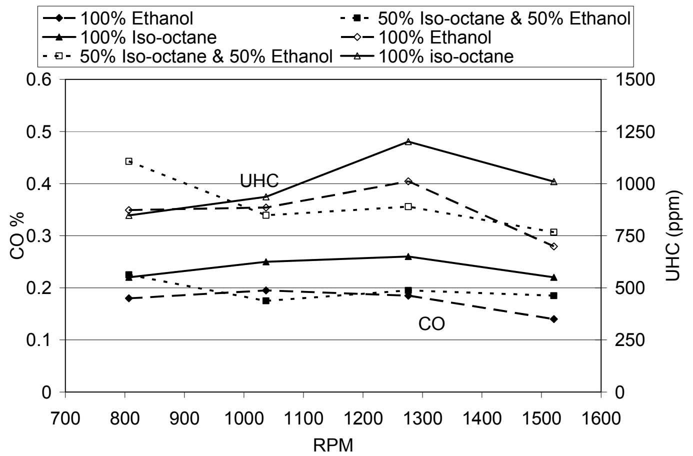

Fig. 14. Carbon monoxide and UHC emissions

In general, the CO and UHC levels are high and require exhaust gas after-treatment. Again, these high levels in CO and UHC emissions might be due to the IDI type of engine being used here, which has greater heat losses that contribute to incomplete combustion especially closer to the cylinder walls. Similar levels of UHC and CO for iso-octane are reported in [20].

5. Conclusion

In this study HCCI combustion of pure ethanol, pure iso-octane, and ethanol/iso-octane blend in lean mixture with air has been studied in an IDI type engine. Based on cycle-resolved in-cylinder pressure measurements, the following conclusions can be drawn:
For the same intake charge temperature and engine speed, the on-set of combustion for ethanol occurs ahead of that for iso-octane. For example, the on-set of combustion in the case of ethanol and iso-octane at 1035 RPM and 136 °C are 5.6 and 2.9 CA BTDC, respectively. It should be noted that ethanol has a higher enthalpy of vaporization than iso-octane and is not taken into account since the charge temperature is maintained constant.

1. The HRR and in-cylinder pressure traces in this IDI engine indicate that the fuel-air mixing has more impact on the on-set of combustion at lower charge temperatures and engine speeds.

2. The thermal efficiency (30 – 43%) values reported in this study are comparable to other HCCI experimental results reported elsewhere and these values are better than those for typical SI engines.
3. The presence of pre-chamber in this IDI type engine ensures stable operation (COV_{IMEP} stays below 5%) due to trapping of residual mass that acts as an ignition source for subsequent cycles once the initial cycle with combustion occurs.
4. The NO_x emissions are very low (less than 10 ppm) for the tested conditions and do not require after- treatment. However, the UHC and CO levels are higher and would require after-treatment.

Acknowledgments

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Appendix A: Nomenclature

Φ	Equivalence ratio,
ATDC	After Top Dead Centre,
BTDC	Before Top Dead Centre,
CA	Crank Angle, in degrees,
CI	Compression Ignition,
COV	Coefficient of Variation,
EGR	Exhaust Gas Recirculation,
HCCI	Homogenous Charge Compression Ignition,
HRR	Heat Release Rate,
IC	Internal Combustion,
IDI	In-direct Injection,
IMEP	Indicated Mean Effective Pressure,
TDC	Top Dead Centre,
T_c	Coolant temperature, in °C
T_i	Initial charge temperature, in °C