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# The effect of ethanol combustion on HCCI engine performance and emissions

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ABSTRACT

Ethanol as a renewable and alternative fuel when utilized in homogeneous charge compression ignition (HCCI) has potential to substantially reduce nitrogen oxides (NOx) and particle matter (PM) and can maintain HCCI sustainability. HCCI engine operation is limited between two ringing and misfire regions that too retard and advance combustion phasing highly affect HCCI engine operation, performance and emissions. In this study, a modified 0.3 liter single-cylinder direct-injection diesel engine has used with a fuel port injection system. A study has done on effect of ethanol combustion on engine performance and emission by varying incoming air temperature and fuel/air mixture which focused on the zones near ringing and misfire regions. The results indicate that too retarding combustion phasing makes HCCI combustion unstable that affects combustion temperature lower than normal combustion which result in reducing indicated mean effective pressure and producing higher hydrocarbon (HC) and carbon monoxide (CO) emissions and too advancing combustion phasing generates high peak pressure rise rate with rapid combustion and reduces IMEP but amount of CO and HC is lower in this situation.

## 1) Introduction

Globalization and the rise in mobility have resulted in the demand for sustainable fuel supply for engines and the reduction in the excess discharge of toxic concentrations of the exhaust gas constituents from the engines in used. To deal with these, there are two major strategies now being adopted at the global stage i.e. i) the use of alternative and sustainable fuels and ii) the shift from traditional internal combustion engine (ICE) technologies towards a better alternatives that enables to offer fuel economy and environmentally-friendly operation. Homogeneously charge compression ignition (HCCI) engine concept is a promising idea that combines well-known characteristics of the spark-ignition (SI) and compression ignition (CI) engine features towards reducing fuel consumption and emissions, providing superior performance [1]. The HCCI engine is similar to SI counterpart for its mixture homogeneity and CI for high compression ignition feature. HCCI engines have higher thermal efficiency than SI and CI engines of similar displacement, while particulate matter (PM) and oxides of nitrogen (NOx) emissions are extremely low in these types of engines [2].

One of the most promising alternative fuels identified as a substitute for gasoline is ethanol. Ethanol, or precisely referred to as bio-ethanol, is typically synthesized from biomass residues through fermentation techniques. Due to the presence of oxygen, it is able to mix and combust well thus producing high combustion efficiency. Due to this factor it is also predicted to reduce exhaust pollutants, i.e. NOx and PM when used in conventional gasoline and diesel engines. Ethanol can be easily synthesized from waste residues, such as palm and sugarcane plantations, with good prospect of continuous future supply [3,4].

The first study on ethanol as fuel for this type of engine was conducted by Christensen et al. (1997) who tested different fuels in a single cylinder engine with constant compression ratio [5]. They used very lean air-fuel mixture in his HCCI operated unthrottled engine gave almost low NOx. Water injection can be used to retard the ignition timing and upper load limit for HCCI engine. Yap et al. (2004) showed using trapping of internal residual gas in naturally aspirated HCCI engine makes lower thermal requirements for the auto ignition process of ethanol [6]. Zhang et al. (2006) used a SI Ricardo engine with valve timing approach. Varying the amount of trapped residuals investigated for HCCI engine fueled ethanol. They showed that valve timing and lambda had great effect ignition timing and combustion duration [7]. Gnanam et al. (2006) investigated the ethanol fuel flexibility of HCCI with adding iso-octane to ethanol for their experiment in a HCCI engine. They studied the effect of mixed fuel on HCCI combustion. In the investigation they found out that by adding iso-octane to ethanol it retards the on-set combustion and consequently IMEP and thermal efficiency decrease and resulted in the increase in the inlet temperature which subsequently advances combustion [8]. Wet ethanol was used in HCCI engine by Martinez-Frias et al. (2007) and they examined the fuel flexibility of ethanol for HCCI and showed that HCCI engine can work on a mixture of 35% ethanol and 65% water by volume [9]. Mack et al. (2009) ran HCCI engine on wet ethanol. In this investigation, by using pure ethanol and water mixture as fuel, they showed that, by increasing in water concentration the cumulative heat release profiles and in-cylinder peak pressures were reduced. They also showed that Hydrocarbon and carbon monoxide emissions had a tendency to increase but NOx levels remained low [10]. The combustion and emission characteristics of a HCCI engine fueled with ethanol were investigated on a modified two-cylinder, four-stroke engine using port injection technique for preparing homogeneous charge by Maurya et al. (2011) [11]. Sjoberg and Dec (2010) showed that no low-temperature heat release (LTHR) occurred in ethanol HCCI combustion [12]. They also found out, ethanol showed a relatively low temperature-rise rate, just prior to its hot ignition point. Sjöberg and Dec (2011) also examined ethanol by comparing it to other fuels. Ethanol's auto-ignition timing has lower sensitivity to the effect of exhaust gas recirculation (EGR) but EGR has retarded the auto-ignition timing [13]. Bahri et al. (2012) modified a single-cylinder, four-stroke, naturally-aspirated, air-cooled, direct injection diesel engine for HCCI operation using ethanol fuel and used experimental data from the HCCI engine to investigate the effect of misfire on the ethanol combustion and HCCI engine operation [14]. Bahri et al. (2013) extended their works to investigate the exhaust gas temperature ($T_{exh}$) and emissions of HCCI engine. They found out that a large number of the HCCI data points have $T_{exh}$ lower than typical catalyst light-off temperatures (250°C) and only 10% of the HCCI data points have $T_{exh}$ higher than or equal to 250°C. The maximum $T_{exh}$ among 100 HCCI data points is about 270°C which is still lower than light-off temperature of some catalytic converters in the market and $T_{exh}$ for an ethanol fueled HCCI engine is strongly dependent on the engine load. $T_{exh}$ as high as catalyst light-off temperature can be achieved by running the engine at higher equivalence ratio (F) which causes fuel penalty in an ethanol fueled HCCI engines [15].

The main objective of this study is to obtain results on the effects of retarding and advancing combustion phasing on both the engine performance and emissions. The outcome of this study can be used to understand ethanol fueled HCCI combustion in greater detail. In this paper, experimental set-up is described and followed by the HCCI experimental conditions are explained and the effects of intake temperature ($T_{in}$) and F on HCCI performance and emissions are discussed in details.

## 2) Experimental Set-up and Instrumentation

A single-cylinder, four-stroke, naturally-aspirated, air–cooled, direct injection diesel engine was modified for HCCI operation using ethanol fuel. The specifications of this engine are listed in Table 1.

Table 1: Yanmar L70AE engine model specifications.

| Parameter (units) | Value |
|---|---|
| Bore (mm) | 78 |
| Stroke (mm) | 62 |
| Compression ratio | 19.5 |
| Displacement (l) | 0.296 |
| Number of valves | 2 |
| Intake valve opening (CAD, aBDC) | 155 |
| Intake valve closing (CAD, aBDC) | 59 |
| Exhaust valve opening (CAD, aBDC) | -59 |
| Exhaust valve closing (CAD, aBDC) | -155 |

Figure 1 shows a schematic of the engine and the instruments used in test bench. The HCCI engine is derived from a four-stroke, air-cooled, direct injection Yanmar L70AE diesel engine which has been converted for single cylinder, port-injection HCCI operation using ethanol fuel. The HCCI engine was coupled to a 30 KW Magtrol eddy-current brake dynamometer for adjusting engine speed and torque.

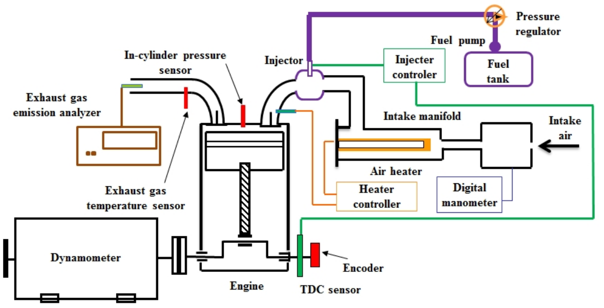


Figure 1: Schematic of the experimental setup used to collect HCCI data

HCCI experiments were done at constant speed when changing other operating parameters. A 3kW electric air heater was installed in the upstream of intake manifold to preheat the air required for facilitating ethanol HCCI operation. Intake manifold also equipped an electronic port fuel injector which controlled electronically for ethanol fuel premixing system. The in-cylinder pressure was mounted inside the cylinder head and measured by a Kistler 601 water-cooled, piezoelectric, high frequency response pressure transducer. A Kistler 2613B optical crank angle encoder with a 0.2°crank angle degree (CAD) resolution was coupled to the crankshaft for engine speed and CAD measurements. DEWE-5000 data acquisition was used for data logging with conjunction with a combustion analyzer DEWECa software for acquiring and analyzing parameters such as indicated mean effective pressure (IMEP), maximum in-cylinder pressure ($P_{max}$) and peak pressure rise rate (PPRR).

The intake and exhaust gas temperature were measured by K-type thermocouples with an accuracy of ±1.5°C in conjunction with a digital temperature indicator. Engine air consumption was measured by an orifice-meter and digital manometer. Exhaust gas emissions of uHC and CO were measured with an five-gas EMS emission analyzer test bench with 4 ppm and 0.06% accuracy, respectively. For each steady-state HCCI operating point, in-cylinder pressure traces were collected and analyzed for a total 120 consecutive cycles.

The tests were conducted in HCCI mode with varying $T_{in}$ and F in three speeds. Experimental data at 100 different HCCI operating conditions are collected for this study. Engine conditions for these 100 operating points are shown in Table 2. The experimental data is used to determine the impact of engine operating conditions on performance and emission in ethanol fueled HCCI engine.

Table 2: Operating conditions of 100 steady-state ethanol HCCI data points

| Variables | Values |
|---|---|
| Engine Speed, N [rpm] | 1350,1550,1750 |
| Intake Manifold Temperature, $T_{in}$ [°C] | 135-166 |
| Fuel-Air Equivalence Ratio, Φ [-] | 0.23 - 0.38 |
| Intake Manifold Pressure, $P_{in}$ [kPa] | 100 |
| Exhaust Gas Recirculation Rate, EGR [%] | 0 |

## 3) Combustion Phasing

The Rassweiler and Withrow method is used to calculate the fraction of fuel mass burnt (MFB) as shown in Figure 2. CA50, and CA90 are defined as the crank angles for 10, 50, and 90 percent MFB respectively. CA10 is selected as SOC and burn duration (BD) is defined as the crank angle period between SOC and CA90 as shown in Figure 2.

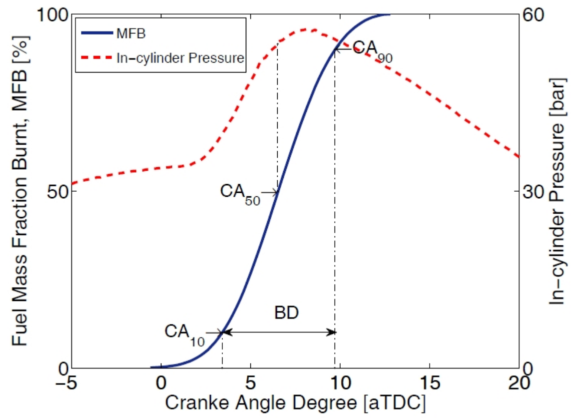


Figure 2: Ignition timing definition of ethanol combustion using in-cylinder pressure trace and fuel mass fraction burnt. (N=1550 RPM, Φ=0.31 and $T_{in}$=153°C)

SOC, CA50 and BD are major ignition timing metrics for HCCI combustion. Finally, the main combustion parameters were calculated from the analysis in-cylinder pressure data.

## 4) Results and Discussion

In the HCCI engine, $T_{in}$ and Φ play an important role for producing the concentration of HC and CO pollutions and the amount of injected fuel practically resulting changing peak pressures and energy release rates which for higher Φ may cause engine damage and cause noise. So, combustion phasing retard should be employed for HCCI fuelled ethanol to avoid extreme pressure rise rates for higher loads but too combustion retard makes HCCI, not to have a stable combustion and in this case erratic combustion with very partial-burn or misfire cycle appear, Since misfire and failure in combustion are not desirable and cause air pollution and engine damage [21]. In these two cases ethanol combustion have great effect on HCCI engine performance and emissions that focused on the zones near knock and misfire regions. In addition, to extend the HCCI operation with range of $T_{in}$, leaner mixtures were investigated in the experiment.

In this work, the effect of ethanol combustion on engine performance, operation, and emissions in HCCI engine fueled ethanol was studied. In this study for extending the HCCI operation in wider range of $T_{in}$, leaner mixture was used in the experiments.

## 5) Ethanol Combustion

The auto ignition timing of HCCI combustion highly depends on the temperature and pressure history during the compression process. The fuel like ethanol has high resistance to auto-ignition due to its high octane number which requires higher $T_{in}$ for ensuring auto-ignition during compression stroke [13]. In this part, ignition timing characteristics of ethanol is considered during several operating condition. Recently, HCCI researchers have shifted towards CA50 as a more appropriate measure of combustion timing in an HCCI engine. Figure 3 shows the CA50 of ethanol in different engine speed, $T_{in}$ and F. It is observed that, in all plots with increasing the $T_{in}$, CA50 advances for all amount of Φ. For higher amount of Φ (Φ=0.38) the operating range is limited by CA50 between, 10-16 CAD aTDC due to penalties in the performance, high noise and ringing and risk of engine damage in highest $T_{in}$ and also high possibility of misfiring and reduction of performance in lower $T_{in}$. For lower Φ, the operating range is wider and CA50 can advance more due to lower peak pressure rise rate. In same $T_{in}$ with increasing amount of the fuel in the mixture, CA50 becomes lower and combustion phasing advances. For higher fuel mixture, CA50 can be retarded more than leaner fuel due to higher combustion temperature. BD is used to determine the combustion duration between the SOC and EOC. Figure 4 shows the relationship between BD and $T_{in}$. The amount of BD is varied over a range of reasonable F and $T_{in}$ which indicates $T_{in}$ strongly affects the ignition timing and BD of the ethanol fueled HCCI combustion.

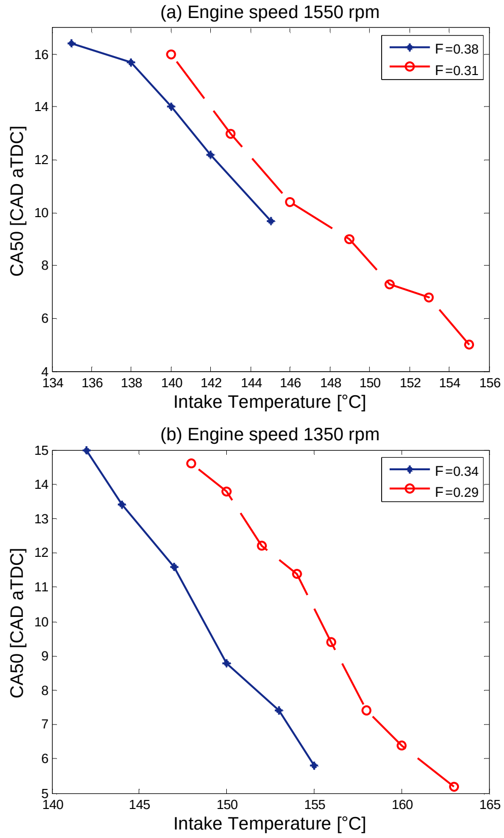


Figure 3: Variation of CA50 versus $T_{in}$.

As can be seen in Figure 4 with the reduction of $T_{in}$, BD becomes long for too retard CA50 near misfire region but with increasing $T_{in}$ (advancing CA50), BD decreases for ethanol combustion. With applying lower $T_{in}$ can elongate the BD to avoid rapid combustion. The auto-ignition of ethanol in HCCI combustion leads to short BD with increasing $T_{in}$ that short BD can cause excessively high pressure rise rates which lead to engine damage and producing engine noise. One possible way of extending the BD in ethanol fueled HCCI engine is decreasing $T_{in}$ and decreasing amount of fuel injected. BD increases with decreasing amount of the fuel in the mixture directly by injected fuel. Also the plots show that ethanol combustion has lower BD for richer mixture which indicates rapid combustion takes place in the leaner mixture and the BD is shortened. For very lean mixture (Φ=0.29), the rate of variation of BD with $T_{in}$ is lower between 12-17 CAD aTDC. The plots show that with increasing F, amount of BD becomes higher in the same temperature.

HCCI suffers from a mechanical limited operating of the HCCI ethanol engine due to misfire at low loads and high rates of pressure rise and in-cylinder peak pressure at high loads. To avoid excessive pressure rise rate and ringing, ethanol fuelled HCCI engines are fuelled with highly diluted mixture. One of the most important parameters for HCCI combustion study is the rate of in-cylinder pressure rise and also plays an important role to detect upper zones for HCCI engine operation.

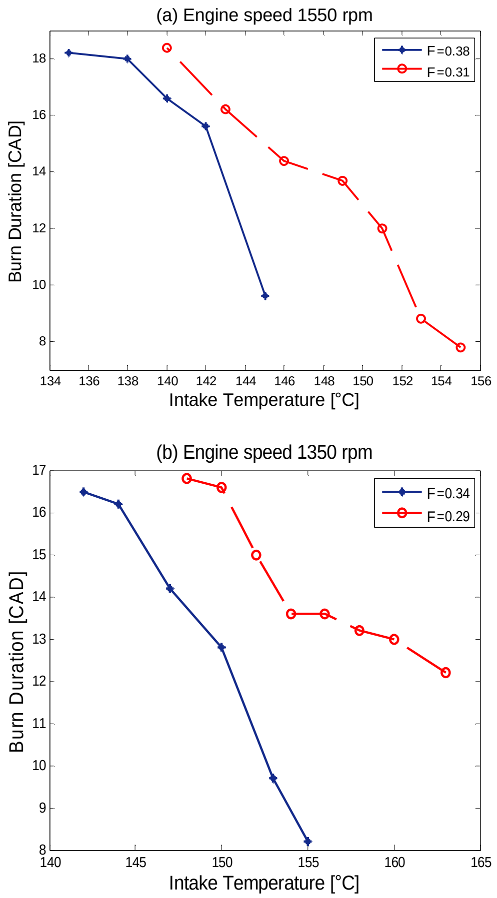


Figure 4: Variation of BD versus $T_{in}$.

Figure 5 shows the rate of variation of PPRR against $T_{in}$. For all cases with increasing the $T_{in}$, the amount of PPRR increases. It can be seen at Figure 5(a) that the PPRR is very high for richer mixtures (8 bar/CAD) because larger amount of fuel is burned and is rather low for leaner mixture (1 bar/CAD). The ethanol combustion rates increases due to increase the fuel rates that causes high PPRR. This situation makes engine work noisy and may damage HCCI engine. Retarding CA50 leads to auto-ignition occur with the downward movement of the piston and away from TDC which decreases PPRR due to the combustion takes place in longer area. As can be seen in Figure 5 (a) for $\Phi$=0.38 by increasing $T_{in}$, there is a sharp increase in PPRR which indicates that running HCCI engine in richer mixture is very risky due to increase temperature and producing engine harsh sound and damage engine with very rapid combustion. Therefore ringing intensity (RI) is often used to define the upper limit of HCCI.

To better illustrate ethanol combustion for the boundary zones near misfire and ringing regions, Figure 6 shows a comparison between an average MFB values were graphed against CAD for two high and low $T_{in}$ for several HCCI operations. By comparing the MFB curves of these two occasions, upon first look, the greatest variation appears to be in the SOC. The SOC ranges from about 0 to 5 CAD. The MFB curve on the right (low $T_{in}$) is very smooth and has only one inflection point. Conversely, the mass fraction burned curve of high $T_{in}$ (Figure 6 (a)) has a very abrupt change its curve and approximately three inflection points. Due to HCCI engine operation conditions, the MFB curves varied dramatically in too advance and retards combustion phasing. The duration of initial combustion is dependent on overall phasing and in-cylinder conditions at the time autoignition. More interesting is the sharp changes in the heat release rates in high temperature. Depending on the operating conditions, this two tiered burning could be beneficial. Although the combustion may appear to be unstable and have some degree of auto-ignition, the combustion is very well behaved.

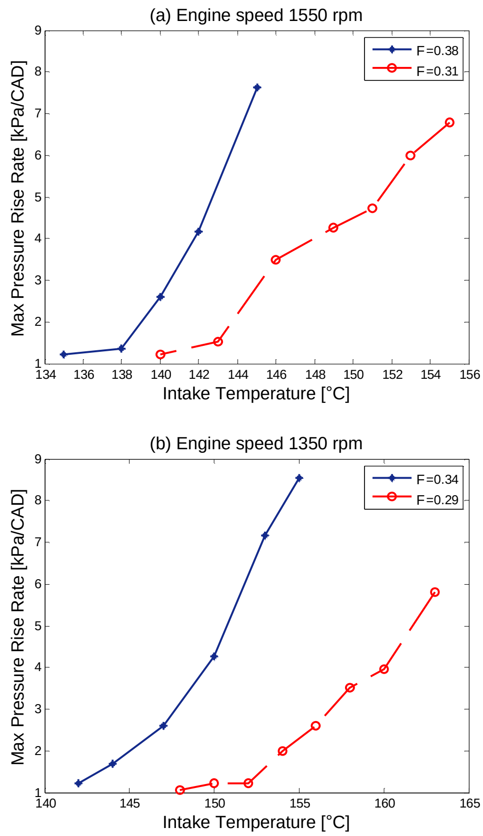


Figure 5: Variation of PPRR versus $T_{in}$

In general, higher $T_{in}$ create more advanced ignition condition, so early ethanol autoignition for HCCI combustion. Advanced autoignition timing causes the bulk of combustion to occur early and fast. For the most advanced autoignition, fresh air and fuel start to burn prior to the end of compression. The most retarded autoignition due to decrease $T_{in}$ is phased such that all of the combustion occurs in the expansion stroke. Figure 7 shows a comparison between PPRR versus CAD for high and low $T_{in}$. The energy release rate of the ethanol combustion is defined in-cylinder pressure rise rate (PRR). For highest $T_{in}$, results in high PRR and thus high combustion induced noise. For very lean mixture

($\Phi$=0.29) at low $T_{in}$ is very low like motoring PRR (below 1 kPa/CAD).

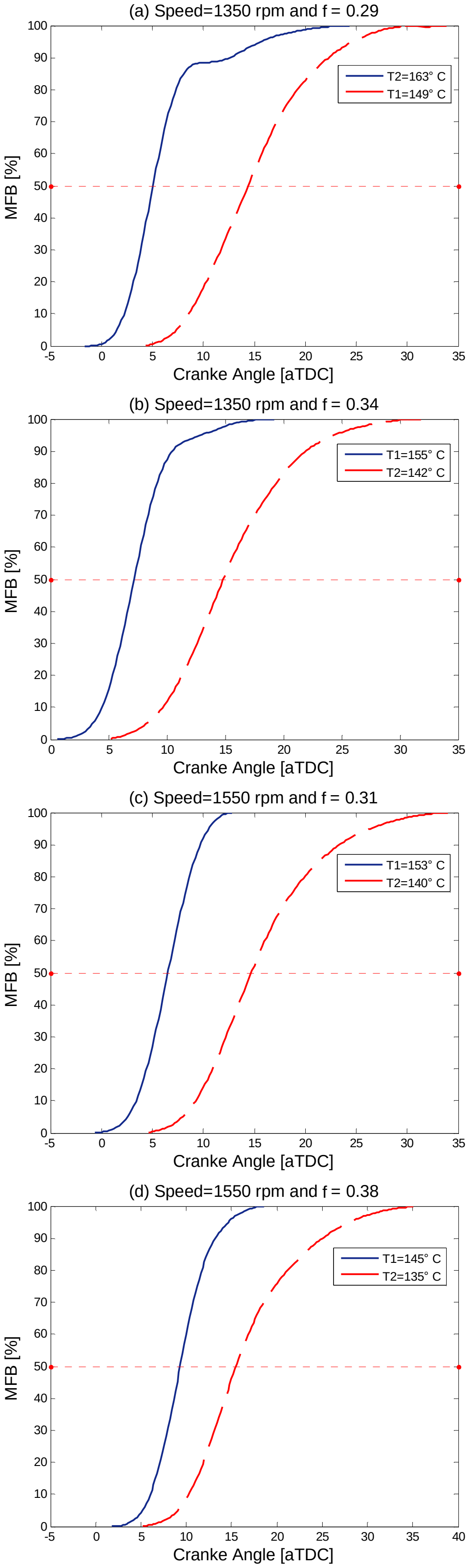


Figure 6: Variation of mass fraction burned against crank angle.

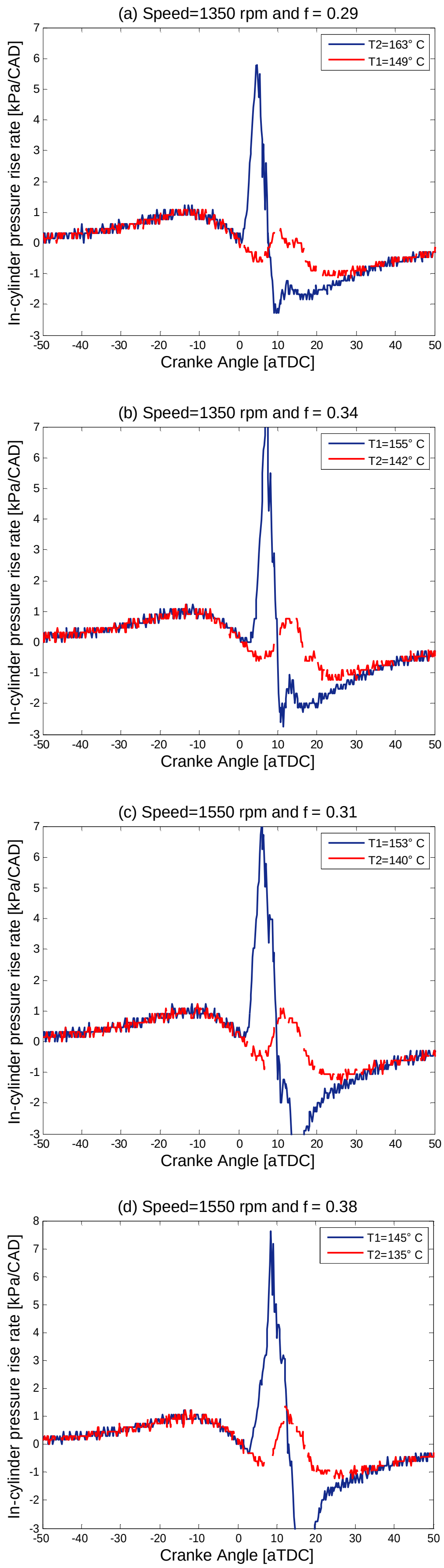


Figure 7: Variation of in-cylinder pressure rise rate against crank angle.

## 6) Engine Performance

The attainable HCCI operating region was mainly limited by the knock limit, misfire, and low IMEP respectively. IMEP is an important indication of the usable power per cycle produced by the engine. Figure 8 shows the variation of IMEP as performance parameters of stable ethanol HCCI combustion at different engine operating conditions and speed. First of all, IMEP is noted to increase with the increase in the $T_{in}$ until reach to intermediate temperature.

After that it starts to drop due to low PRR and unstable combustion which take place near ringing and misfire regions. As can be seen for richer mixture the amount of IMEP for lower $T_{in}$ is higher than higher $T_{in}$ but for very lean mixture with increasing temperature IMEP increases until reaching to the ringing region and due to rapid combustion and increasing PPRR the, IMEP has a sharp drop. Higher IMEP is achieved by increasing fuel in the mixture and selecting moderate $T_{in}$. One of the limitations of HCCI operation is the rate of combustion and hence that of rate of pressure rise is increased and causes the limitation for increasing fueling and IMEP. As can be seen acceptable performance, take place in intermediate $T_{in}$.

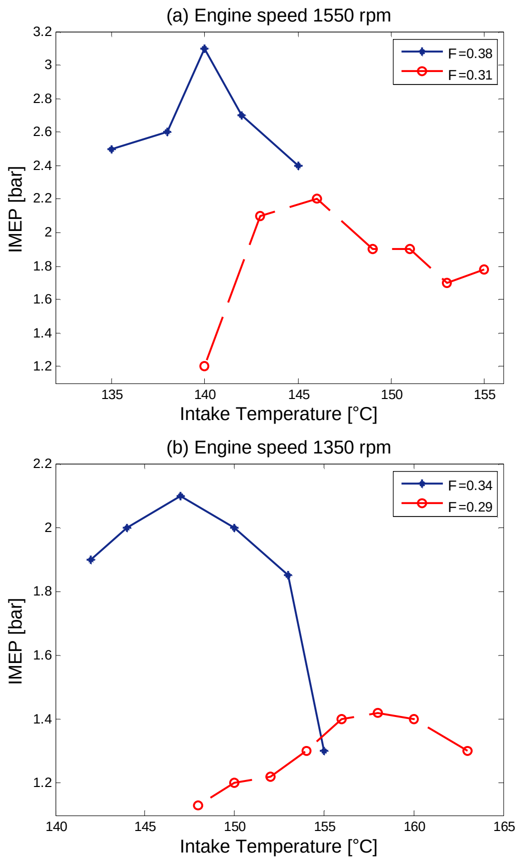


Figure 8: Variation of IMEP versus $T_{in}$.

Figure 9 show the variations of IMEP as a function of $T_{in}$ ,Φ, CA50 and BD for 100 steady-state operating points are studied. These 100 points include normal HCCI operating conditions within misfire and ringing limits as previously shown in Table 2. Figure 9 (a) shows that the operating points with higher $T_{in}$, have lower IMEP which shows HCCI operation near ringing region significantly shorten the amount of IMEP due to rapid combustion and higher amount of PPRR. Figure 9 (b) indicates that with increasing amount of fuel in the mixture (higher F), IMEP increase at moderate $T_{in}$ as stated in Figure 8. As can be seen in Figure 9 (c) and Figure 9 (d), CA50 and BD do not have strong correlation with IMEP.

## 7) Exhaust Emissions

HCCI engines operate with relatively high emissions of HC and CO with comparing with the conventional engines depending on operating conditions. The emissions of CO and HC from the ethanol fueled HCCI engine are highly sensitive to ethanol combustion. With decreasing $T_{in}$ high amounts of HC and CO emissions are produced due to cold combustion. An increase in $T_{in}$ helped to reduce HC and CO. HC exhaust emissions are plotted against $T_{in}$ in Figure 10.

It should be noted that HC emissions increased due to decreasing $T_{in}$ and retarding CA50 for both Figure 10 (a) and Figure 10 (b). Higher HC corresponded to lower F. When CA50 is high due to decrease $T_{in}$, HC increased sharply because it reaches the misfire region. When CA50 advanced, HC increases with increasing $T_{in}$.

Figure 11 illustrates that CO emissions are highly dependent on $T_{in}$. When CA50 advanced with increasing Tin, CO decreased considerably, especially for higher F. HCCI engines operating near the misfire region (lower $T_{in}$) had sharp increases of CO due to lower combustion temperatures, due to lower $T_{in}$ and unstable combustion. CO emissions decrease for large load and light load with increasing $T_{in}$. Richer F have lower amount of CO.

With the emissions concentrations, exhaust after-treatment can be safely dispensed. The smoke emissions also showed a significant reduction, while CO and HC emissions increased substantially. An oxidizing catalyst would be effective at removing the regulated.

## 8) Conclusions

The effect of $T_{in}$ and F on ethanol combustion, HCCI performance and emissions was investigated using a single-cylinder diesel engine which was converted into a HCCI engine mode. Below the list of major findings from the extensive laboratory work undertaken is shown:

1- Ethanol combustion has lower BD for richer mixture which indicates rapid combustion take place in the leaner mixture and the duration of combustion is shorten and for very lean mixture (Φ=0.29), the rate of variation of BD with $T_{in}$ is lower.

2- PPRR is very high for richer F and is rather low for lower amount of F.

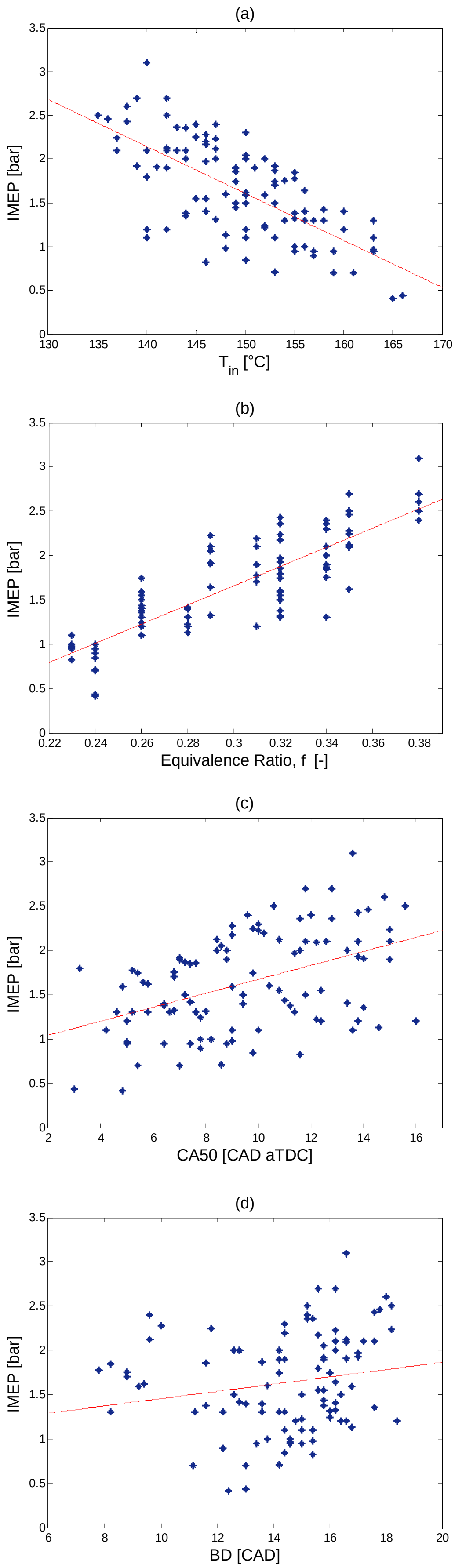


Figure 9: Variations of IMEP as a function of $T_{in}$ ,Φ, CA50 and BD for 100 steady state operating points.

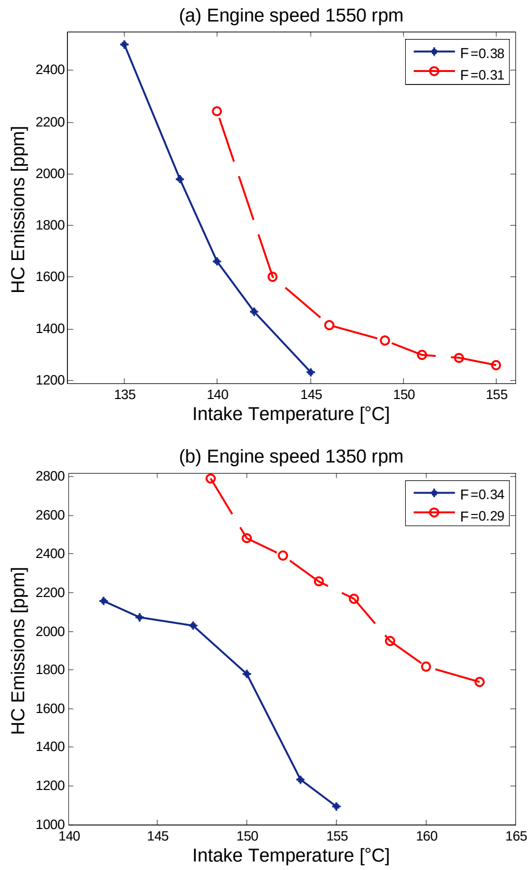


Figure 10: Variation of HC concentration versus $T_{in}$.

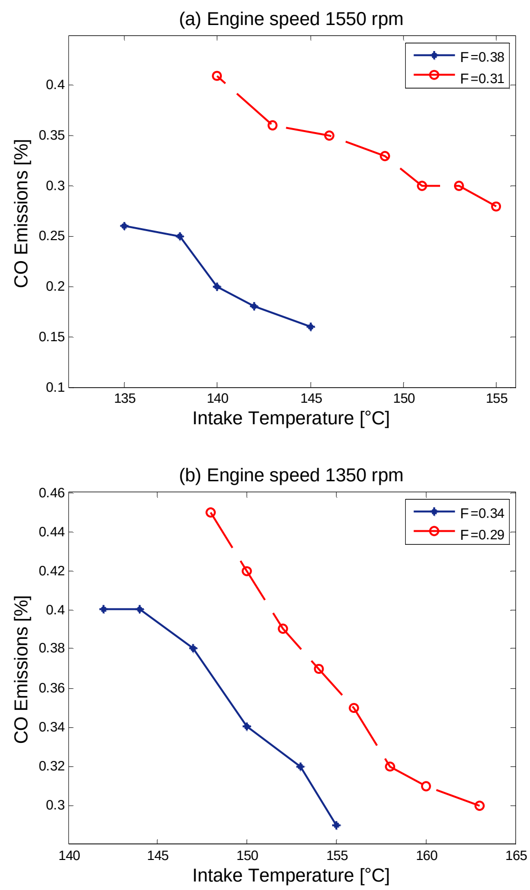


Figure 11: Variation of CO concentration versus $T_{in}$.

3- IMEP is noted to increase with the increase in the Tin until reach to intermediate temperature. After that it starts to drop due to low pressure rise rate and unstable combustion which take place near ringing and misfire region.
4- When CA50 is high due to decrease Tin, HC increased sharply because it reaches the misfire region. When CA50 advanced, HC increases with increasing Tin.
5- The CO emissions are highly dependent on Tin. When CA50 advanced with increasing Tin, CO decreased considerably, especially for higher $\Phi$.

NOMENCLATURE

| | |
|---|---|
| $\Phi$ | Equivalence Ratio |
| aBDC | After Bottom Dead Center |
| aTDC | After Top Dead Center |
| AFR | Air fuel ratio |
| BD | Burn Duration |
| CAx | Crank Angle of x% of Mass Fraction Burnt (e.g. CA50) |
| CAD | Crank Angle Degree |
| CI | Compression Ignition |
| CO | Carbon Monoxide |
| DAQ | Data acquisition |
| EOC | End of combustion |
| HC | Hydrocarbons |
| HCCI | Homogeneous Charge Compression Ignition |
| HRR | Heat Release Rate |
| ICE | Internal Combustion Engine |
| IMEP | Indicated Mean Effective Pressure |
| LTHR | low-temperature heat release |
| MBF | Mass Fraction Burnt |
| $NO_x$ | Oxides of Nitrogen |
| $P_{max}$ | Maximum In-cylinder Pressure |
| PM | Particulate Matter |
| PPM | Parts Per Million |
| PPRR | Peak Pressure Rise rate |
| RPM | Revolution per Minute |
| SI | Spark Ignition |
| SOC | Start of Combustion |
| TDC | Top Dead Center |
| $T_{exh}$ | Exhaust Gas Temperature |
| $T_{in}$ | Intake Temperature |

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# بررسی تاثیر احتراق اتانول بر عملکرد و آلودگی موتور احتراق تراکمی با مخلوط همگن

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## چکیده

استفاده از اتانول به عنوان سوخت جایگزین و تجدیدپذیر در موتور های احتراق تراکمی با مخلوط همگن (HCCI) در توسعه موتورهای احتراق داخلی نقش بسزایی داشته و باعث کاهش اکسیدهای نیتروژن و ذرات معلق گردیده است. موتور HCCI در دو محدوده عملکردی با احتراق ناقص و احتراق با صدای شدید کار می کند، که تاخیر و جلو افتادن احتراق تاثیر فراوانی در کارکرد این موتورها، عملکرد و آلودگی آنها خواهد داشت. در این مقاله، یک موتور 0/3 لیتر دیزل تک سیلندر با پاشش مستقیم به موتور HCCI با پاشش غیرمستقیم تغییر یافته است. در این تحقیق تاثیر احتراق اتانول بر روی عملکرد و آلودگی موتور HCCI بوسیله تغییرات دمای هوای ورودی و نسبت سوخت به هوا در مناطق عملکردی نزدیک به احتراق ناقص و احتراق با صدای شدید صورت گرفته است. نتایج بدست آمده نشان می دهد که تاخیر افتادن زیاد زمان احتراق، احتراق HCCI را ناپایدار کرده که باعث کاهش دمای احتراق از حد طبیعی شده و همچنین باعث خواهد شد که فشار متوسط موثر اندیکاتوری کاهش یافته و مقدار هیدروکربن های نسوخته و اکسیدهای کربن افزایش می یابد. جلو افتادن زیاد زمان احتراق بیشینه فشار داخل سیلندر را افزایش داده و در نتیجه مدت احتراق کاهش یافته و بیشینه اندازه افزایش فشار زیاد در داخل سیلندر تولید خواهد شد. مقدار هیدروکربن های نسوخته و اکسیدهای کربن در این وضعیت نسبت به حالت قبل کاهش پیدا کرده و فشار متوسط موثر اندیکاتوری نسبت به حالت کارکرد نرمال کاهش پیدا خواهد کرد.



## اطلاعات مقاله

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