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ORIGINAL ARTICLE

An experimental study on premixed charge compression ignition-direct ignition engine fueled with ethanol and gasohol

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Combustion duration

Abstract This paper investigates the combustion, performance and emission characteristics of a partial Premixed Charge Compression Ignition-Direct Injection (PCCI-DI) Engine with premixed fuels ethanol and gasohol (90% gasoline and 10% ethanol by volume) along with direct injection of diesel fuel into the combustion chamber. The experiments were conducted in a four stroke, naturally aspirated, air cooled, constant speed diesel engine with 20% premixed fuels from no load to full load condition. The addition of premixed fuel enhances the air fuel mixture strength and for that the combustion duration is decreased in dual fuel operation. From this experiment it was observed the 70% and 67% reduction in smoke emission from premixed gasohol and ethanol fuel when compared to neat diesel operation. In addition to that, the oxides of nitrogen emissions were reduced to 30% and 24% for premixed gasohol and ethanol fuel. In particular, premixed gasohol reduces the smoke and oxides of nitrogen emissions more than the ethanol and also, significant increase in brake thermal efficiency was noted in 20% premixed gasohol and ethanol in dual fuel mode, when compared to neat diesel operation.

1. Introduction

The day to day challenges facing today is the increased emissions which is the environmental challenge. The new combustion concept, namely Homogeneous Charge Compression Ignition (HCCI), has taken the advantage of the working principles of both the Spark Ignition (SI) and Compression Ignition (CI) engines. Here, the mixture preparation is like an SI engine and the combustion is like a CI engine. The HCCI engine operates at nearly constant volume combustion, resulting in high thermal efficiency and improved fuel economy. Lower Oxides of Nitrogen (NOx) could be achieved due to the localized mixture being relatively lean by homogeneous nature [1]. Particulate emission can be reduced significantly due to homogeneous charge combustion. Even though HCCI has the advantage of a high emission reduction potential and improved fuel economy, it has many challenges such as obtaining the homogeneous mixture and controlled auto ignition [2]. Many institutes have already studied HCCI, but

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only a few of them performed experiments using biodiesel as a potential alternative fuel [3].

HCCI is a strategy that has shown the possibility of both lower emissions and lower fuel consumption than SI combustion. However, HCCI combustion can be sensitive to changes in fuel composition. SI engines use a homogeneous air-fuel mixture that is compressed and subsequently ignited by an electric spark. CI engines compress the air charge to a higher level than SI engines and then the fuel is injected into the air, which is hot enough after compression to ignite the fuel. This results in a highly inhomogeneous mixture. It has been shown that an engine can be run using a combination of the SI and CI strategies, by utilizing a homogeneous mixture, but relying on the compression to ignite the mixture. The HCCI concept involves premixing of fuel and air prior to its induction into the cylinder then igniting the fuel–air mixture through the compression process. The combustion occurring in an HCCI engine is fundamentally different from that in an SI or CI engine. HC and CO emissions in HCCI are normally higher than their equivalent of diesel engines. However, reducing HC and CO emissions from HCCI engines is easier than reducing NOx and soot emissions from diesel engines. High HC and CO emissions in HCCI are mainly due to low in-cylinder temperature caused by lean-burn or high-dilution combustion. This can result in incomplete combustion and decrease of post-combustion oxidation rates inside the cylinder. As the charge is made leaner by decreasing fueling or increasing EGR rates, the production of HC and CO is dominated by incomplete bulk-gas reactions.

A lot of researches have been performed in this field [1–10]. However, this concept has major challenges such as combustion initiation and combustion duration control. To overcome these difficulties a lot of work has been carried out in the field of HCCI mode by varying inlet air heating [1], Variable Compression Ratio (VCR) [2] and Variable Valve Actuation (VVA) [9] to alter the effective compression ratio and to trap the residual gases respectively. The Exhaust Gas Recirculation (EGR) rates are also varied to reduce the heat release rate and as a result to control the combustion at higher loads [6]. In addition, the additives were further added to the fuel to boost the physical and chemical properties.

The fuel injection is one of the key parameters to achieve the HCCI mode. The combustion processes that take place inside a diesel engine are essentially dependent on the way in which the fuel is injected into the combustion chamber. The most important criteria are the timing and duration of the injection, the degree of atomization and the distribution of the fuel inside the combustion chamber, the timing of ignition, the mass of fuel injected relative to the crankshaft rotation, and the total amount of fuel injected relative to engine load [10].

An experiment on diesel fuel vaporizer is conducted to prepare the homogeneous diesel vapor air mixture by mounting diesel fuel vaporizer in the intake system. The above investigation reduces the oxides of nitrogen emissions by more than 75% for diesel vapor induction with 10% Exhaust Gas Recirculation compared to the conventional mode of operation [6]. The effects of premixed gasoline fuel and direct injection timing on partial HCCI were analyzed. The results exhibited a significant reduction of oxides of nitrogen and smoke emissions with slight increase in carbon monoxide and unburned hydrocarbon emissions. To overcome this difficulty a premixed PCCI combustion concept is undertaken.

Many different strategies have been formed from this basic idea and they have different names such as Controlled Auto-Ignition (CAI), Low Temperature Combustion (LTC) and Premixed-Charge Compression Ignition (PCCI) [11–20]. The HCCI operates much better and has no flame front, which results in low in-cylinder temperatures, and hence, low NOx formation. The load in an HCCI engine is controlled by the amount of fuel, allowing unthrottled operation. This reduces the pumping losses and decreases the fuel consumption [5].

Even though HCCI combustion can provide emissions and fuel consumption benefits compared to SI combustion, it is still important to investigate the effect of fuel consumption. To overcome these difficulties a PCCI Combustion concept is undertaken for reducing the oxides of nitrogen and soot emissions. The PCCI mode of operation involves the preparation of a premixed charge outside the cylinder. A partial amount of the total fuel supply is injected into the intake manifold where it is mixed with the intake air and the mixture enters the combustion chamber and the rest of the fuel is injected as usual. The premixing of the fuel with the intake air raises the equivalence ratio of the charge entering, and hence the overall non-homogeneity is reduced in the combustion chamber [17].

The port injection of diesel fuel is very difficult for the environment is too cold for the fuel to vaporize. In the diesel engine, the combustion and emission characteristics are greatly influenced by the quality of atomization and, in particular, by the fuel–air mixture present in the combustion chamber. Various methods were tried to achieve proper vaporization of the fuel in the intake manifold. Processes such as hot and cold EGR, preheating the air and large premixing chamber were utilized. Each process has its own set of advantages as well as disadvantages [7]. For example, preheating the air will not only increase the fuel atomization rate, but also decrease the air density, thereby drastically affecting the volumetric efficiency. Hot EGR, if employed, will increase the fuel vaporization, but it would also raise the net chamber temperature, thereby increasing the chance of NOx production, and hence the EGR quantity would necessarily require an automatic control mechanism if it is to be used under different loaded conditions [5].

The performance and emission characteristics of the engine with a PCCI mode and its results were compared with the conventional diesel mode operation. The results of the experimental investigations were analyzed and it was found that the PCCI mode operation results in a better performance than the conventional engine. The reductions in emissions were the primary area of investigation and the area of interest. It is involved in testing the feasibility of a PCCI concept in achieving the simultaneous reduction of NOx and smoke [8].

The present work deals with the study of performance, combustion and emission characteristics of PCCI concept in a direct injection diesel engine with port fuel injection of ethanol and gasohol fuel. The partial premixing is achieved by using two injectors, namely the main injector and an auxiliary injector. In this experimental work a stationary four stroke, single cylinder, constant speed, air cooled diesel engine was adapted to operate in premixed charge compression ignition mode with port fuel injection technique. The experiments were conducted with a 20% premixed gasohol and ethanol fuel in the intake port. The experimental results obtained are compared with the baseline diesel fuel. The main objective of this paper work is to reduce the particulate matter and oxides of nitrogen emissions simultaneously.

2. Experimental setup and test procedure

The research engine was based on a single cylinder, four stroke air cooled, constant speed and direct injection diesel engine developing 4.4 kW at a constant speed of 1500 rpm. The test engine specifications are shown in Table 1 and the experimental setup is shown in Fig. 1.

The fuel tank is connected to graduated burette to measure the quantity of fuel consumed in unit time. An Orifice meter with U-tube manometer is provided along with an air tank on the suction line for measuring air consumption. The loading is done by means of a dynamometer. The load is controlled by changing the field current. The cylinder pressure is measured with piezoelectric pressure transducer. A charge amplifier is then used to produce an output voltage proportional to this charge. An AVL 365C Angle Encoder Indi Advanced is mounted rigidly on the camshaft of the engine. The test rig is installed with AVL software for obtaining various results during operation. An AVL415 smoke meter is used to measure the smoke opacity expressed in terms of percentage. A five gas analyzer is used to obtain the exhaust gas composition. All emissions such as Carbon monoxide, Carbon dioxide, Un-Burnt Hydrocarbons, Nitrogen oxide and unused oxygen are found in the gas emission analyzer. In this cable one end is connected to the inlet of the analyzer and the other end is connected at the end of the exhaust gas outlet.

Uncertainty analysis is considered necessary to establish the efficiency of the measurement. The percentage for the various parameters is given in the Table 2.

To show how the premixed fuel affects engine performance, the authors employ the premixed ratio (r_p), which is defined as the ratio of the energy of premixed fuel to the total energy.

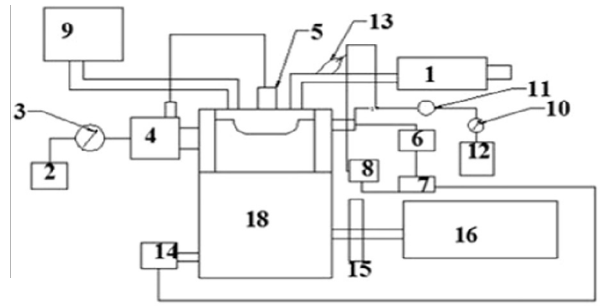
The premixed ratio is r_p is given by the Eq. (1)

$$r_p = \frac{m_p h_p}{m_p h_p + m_d h_d} \tag{1}$$

where m_p and m_d show the mass consumption of premixed fuel and directly injected diesel fuel respectively, and h_p and h_d are the calorific values of premixed fuel and diesel. The test experiment was conducted on a premixed ratio of 0.2, from which the mass flow rate of premixed fuel is calculated. The injection duration is calculated from the bench test in milliseconds and set in electronic control unit. The premixed fuel is completely

Table 1 Technical specifications of the engine.

Engine type	Kirloskar Oil EngineTAF1
Bore	87.5 mm
Stroke	110 mm
Swept volume	661.5 cc
Injection timing	23° bTDC
Nozzle opening pressure	220 bar
Rated output	4.4 kW
Rated speed	1500 rpm
Compression ratio	17.5:1
Cooling system	Air



1.Air surge tank
2.Diesel fuel tank
3.Flow meter
4.Fuel injection pump
5.Fuel injector
6.Charge amplifier
7.Data acquisition
8.Relay and temperature controller
9.Exhaust gas analyzer
10.Flow meter
11.Fuel control valve
12.Diesel fuel tank
13.Fuel injector
14.Crank angle encoder
15.Flywheel
16.Dynamometer
17.pressure sensor
18.Diesel engine

Figure 1 Experimental setup.

Table 2 Uncertainty analysis.

Error analysis parameters	Uncertainty (%)
Brake power	4.6
Total fuel consumption	0.75
Brake thermal efficiency	6.09
NOx	2
HC	2.8
CO	0.08
Smoke	2.2
Exhaust gas temperature	1.2

Table 3 Comparison of fuel properties.

Properties	Units	Diesel	Gasohol	Ethanol
Calorific value	kJ/kg	42,500	44,000	26,000
Density at 15 °C	Kg/m^3	860	750	785
Flash point	°C	74	−43	13
Cetane number	Nil	49	3	8.32
Specific heat	kJ/kg K	1.8	2.4	2.2
Latent heat of vaporization	kJ/kg	250	305	940

controlled by the electronic control unit. The properties of the test fuels used are listed in Table 3. The use of ethanol and gasohol blends in conventional diesel engine is restricted to low mixtures thus allowing an improvement in fuel efficiency and a reduction of tailpipe emissions.

3. Result and discussion

In this work, the combustion characteristics such as Pressure, heat release rate, ignition delay, combustion duration, mass burn rate with respect to crank angle of port injected premixed gasohol and ethanol fuel are compared with the baseline direct injection diesel combustion. Similarly the performance characteristics such as brake thermal efficiency and brake specific energy consumption and emission characteristics such as brake specific carbon monoxide, brake specific unburned hydrocarbon, oxides of nitrogen and smoke emissions were compared with baseline diesel fuel. The result of this exhaustive research has brought out the significance of dual fuel to the emerging needs of HCCI combustion.

3.1. Combustion analysis

3.1.1. Pressure vs. crank angle

The variation of cylinder pressure with the crank angle diagram is shown in Fig. 2. It is observed that the premixed gasohol and ethanol fuel exhibits a higher cylinder pressure compared to that of baseline diesel fuel. Distinctly premixed gasohol shows higher peak pressure than premixed ethanol. The lower calorific value of ethanol is the reason for the reduced peak pressure than that compared to premixed gasohol. The maximum peak pressure of 72.60, 68.85 and 68.2 bar occurs at 6°, 7° and 8° aTDC for premixed gasohol, ethanol and neat diesel fuel. The cylinder peak pressure and maximum rate of pressure rise for premixed gasohol were found as higher due to improvement in premixed combustion due to increased flame rapidity that leads to a complete combustion [2]. Another reason might be higher energy release rate, so the peak cylinder pressure shifted 1° and 2° CA bTDC and it is slightly away from the diesel peak pressure regime for premixed ethanol and gasohol fuel. The increased ignition delay would increase the amount of fuel burnt during the premixed combustion phase. Hence there is an increase in peak pressure that would have only been to shift in the heat release pattern on premixed combustion region.

3.1.2. Heat release rate

The variation of heat release rate with the crank angle diagram is shown in the Fig. 3. The heat release rate was calculated from the first law of thermodynamics [12].

$$\frac{dQ}{d\theta} = \frac{k}{k-1} P \frac{dV}{d\theta} + \frac{1}{K-1} V \frac{dP}{d\theta} \quad (2)$$

where k is the polytropic coefficient, P is the in-cylinder pressure and V is the instantaneous cylinder volume. The start of combustion is calculated from the differential (mass fraction

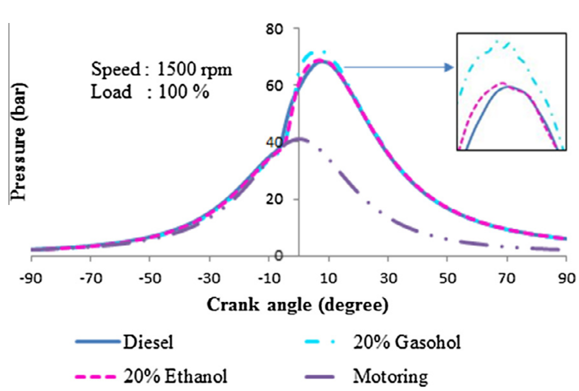


Figure 2 Variation of pressure with crank angle.

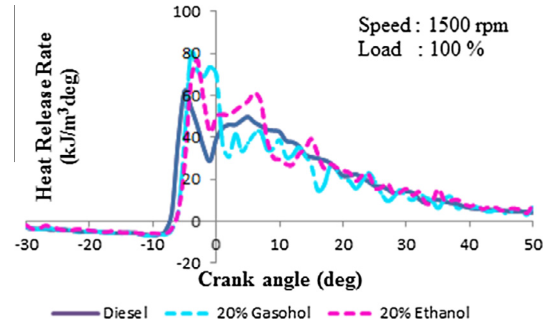


Figure 3 Variation of heat release rate with crank angle.

Table 4 Ignition delay in terms of crank angle and milliseconds.

Load (%)	Ignition delay (°CA)			Ignition delay (ms)		
	D	G	E	D	G	E
0	17.83	19.73	18.51	1.98	2.19	2.05
25	17.10	19.01	18.36	1.9	2.11	2.04
50	16.43	18.15	18.08	1.82	2.01	2.00
75	16.15	17.27	17.34	1.79	1.91	1.92
100	15.58	16.61	16.46	1.73	1.84	1.82

D – Diesel, G – 20% Gasohol + 80% Diesel and E – 20% Ethanol + 80% Diesel.

burned per °CA) heat release curve. It is observed from the graph that premixed gasohol and ethanol shows the higher heat release rate than conventional diesel operation. Due to the premixed gasohol and ethanol fuel injection, the heat release curve dips into the negative range before its steep rise. The partial premixed mixture inside the cylinder during the compression stroke would have reacted with the intake air and emitted heat at low temperature. The subsequent zero transition is taken as the start of combustion. The maximum heat release rate is 81.10, 77.08 and 62.02 kJ/m^3 degree occurs at 3°, 4° and 5° bTDC crank angle for premixed ethanol, gasohol and formal diesel fuel mode. From the result, the heat release rate results for the premixed gasohol and ethanol fuel improves the premixed combustion phase and also the rate of heat release pattern was delayed by premixed ethanol and gasohol fuel mixing and becomes closer to TDC. In particular premixed gasohol shows 5% increased heat release rate than premixed ethanol fuel. The reason might me due to the increased calorific value of the gasohol fuel. The diffusion and after burning regime for premixed gasohol shifted toward the TDC caused the complete combustion.

3.1.3. Ignition delay

The ignition delay is nothing but the time lag between the start of fuel injection and the start of fuel combustion. The corresponding values of ignition delay for all loads measured in terms of crank angle degree and converted in terms of milliseconds are shown in Table 4. It is observed from the table that ignition delay decreases gradually from no load to full load conditions. In particular premixed gasohol and ethanol fuel exhibits higher ignition delay than the neat diesel fuel operation. It has been reported by a number of researches that dual fuel operation with various liquid and gaseous induced fuels generally increases the ignition delay [5,10]. The reason for the decrease in ignition delay with an increase in the engine load is due to increase of in-cylinder temperature. So many numbers of factors are involved in increasing the ignition delay of a dual fuel engine. The pilot fuel, diesel spray was surrounded by air and premixed gasohol and ethanol fuel vapor mixtures and the reaction with this alcohol air mixture can affect the ignition of the pilot fuel [10]. The premixed gasohol and ethanol has high latent heat of vaporization which reduces the compression pressure and temperature. In addition, very lower Cetane value of the premixed ethanol and gasohol fuel may have also involved in increasing the ignition delay, since the auto ignition temperature is higher for the lower Cetane valued fuel.

3.1.4. Combustion duration

The combustion duration is nothing but the time lag between the start of combustion and the end of combustion. The corresponding values of combustion duration for all loads measured in terms of crank angle degree and converted in terms of milliseconds are shown in Table 5. It is observed from the table that combustion duration gradually increases from no load to full load conditions. The combustion duration is calculated by means of mass burned rate analysis. The mass burn rate of the fuel with respect to crank angle is shown in Fig. 4. The information on the crank position with respect to the start and end of combustion processes can be gained by mass fraction burned analysis.

$$x_b = \left\{1 - \exp\left[-a\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^{m+1}\right]\right\} \tag{3}$$

$$m = \frac{\ln\left(\frac{\ln 0.1}{\ln 0.9}\right)}{\ln\left(\frac{\theta_{90}-\theta_0}{\theta_{10}-\theta_0}\right)} \tag{4}$$

where a is a constant which is equal to 2.3026 for mass fraction between 0% and 90%, θ – crank angle, θ_0 – start of combustion, $\Delta\theta$ – combustion duration in terms of degree and m – form factor.

From the plots, the rate at which fuel–air mixture burns increases from a low value to a maximum and then decreases as the combustion process ends. In particular premixed gasohol and ethanol fuel exhibits higher combustion duration than

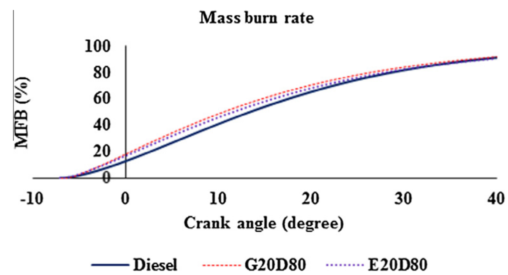


Figure 4 Variation of mass burn rate with respect to the crank angle.

the neat diesel fuel operation. The reason might be the longer ignition delay of the premixed gasohol and ethanol fuel.

3.2. Performance analysis

3.2.1. Brake thermal efficiency

The variation of brake thermal efficiency with brake mean effective pressure is shown in Fig. 5. It is evident from the graph that brake thermal efficiency increases in dual fuel mode. The increase in brake thermal efficiency from no load to full load condition indicates the capability of the combustion system to convert the fuel energy into mechanical work. Generally, the highest thermal efficiency occurs at the full load, where only the combustion efficiency is maximized. The brake thermal efficiency at full load is 31.12% and 31.86–20% premixed gasohol and ethanol fueled diesel, which is 4.65% and 3.69% higher as compared to diesel. The BTE increases due to an enhanced combustion rate of premixed fuel. The combined effect of lower density and lower fire point of the premixed gasohol and ethanol increases the combustion rate.

3.2.2. Brake Specific Energy Consumption (BSEC)

The mass of fuel consumption is shown in Fig. 6. If two different fuels of varying density are blended, then the brake specific energy consumption is measured as a replacement for Brake Specific Fuel Consumption (BSFC).

$$\text{BSEC} = \text{BSFC} \times \text{CV} \tag{5}$$

where CV is the calorific value of the fuel, the brake specific energy consumption is represented in terms of MJ/kW h. It is clear that BSEC is lower for premixed gasohol and ethanol fuel compared to the normal diesel mode. Generally BSEC decreases with an increase in the brake power. This may be

Table 5 Combustion duration in terms of crank angle and milliseconds.

Load (%)	Combustion duration (°CA)			Combustion duration (ms)		
	D	G	E	D	G	E
0	23.12	27.70	27.72	3.14	3.44	3.57
25	26.10	28.59	28.38	3.55	3.62	3.66
50	28.59	30.74	30.49	3.90	3.95	3.93
75	31.93	33.64	32.89	4.30	4.37	4.28
100	37.55	37.10	38.45	4.99	4.83	4.99

D – Diesel, G – 20% Gasohol + 80% Diesel and E – 20% Ethanol + 80% Diesel.

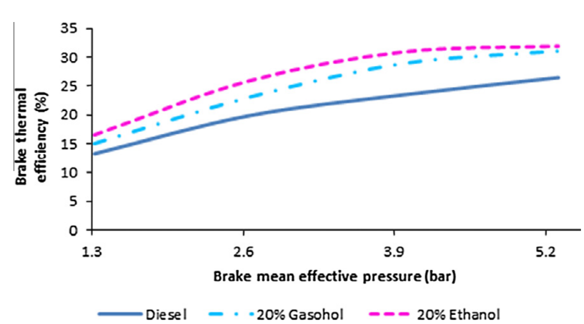


Figure 5 Variation of brake thermal efficiency with respect to brake mean effective pressure.

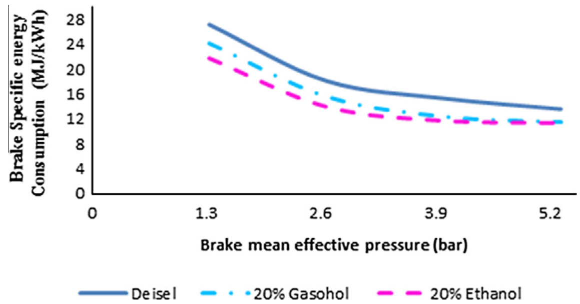


Figure 6 Variation of brake specific energy consumption with respect to brake mean effective pressure.

due to the decrease of the calorific value by adding the premixed gasohol and ethanol fuel. The brake specific energy consumption for premixed gasohol and ethanol fuel at 20% load is 24.09 MJ/kW h and 21.69 MJ/kW h respectively, and at full load is 11.56 MJ/kW h and 11.29 MJ/kW h respectively. For diesel it is 27.19 MJ/kW h at 20% load and 13.59 MJ/kW h at full load. The above trend agrees with the result of Hansah et al. [6].

3.3. Emission analysis

3.3.1. Brake specific carbon monoxide

The brake specific carbon monoxide emissions with brake mean effective pressure for premixed gasohol and ethanol fuel operation are compared to base diesel. The carbon monoxide and unburned hydrocarbon emissions are measured in terms of parts per million and they are converted into grams per kilowatt hour as shown in Fig. 7. From the graph it is observed that carbon monoxide emission for premixed ethanol and gasohol operation is higher than a formal mode of diesel engine operation. Specifically the carbon monoxide emission is increased to 70% at part load condition and it is reduced significantly equal to the diesel fuel operation at the full load. This might be due to the reduced in-cylinder temperature than the full loaded condition for the premixed gasohol and ethanol fuel. It may be caused by the cooling effect of vaporizing premixed fuel. Due to lean burn and decrease in the diffusion and post-combustion oxidation, the in-cylinder temperature is reduced which leads to increased carbon monoxide emission. The other reason for the decrease in carbon monoxide might be the increase in ignition delay which gives the adequate time

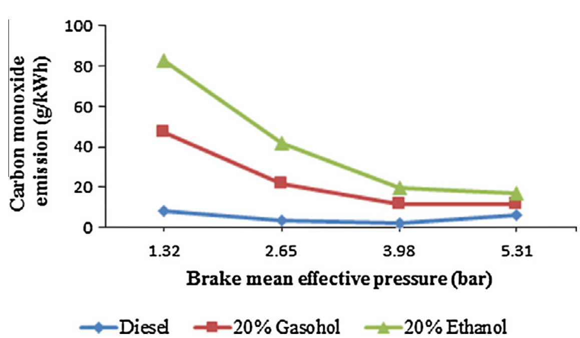


Figure 7 Variation of carbon monoxide with brake mean effective pressure.

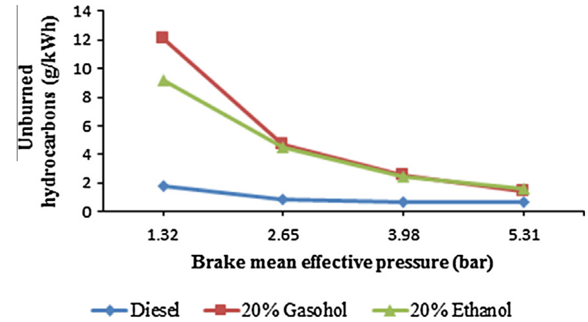


Figure 8 Variation of unburned hydrocarbons with brake mean effective pressure.

for the burning of the fuel. The ignition delay increases from 1.73 ms for conventional mode of diesel operation to 1.84 ms and 1.82 ms for 20% premixed ethanol and gasohol fuel at low load condition. This might be due to the effect of oxygen atoms in ethanol at the full load condition. The increased ignition delay leads to increased combustion of certain quantity of fuel and the carbon monoxide oxidation rate. The decrease in carbon monoxide at the full load results in much significant reductions in unburned hydrocarbon emissions.

3.3.2. Brake specific unburned hydrocarbon

The brake specific unburned hydrocarbon emissions with brake mean effective pressure of premixed gasohol and ethanol fuel compared with diesel are shown in Fig. 8. The formation of unburned hydrocarbon emission is due to a foresaid fundamental reason as explained in the carbon monoxide emissions. From the graph (Fig. 8) it is observed that unburned hydrocarbon emissions for premixed ethanol and gasohol are higher than a formal mode of diesel engine operation. The reason might be due to the proximity of lower combustion temperature of premixed gasohol and ethanol at the part loads. The unburned hydrocarbon emissions decrease with increase in brake mean effective pressure. This may be the result of higher gas temperatures in the full load which would favor more complete oxidation. In particular premixed ethanol shows 5% lesser emission than premixed gasohol at low load. The escape of some quantity of premixed fuel from the inflammable region through crevice volume is one of the reasons for the increase in unburned hydrocarbon emission than conventional mode. Due to high latent heat of vaporization, the premixed fuel, which lowers the temperature inside the combustion chamber and forms a thick quenching layer, affects the burning of diesel fuel around the spray. The oxygen self-contained ethanol fuel assists in burning fuel more completely, so the unburned hydrocarbon concentration is lower with premixed ethanol and gasohol than with diesel. At full load, the decrease in carbon monoxide and unburned hydrocarbons shows the improvement in overall combustion efficiency.

3.3.3. Brake specific nitrogen oxide

The variation of brake specific nitrogen oxide given in terms of g/kW h with respect to diesel is shown in Fig. 9. It is observed from the graph that oxides of nitrogen emissions are reduced to 30% and 24% on average for premixed gasohol and ethanol fuel than the neat diesel fuel operation. The reason might be the formation of partially lean air fuel mixture inside the combustion chamber compared to diesel combustion due to lower combustion temperatures, since oxides of nitrogen formation are very sensitive to the temperature of the cycle. In particular premixed ethanol shows 9.5% lesser NOx emission at the full load than the gasohol due to lower combustion temperature caused by the lower heating value of the ethanol. Another reason may be due to high latent heat of vaporization which increases the specific heat of the premixed fuel mixture. The increase in specific heat reduced the peak combustion temperature. But at the fully loaded condition oxides of nitrogen emission increases slightly for the premixed gasohol and ethanol fuel. The cause may be due to the increased fuel octane number of the premixed fuel, since a higher temperature is required to get auto ignition. Another reason might be due to the strength of air fuel mixture which is rich that increases the combustion rate and higher rates of pressure rise. It is due to the reason that at the richest zones, sudden auto ignition of the mixture takes place and combustion temperature will also locally be higher. At the instance of higher combustion temperature few diatomic nitrogen breakdown of monatomic nitrogen forms more oxides of nitrogen.

3.3.4. Smoke opacity

The variation of smoke emission with respect to brake mean effective pressure is shown in Fig. 10. The smoke emission decreased significantly for premixed gasohol and ethanol fuel than with conventional mode of diesel operation owing to more oxygen and less carbon in the premixed fuel. The reason might be that premixed combustion will be more as compared to conventional engines, during which the fixed oxygen atom in the premixed ethanol and gasohol can improve combustion in fuel rich regions, which reduces the smoke formation. In particular, from the graph it is observed that the smoke emission concentrations are smaller with premixed gasohol about 30% than the ethanol fuel at the full load condition. The smoke emissions are significantly reduced to 70% and 67% for premixed gasohol and ethanol fuel as compared to normal mode of operation. The other reason for the reduction in smoke emission at the part loads is due to the disappearance of rich regions of mixture around the fuel spray in diesel combustion and air fuel mixture becomes partially homogeneous. This may also due to the high latent heat of premixed fuel which increases ignition delay period. Smoke emission is decreased due to reduced quantity of diesel that was injected for premixed combustion and the absence of localized fuel–air mixture is also the one of the reasons for the reduction in smoke emission.

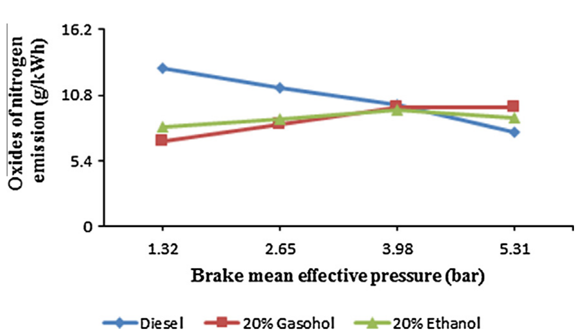


Figure 9 Variation of oxides of nitrogen with brake mean effective pressure.

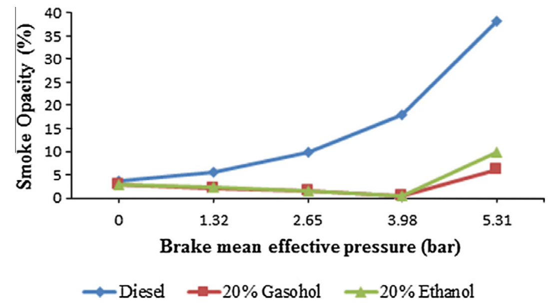


Figure 10 Variation of smoke opacity with brake mean effective pressure.

4. Conclusion

This research work has been done with comprehensive observance in mind about the current and future needs of HCCI. The result of this extensive research has brought out the application of dual fuel to the emerging needs of HCCI combustion. Various parameters such as exhaust gas temperature, oxides of nitrogen, carbon monoxide emissions, unburned hydrocarbons, oxides of nitrogen, smoke opacity, brake thermal efficiency and brake specific energy consumption are measured and analyzed. It is observed that there is a reduction in the emission level of smoke and oxides of nitrogen with the same power out of the conventional diesel engine. The salient details of the project work are as follows, which was concluded, based on the experimental results.

(1) In the experimental work, 20% premixed gasohol and ethanol fueled diesel exhibits longer ignition delay and hence the shorter combustion duration as compared to diesel. Thus the heat release rate is higher as compared to that of diesel.
(2) The brake thermal efficiency at full load is 31.12% and 31.86% for 20% premixed gasohol and ethanol fueled diesel, which is 4.65% and 3.69% higher as compared to diesel respectively.
(3) The carbon monoxide emissions and unburned hydrocarbon emissions were higher at part and medium loads and decreased gradually to the level of diesel at the full load.
(4) The smoke emission was significantly reduced to 70% and 67% from 20% premixed gasohol and ethanol fueled diesel compared to direct injected diesel fuel.
(5) Oxides of nitrogen emission were marginally reduced to 30% and 24% premixed gasohol and ethanol fuel.

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