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Experimental Investigation of PPCI Engine fuelled with Ethanol

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Abstract. Emissions are among those crucial factors that create a huge stress on the people related to engine industry. LTC concepts like HCCI and PPCI are proved as potential solutions for reducing NO_X emissions. The impact of ethanol premixed fraction on PPCI mode diesel engine has been investigated in this study. Various ethanol flow rates like 0.27, 0.45, 0.54, 0.72 and 0.81 kg/hr are fumigated in conventional single cylinder diesel engine. Ethanol is used due to the fact that it is available naturally as well as manufactured industrially. The combustion, emission and performance characteristics are plotted for each and every load with various premixed fractions and then the results are compared to pure diesel operation. It is clear from the results that the range of stable operation is being confined by ethanol for different premixed fractions. It can be safely said that for all stable operation, the NO_X emission is exceptionally less compared to pure diesel, though, the Un burnt Hydrocarbon & Carbon monoxide emissions are comparatively more. NO_X and smoke are decreased for all the premixed fractions at full load. At medium load operations, less NO_X, smoke opacity, CO and HC is obtained. Also, at low load operations, there is less NO_X and smoke opacity up to premixed fraction 0.3, but HC and CO emissions increases on further increase in the premixed fraction.

Keywords – Premixed fraction, homogeneous, HCCI, combustion, NOx, temperature.

1. Introduction

In the current scenario, most of the engine manufacturers produce petrol and diesel engines. The benefit of petrol engine is that, there is a uniform mixture of fuel and air inside the combustion chamber that decreases soot formation. But, at high compression ratio, they start to knock. So, the CR in an SI engine is limited to 10:1. On the other hand, in a diesel engine, power efficiency and fuel economy is high as there is higher CR possible and no throttling losses. But, CI engine causes high amount of NO_X and soot emissions. Hence, it is required to create a highly efficient and eco-friendly combustion system.

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1.1. Energy scenario of petroleum products in India
The production of petroleum based products in India during 2017-18 is being depicted in Fig 1. It is to be noted that the manufacture and distribution of diesel based automobiles have been more relative to the petrol based vehicles in the last few decades.

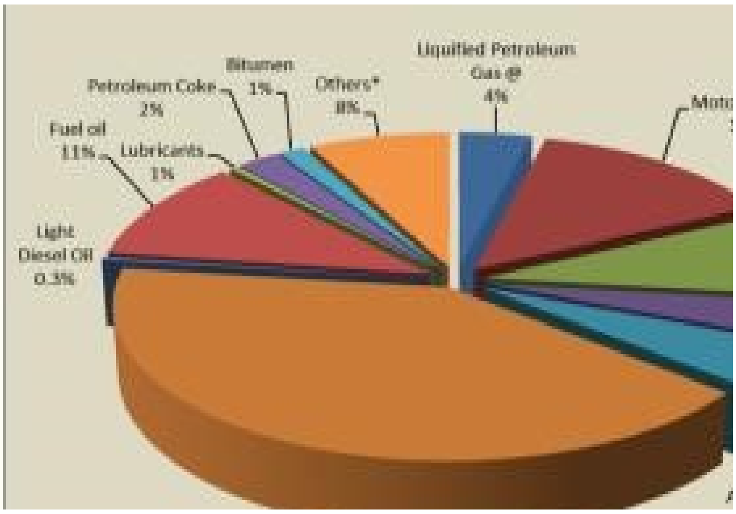

Figure 1 Distribution of petroleum products in India during 2017-18

1.2. Necessity of alternative fuels and other combustion technology

*1.2.1. Exhausting fossil fuel*Crude oil is exhaustible in nature. It will be exhausted in a few years.

*1.2.2. Emissions*Emission is the main cause that inspires the scientist community to work towards alternative fuels and new technology. Some very harmful pollutants are released from engine exhaust. These are the main pollutants:

(a) UHC
(b) CO
(c) NO_X
(d) Particulate matter

1.2.3. Green house effect(Global warming) It is the effect produced as a result of increase in green house gases near the bottom region of the Earth's atmosphere that create a layer. These layers of gases then decrease the outgoing thermal radiations, which in turn makes the Earth's climate warm. This greater than before temperature affects the natural cycle of climate change around us.

*1.2.4. Acid rain*Some pollutants like SO_2 and NO_2 combine with water molecules in atmosphere and produce acid, when this acid come back to the Earth in form of rain or snow, it is known as acid rain. The acid is known to corrode important architectures, ruin the crops, and degrade clean water.

1.3. Alternative fuels for CI engines
Petrol and diesel were the main source of liquid fuel for a long time. These were the products of fossil fuels. But now, the time has changed and some alternative renewable fuels are well-known in the industry, like alcohol.

Alcohol has a benefit that it can be produced from natural sources as well as manufactured in a laboratory. Hence, alcohol is an excellent alternate for fossil fuels. These are the main alcohols worth mentioning:

(a) Methanol
(b) Ethanol

1.4. HCCI concept

HCCI (Homogeneous Charge Compression Ignition) is an unconventional and new combustion technique. In this technique, air and fuel are mixed homogeneously outside the combustion chamber like in SI engine and then the charge is compressed until ignition like in CI engine. It enjoys the benefits of both the conventional combustion techniques. It has low soot emissions like that of SI engine and high power efficiency like that of diesel engine.

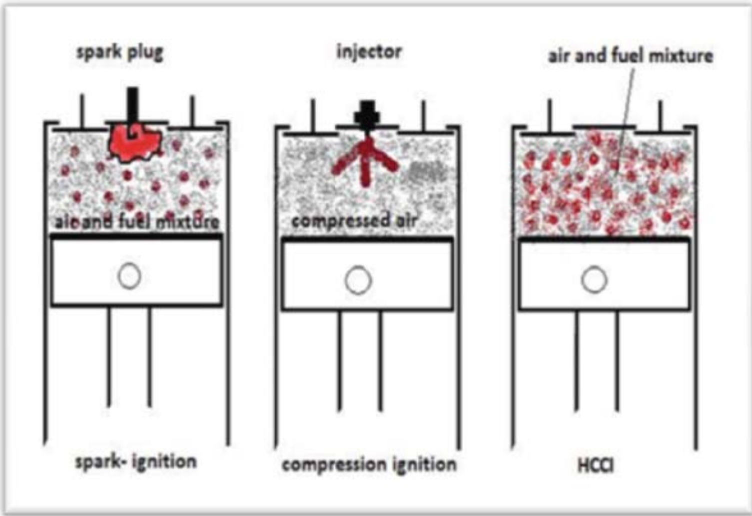

Figure 2 SI, CI and HCCI mode working of engine

1.5. PPCI concept

There are definite benefits of HCCI over conventional combustion modes but at the same time there are 2 challenges in HCCI mode. There is limited range of operation as engine misfires at low load and knock at high load, also combustion cannot be directly controlled.

PPCI mode eliminates the demerits of HCCI and adds an extra advantage of dual operation mode. In PPCI mode, the entire fuel isn’t homogeneously mixed with air, some fuel gets mixed outside combustion chamber and then injected in the chamber by port fuel injection technique, and some fuel gets injected in the conventional direct injection method. Both the quantities can be adjusted to get the optimum results in terms of emission and efficiency. Various research have been done to find out the benefits of such partially premixed combustion mode [24, 25, 26]

Rakesh Kr Maurya et al. [7] prepared the homogeneous charge using PFI technology. 2 cylinder engine was taken where 1 cylinder was working on HCCI and the other one was left as a regular diesel engine. The speed of engine was kept fixed as 1500 rpm for achieving the stable HCCI combustion. Experiments were carried by changing the equivalence ratio and inlet charge temperature. The highest indicated thermal efficiency for ethanol obtained was 44.78 % and max. IMEP of 4.3 bar was found at air fuel ratio 2.5 and intake air temperature 120^{o}C.

SrinivasPadala et al. [26] also used port fuel injection technology for injecting ethanol in the intake air. Dual fuel technique was implemented on a single cylinder diesel engine. Diesel was injected conventionally in the chamber. They wanted to investigate the impact of diesel injection timing along with ethanol as premixed fuel on the engine emission and performance. According to their findings, ethanol replaced 60 % of the energy contributed previously by the diesel, and efficiency was increased by 10% over the conventional mode operation.

The researches above used costly micro controllers in order to supply ethanol by port fuel injection while in the present study a basic inexpensive arduino based circuit was employed for ethanol injection. A single cylinder diesel engine is transformed into PPCI mode and experimentation is carried out using ethanol as port fuel while diesel as direct injected fuel.

2. Experimental methodology

Ethanol as mentioned earlier has lean burn properties along with high latent heat of vaporization, is most suitable for low temperature combustion technology like PPCI. Hence, ethanol is used as a premixed fuel.It also allows a denser fuel–air charge. The diesel is used for direct injection, as a complementary fuel for expanding the operation range of engine in case of misfires or knocking. The speed of the engine was kept fixed as 1500 rpm with help of mechanical governor.

Table 1 Characteristics of Ethanol versus Diesel

CHARACTERISTICS	ETHANOL	DIESEL
Formula	C_2H_5OH	$C_{12}H_{26}$
Cetane no.	5-12	46-51
Density at 20°C (kg/m^3)	790.7	820-900
Latent heat of vaporization (kJ/kg)	840	270
Stoichiometric A/F Ratio	9.0	14.5
Lower heating Value of Fuel (MJ/kg)	26.9	42.5
Specific gravity	0.785	0.84-0.88
Self-ignition temperature	423	210
Molecular weight (kg/kmol)	46	170
Boiling temperature at 1 atm (°C)	78.3	190-280

2.1. Experimental set-up

A Kirloskar made single cylinder diesel engine attached with an eddy current dynamometer was used. The engine can be loaded with desired load using the dynamometer consisting of a controller. All the engine data associated with heat release, power, pressure, mass fraction burnt with respect to crank angle and load is captured by Data Acquisition system. Diesel fuel consumed was measured using a fuel level indicator. The flow rate of intake air in the engine was measured using a U-tube manometer and an orifice-meter.

Table 2 Specifications of theengine

Maker	Kirloskar
Model	TV1
Type	Water Cooled,Four Stroke, Diesel Engine
Compression Ratio	17.5:1
Cubic Capacity	661
Bore and Stroke	8705mm × 110mm
Piezo Sensor	Make PCB Piezotronics, Model HSM111A22, Range 5000 psi, Diaphragm stainless steel type & hermetic Sealed
Opening of Inlet Valve	4.5˚ before TDC
Closing of Inlet Valve	35.5˚ after BDC

Opening of Exhaust Valve	35.5˚ before BDC
Closing of Exhaust Valve	4.5˚ after TDC
Fuel Injection Starts	23˚ before TDC
Rated Power	5.2kW @ 1500rpm
Drum Brake Diameter	185mm
Temperature Sensor	Make Radix Type, Ungrounded, Sheath Dia. 6mmX110mmL, SS316, Connection 1/4" BSP(M) adjustable compression Fitting
Eddy Current Dynamometer	Model AG10, of Saj Test Plant Pvt. Ltd.
Load Sensor	Make:SensotronicsSanmarLtd., Model 60001, Type S Beam, Universal, Capacity 0-50kg
Crank Angle Sensor	Make:Kubler-Germany Model 8.3700.1321.0360 Diameter: 37mm Shaft Size 6mm X Length 12.5mm
Orifice Diameter	20mm

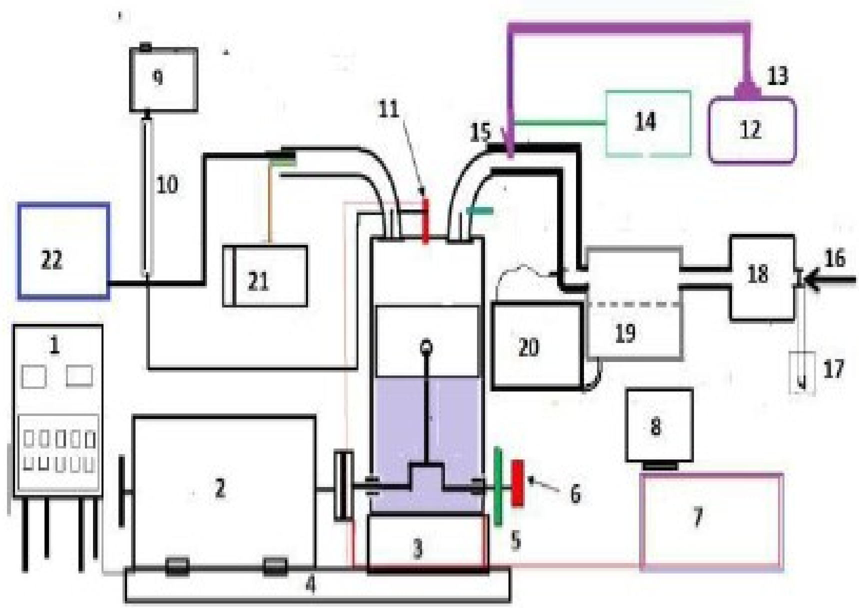

Figure 3 Schematic layout of experimental set up

1.Load cell	9. Fuel tank	17.Orifice meter
2.Alternator	10. barrette	18.Air box
3.Engine	11.Injector	19.Heater
4.Engine bed	12.Secondary fuel tank	20.Temperature controller
5.TDC sensor	13.Fuel pump	21.Exhaust gas analyzer
6. Encoder	14.ECU	22.Smoke meter
7. DAS.	15. Port injector	
8.Computer	16. Intake air	

2.2. Premixed fraction

Premixed fraction is the fraction of energy contributed by ethanol that is the premixed fuel in the whole supplied energy.

$$\boldsymbol{PF = \frac{Premixed\ Fuel\ Energy}{Premixed\ Fuel\ Energy\ + Direct\ Injected\ Fuel\ Energy}}$$

Premixed fuel energy = (mass flow rate of ethanol)*(CV of ethanol)
Direct injected fuel energy = (mass flow rate of diesel)*(CV of diesel)

2.3. Engine adjustment

In order to operate engine in PPCI mode, certain modifications had to be done. Intake manifold is equipped with specially designed port fuel injection system. PFI consists of 5 main parts, solenoid fuel injector, and a control circuit for injector, a program for operating the circuit, fuel tank and a fuel pump.

An electronic solenoid valve works as an electronic fuel injector which allows or doesn't allow the fuel to pass through in accordance with the programming. A predetermined quantity of pressurized fuel is allowed by the fuel injector into the system. In present study, the fuel injector used is setup with constant 3 bar fuel pressure.

In place of ECU unit, a simple arduino board is used to control the timing plus the quantity of fuel.A computer program is written to adjust the pulse width and that program is burnt in the ECU. The quantity of fuel to be injected is decided by the pulse width. The program is modified again and again to conduct the experiment with different quantities of fuel injection.

A DC electric fuel pump (12 volt) is kept in the fuel tank to boost ethanol from tank to PFI system. The pump maintained a 3 bar constant pressure inside the fuel line.

2.4. Experimental process

Ethanol is injected by PFI technique along with intake air in the suction stroke. The homogeneous mixture of ethanol and air gets compressed in the compression stroke. Then at the end of compression stroke diesel is injected conventionally. The amount of diesel was controlled by a mechanical governor in order to keep the constant speed of 1500 rpm.

In this study, these mass flow rates of ethanol 0.27, 0.45, 0.54, 0.72 and 0.81 kg/hr are injected.

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2.5. Calculations Involved

The mass flow rate of Diesel $\frac{dM_I}{dt}$ for each load and for every mass flow rate of port fuel i.e. ethanol, is provided by the Data Acquisition System integrated with our engine, using fuel level indicator attached to the engine. This quantity is given in Kg/hr.The mass flow rate of Ethanol $\frac{dM_E}{dt}$ is kept constant for one set of readings for each load and hence diesel fuel has to be automatically adjusted to give a uniform speed of 1500 rpm with the help of governor mechanism. This mass flow rate is then changed to some other value and kept constant for another set of readings.

The ethanol quantity is controlled by the pulse given by microcontroller. The program burnt in the ECU works on millisecond (ms), hence, in order to find the relationship between amount injected in Kg/hr by the injector and ms passed, we have to perform a bench testing of solenoid injector.
For 1500 rpm of a 4 stroke engine, there will be 750 injections in a minute, i.e.

$$No.of\ Injections\ per\ minute = \frac{1500}{2} = 750\ Injections\ per\ min$$

$$750\ injections\ per\ \min = \frac{750}{60 * 1000} = \frac{1}{80}\ injections\ per\ msec$$

It means to maintain 1500 rpm we have to inject port fuel once in every 80 msec.
The bench testing of our solenoid injector gives the result as it can inject 0.00253 mL/ms at 3 bar pressure setup used in the experiment. So we need 5 different pulses for 5 different flow rates required in this study, which works in following loops:-

1) 3 ms open---77 ms closed
2) 5 ms open---75 ms closed
3) 6 ms open---74 ms closed
4) 8 ms open---72 ms closed
5) 9 ms open---71 ms closed

Taking into account the density of Ethanol as 790.7 kg/m^3, the following table provides the flow rates of Ethanol $\frac{dM_E}{dt}$ in kg/hr:-

Table 3 Mass flow rate calculations for ethanol

ms	ms* mL/ms	mL/hr	kg/hr
3	3*0.00253	341.55	0.27
5	5*0.00253	569.25	0.45
6	6*0.00253	683.1	0.54
8	8*0.00253	910.8	0.72
9	9*0.00253	1024.65	0.81

Then, the PF can be evaluated by the formula

$$PF - \frac{\frac{dM_E}{dt} * (CV)ethanol}{\frac{dM_E}{dt} * (CV)ethanol + \frac{dM_D}{dt} * (CV)diesel}$$

Where,
$(CV)_{ethanol}$ =26.9 MJ/kg

$(CV)_{diesel}$ = 42.5 MJ/kg
The heat release follows 1st law of thermodynamics for a closed system:

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

Brake Thermal Efficiency (η) is evaluated by the formula

$$\eta = \frac{Brake\ Power}{\frac{dM_E}{dt} * (CV)ethanol + \frac{dM_D}{dt} * (CV)diesel}$$

3. Results and discussions

An average of 60 cycles has been taken in every experiment to avoid cyclic variations during the combustion analysis. The results of experimental investigation of combustion, performance and emission characteristics of modified engine are plotted then compared them with conventional diesel operation.

3.1.Combustion Characteristics

Combustion characteristics vary for different load operations and for different premixed fractions. Full load operation is performed at 5.2 kW @1500 rpm. In the present study, the results shown are at 80% of full engine load as combustion results are best shown at 80% of full load.

3.1.1.Cylinder Pressure versus Crank Angle diagram

It is certain that the variation of cylinder pressure in the combustion chamber is connected to the combustion characteristics of test fuel. The variations in the cylinder pressure with crank angle at 80% load for various premixed fractions are shown in fig 4. The beginning of injection of diesel was fixed at 23° Crank Angle before TDC, while ethanol was inducted with the air. It is obvious from the figure, that irrespective of the PF, the engine cylinder pressure increases for CI engine from compression stroke to power stroke of the engine.

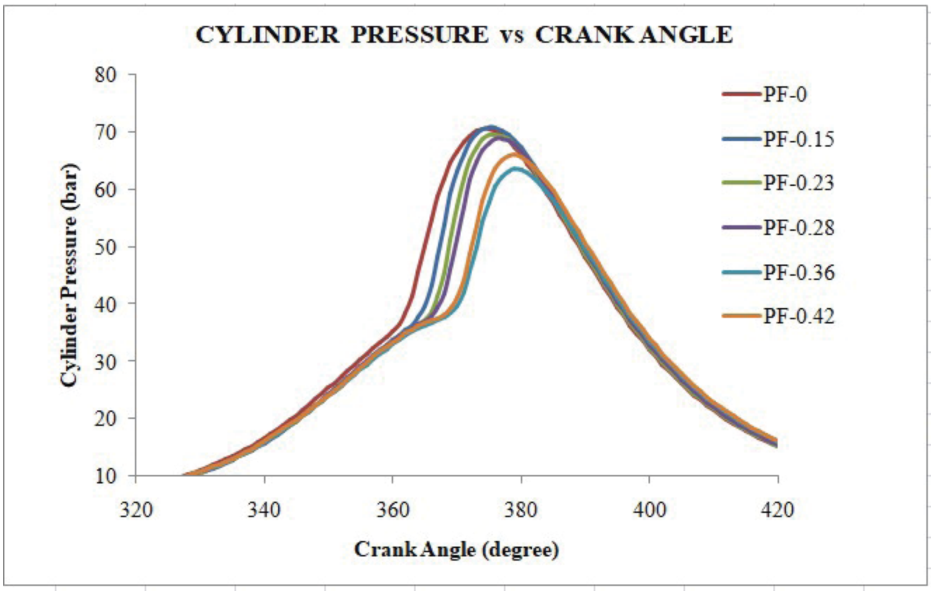

Figure 4 Variations in the cylinder pressure with crank angle at 80% load

It is required that the maximum pressure inside the cylinder occurs near TDC for more power output. For pure diesel i.e. premixed fraction=0), the cylinder pressure is seemed to be highest and close to TDC also. When the ethanol premixed fraction is increased, it follows with a reduction in cylinder pressure. It may occur because of vaporization cooling of ethanol [25]. It causes ignition delay. Hence, peaks are shifted towards right.

Further it can be noticed that, cylinder pressure drops with increase in premixed fraction shows only for lower premixed fraction. At higher values of premixed fractions, higher pressure rise is seen. It is due to the fact that at the end of compression stroke ethanol auto-ignites. For the values of premixed fractions 0.15, 0.23, 0.28, 0.36, and 0.42 the maximum Cylinder pressures are 70.90, 69.67, 68.93, 63.59 and 66.07 bar respectively. The peak cylinder pressure at this load for diesel operation is obtained as 70.64 bar. Further rise in PF at this load will cause misfire.

3.1.2. Heat release rate versus Crank angle diagram

HRR is among those crucial factors that characterize the combustion process. It can be defined as the measure of rapidness with which the chemical energy of fuel gets converted into heat. The calculation of heat release rate depends on cylinder pressure and the rate of pressure rise. The plot of HRR vs. crank angle at 80 % load for various premixedfractions is shown in fig.5.
There is no heat release during compression. It occurs once the fuel injection starts. When auto ignition occurs, the HRR increases quickly as the ethanol starts to burn. The combustion of ethanol gives a distinctive height to the HRR plot. By increasing the PF, peaks of HRR increase. It may be due to the fact that ethanol causes ignition delay because of its high value of latent heat of vaporization but once ignited, it gives high amount of energy in very short duration of time. Hence, peaks of HRR increases. Also ethanol is an oxygen rich fuel, due to availability of more oxygen, peaks of HRR increases. The shift of peaks towards right of the graph shows that more premixed fuel is burnt and less diesel fuel is burnt.

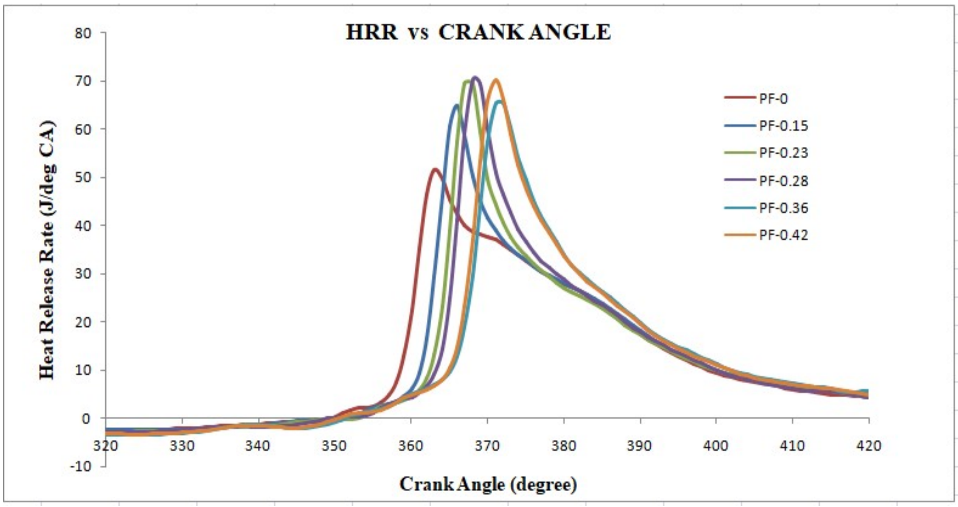

Figure 5Variations in the HRR with crank angle at 80% load

Maximum HRR for conventional diesel mode at 80% load is found to be 51.55 J/^{o}CA while with premixed fuel, the maximum HRR is found to be 70.25 J/^{o}CA at premixed fraction 0.42 at 80% load.

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3.1.3. Ignition delay

It is known as the time delay between beginning of injection of fuel and beginning of ignition. It depends on temperature of the mixture, A/F ratio, pressure and other properties of fuel. It is measured in crank angle degrees. The ignition character of the fuel is shown by the ignition delay. So, it becomes a crucial parameter that affects various engine parameters like efficiency, emissions etc. Excessively long ID simply represents that ignition is too late after TDC, which causes rapid cooling of the charge. This leads to partial combustion which ultimately reduces output power as well as thermal efficiency. Fig.6 depicts the plot between Ignition delay and PF for each and every load. It can be observed that more the load less is the ignition delay because of rich fuel supply.

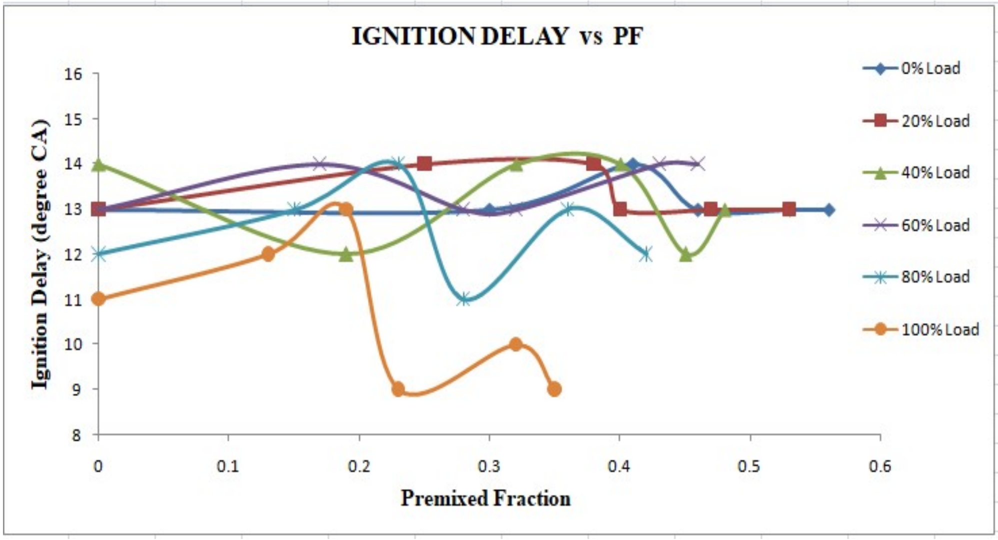

Figure 6 Variation of Ignition Delay with Premixed fraction

The ignition delay decreases from 13^{o} crank angle at PF 0.19 to 9^{o} crank angle at PF 0.23 at full load. This is because the advanced combustion of ethanol. The ignition delay reduces as A/F ratio begins to rise for medium loads. Moreover, larger value of ignition delay is observed with the rise of quantity of ethanol. It can be explained on the basis of large latent heat of vaporization of ethanol. Ethanol continues to absorb the thermal energy inside the engine for its vaporization. Hence, at first the Ignition delay increases with increasing premixed fraction. Further, decrease in the value of ID with higher premixed fractions can be explained on the basis that at the end of compression stroke, ethanol auto ignites.

3.1.4. Combustion duration

It's not easy to accurately define the beginning of combustion. However, it may be determined best by finding the change of slope of HRR. The combustion duration is known as the degrees of crank angle rotation from beginning of combustion to 90% heat release. Fig.7 depicts the plot of combustion duration vs. PF for each and every load. With increase in load, combustion duration decreases because of rich fuel in the combustion chamber, which in turn causes high rate of combustion. Due to this reason HRR and cylinder pressure both are high at higher loads.

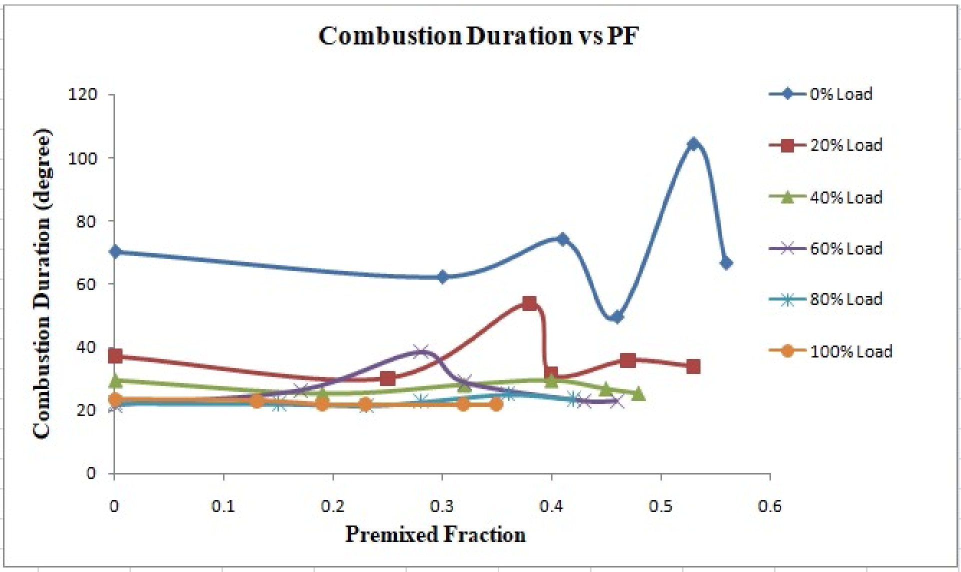

Figure 7 Plot of Combustion Duration versus Premixed fraction

The CD reduces with higher values of premixed fraction because of larger ignition delay. Also, higher oxygen content in ethanol and enhanced mixing of air-fuel leads to faster rate of combustion which decreases combustion duration. The rapid burning of the premixed fuel is the reason for shorter combustion duration.

3.2. Performance characteristics

3.2.1. Brake Thermal Efficiency(BTE)

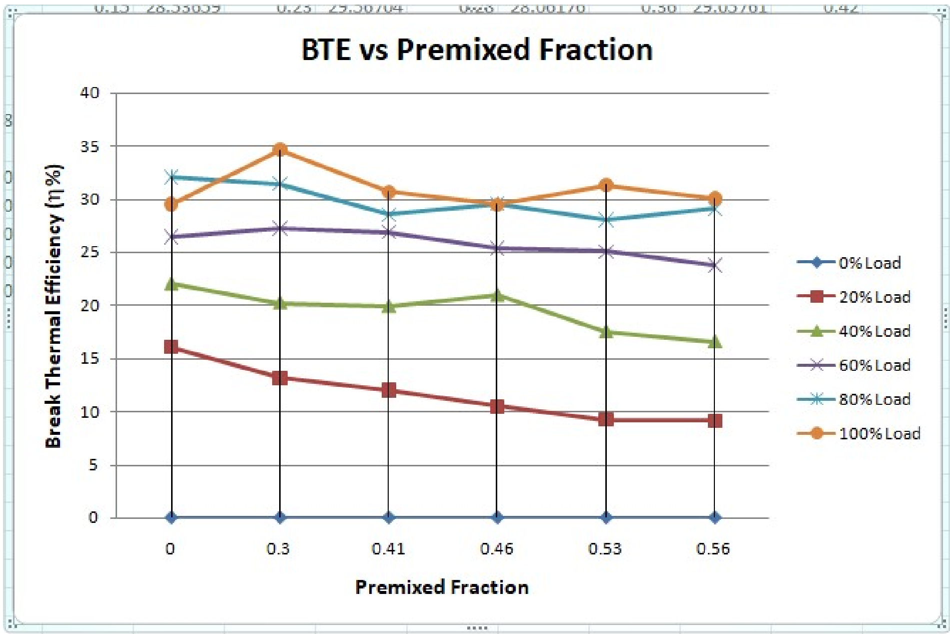

Figure 8 Plot of BTE with Premixed fraction

BTE is known as the ratio of useful power to the rate of heat energy supplied by combustion of fuels. The variation of efficiency with various premixed fractions under different engine loads, is depicted in figure 8.It can be noticed that, at small values of loads, BTE goes down up to 9.15%, while at higher loads, it goes upto 34.65%. The reason for the above observations can be the fact that at low loads, lean A/F mixture is formed which degrades the combustion. Also, the larger value of latent heat of vaporization of ethanol can cause quenching of the combustible charge leading to reduced efficiency.

3.2.2. Exhaust gas temperature
Exhaust gas temperature is the indirect depiction of BTE i.e. how efficiently engine is deriving useful power from fuel supplied. The higher value of EGT at same load would indicate more loss of heat in exhaust which in turn is indicating bad efficiency. The EGT is a function of cylinder temperature and expansion process.

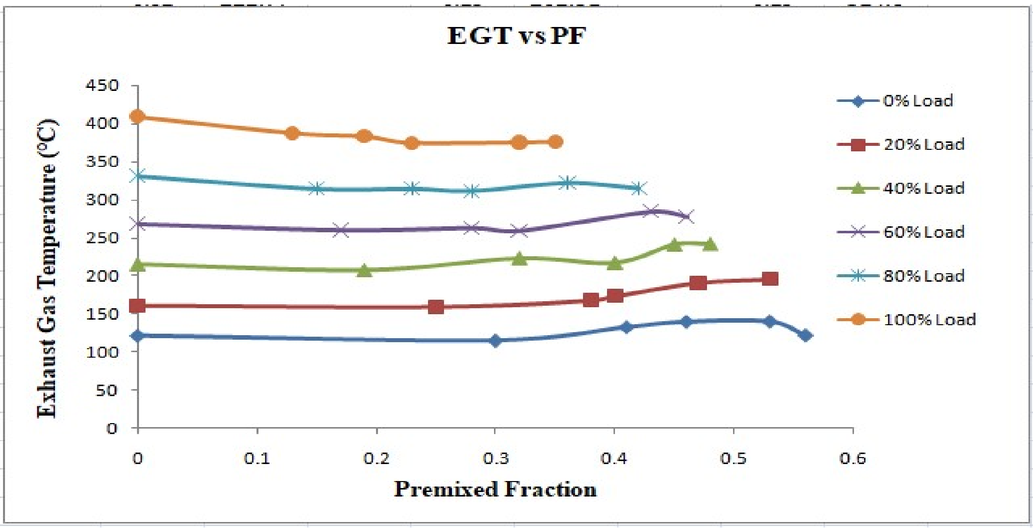

Figure 9 Variation of Exhaust gas temperature with PF

Fig.9 depicts the plot of EGT vs. PF for each and every load. Nearly proportional rise in EGT is seen with varying load for all the premixed fractions tested. This more value of EGT with higher loads can be explained because of extra fuel needed to generate additional power. Further, it may be observed that with increasing premixed fraction, the exhaust gas temperature slightly decreases for higher loads and remain almost constant for lower and medium loads. The reason may be that at larger value of loads, the cooling down of charge occurs due to high latent heat of vaporization of ethanol. Also, BTE is slightly improved which shows the heat loss in the exhaust is less. Hence, lower value of EGT. In case of lower and medium loads, on increasing the value of premixed fraction, the ignition delay increases i.e. late combustion, which means less conversion of heat to useful work. This balances out the slight improvement in BTE. Hence, the EGT is maintained nearly constant.

3.3.Emission characteristics

3.3.1 .Carbon monoxide (CO) emission
Partial combustion and heterogeneous A/F mixture are the reasons for high CO emissions. Fig.10 depicts the plot of carbon mono oxide vs. PF for each and every load. For pure diesel (PF=0), the CO reduces from 0.06% (at no load) to 0.01% (at full load). This is due to

complete combustion at higher loads. Homogeneity of charges increases with increase in premixed fraction so the CO should decrease but for lower loads, lower engine temperature causes inefficient combustion leading to higher CO emission on increasing PF. At 80% load, fig.10 indicates nearly fixed value of CO for all PF. This is because higher PF means more charge is there which means more chance of incomplete combustion, but this is balanced out with the fact that there will be more homogeneous A/F mixture inside, which leads to complete combustion. Hence, at 80% load, the CO emissions remain almost constant.

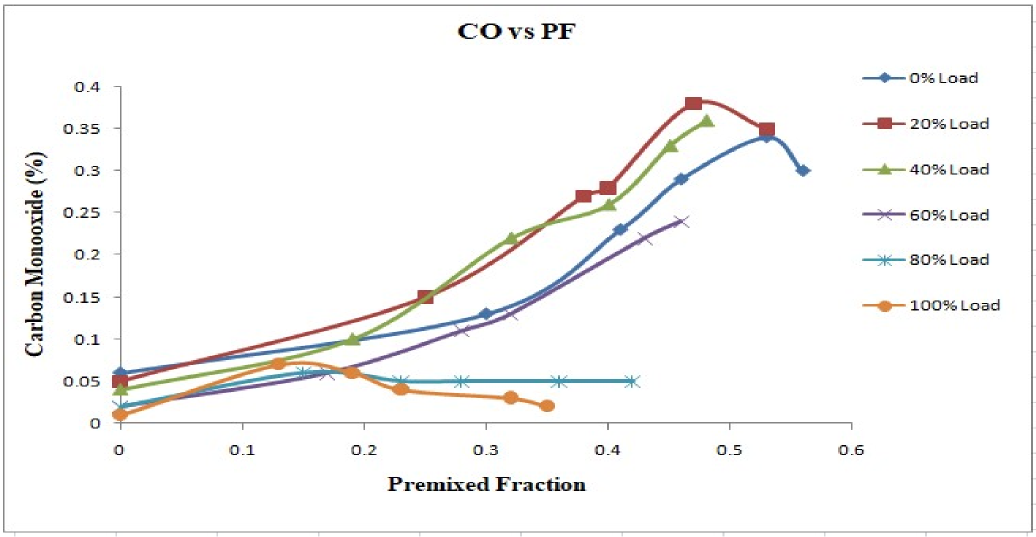

Figure 10 Plot of CO (%) with PF

The homogeneous mixture and higher engine temperature dominates and leads to complete combustion at 100% load, which ultimately leads to lower CO emissions. It is to be noted that for all loads, with increase in premixed fraction, the CO emission is more in comparison to regular diesel operation due to increasing amount of carbon inside engine.

3.3.2.Unburned Hydrocarbon emission

UHC emissions are one of the main demerits of premixed charges. It is caused due to condition of misfiring and partial combustion of hydrocarbon fuels. More the PF more is the UHC emission. Fig.11 depicts the plot of UHC emission vs.PF for each and every load. The larger value of latent heat of vaporization of ethanol slows down the vaporization and the mixing of fuel and air, causing degraded combustion at lower temperature. The degraded combustion increases the production of UHC and that rises with rising premixed fraction. It can be easily seen that UHC is more at lower value of loads and less at higher value of loads. This is caused because at low loads, more quantity of air, bad homogeneity of mixture and low EGT. With increasing loads at different premixed fractions, UHC emissions reduce asA/F mixture becomes more homogeneous. The UHC emission is quite low at full load. At full load, the UHC goes as high as 177 ppm only however, it goes upto 2735 ppm for low load.

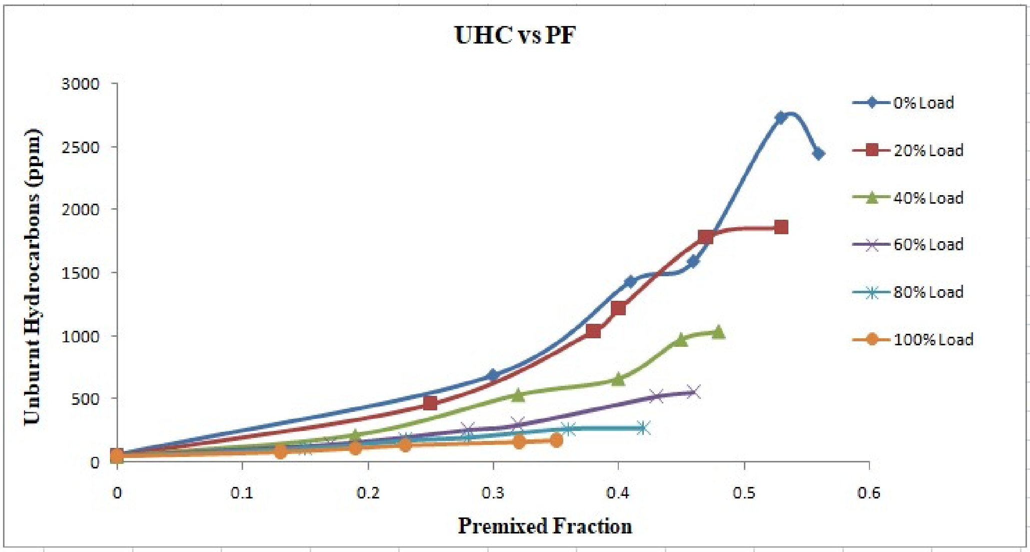

Figure 11 Variation of Unburnt Hydrocarbon (ppm) with premixed fraction

3.3.3. Nitric oxide (NO) emission

NOx are the emissions consisting of nitric oxide & nitrogen dioxide. The 2 main reasons of higher NO_X emission are more amount of oxygen available and high combustion temperature. Their reduction is the main aim of nearly all LTC operations, like PPCI technology.

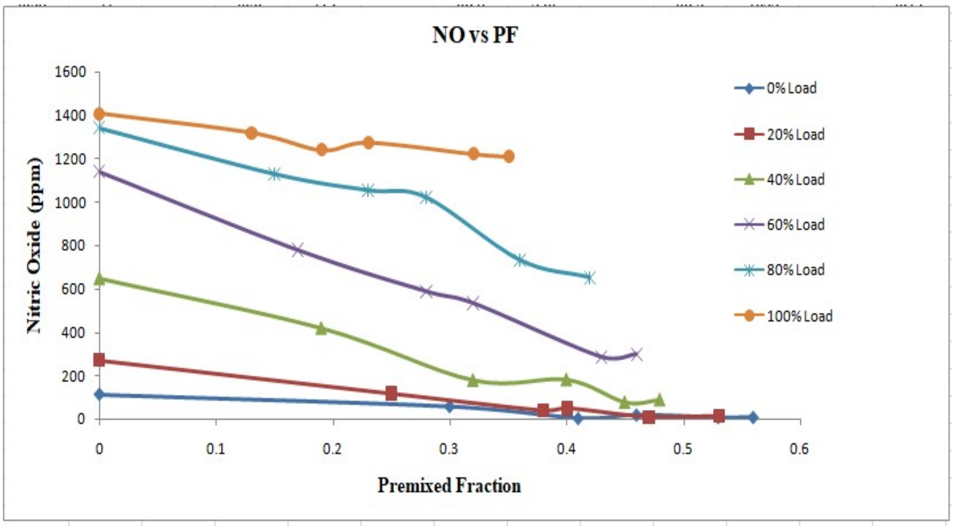

Figure 12 Variation of NO (ppm) with premixed fraction

Ethanol has a very interesting property of higher latent heat of vaporization which implies lesser combustion temperature causing less NO_X formation. Fig.12 depicts the plot of NO_X vs. PF for each and every load. At high engine load, the rich mixture tends to decrease the quenching effect, but in comparison to the pure diesel mode, now diesel has to burn in a mixture of air and ethanol. Hence, comparatively low oxygen is there to produce NO_X. There is a definite decrease in the NO_X for all load operations. It can be seen in the fig., the NO_X emission drops off for higher PF because high PF means LTC, which implies lower NO_X

values. It may be noted that the NO values are decreasing with increasing PF but increasing with increasing load at a particular PF. It is due to the fact that as load increases, engine temperature also increases, which causes higher NO_X values. The NO_X value reduces efficiently from conventional diesel mode at each and every load, which satisfies the purpose of using PPCI mode in place of diesel mode.

3.3.4. Smoke opacity

Smoke is caused by poor engine design parameters. Smoke opacity may be defined as the indirect measure of soot quantity in the engine exhaust. High value of smoke opacity is caused due to gathering of fuel as well as non homogeneity of charges. Smoke-meter just gives a number to the visibility through the black smoke emission using the physical principle of extinction of a light beam by scattering and absorption.

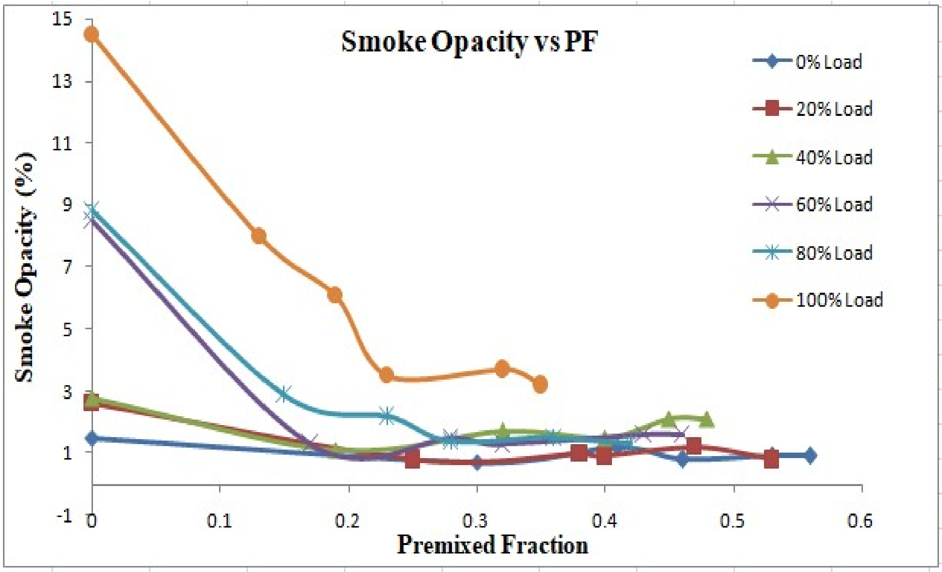

Figure 13 Plot of Smoke Opacity (%) with PF

Fig.13 depicts the plot of smoke opacity vs. PF for each and every load. It is usually proportional to the engine load because as load rises, the charge temperature at injection also rises, which ultimately reduces the ignition delay. The reduction in ignition delay restrains the thermal cracking and causes formation of soot. It can be noticed in figure that when PF is increased, smoke density decreases very much. For pure diesel mode, the smoke opacity varies from 1.5% (at no load) to 14.5% (at full load). At larger value of premixed fraction, this range is decreased as 0.9% to 3.2% respectively. At full load, it is obtained to be 3.2% (PF= 0.35) which is much smaller than full load diesel, which is 14.5%.

4. Conclusions

This study concludes that at full load operation (100%), reduction in NO_X and smoke opacity is achieved for all the premixed fractions. Hence, the purpose of LTC concept of PPCI mode is fulfilled as it decreases NOx and smoke compared to diesel engine mode. For CO and HC emissions at 100 % load, CO is greater in comparison to diesel mode but it decreases with increase in PF. While HC keeps on increasing with PF. But the increment in CO & HC is slight and insignificant in comparison to a greater drop in NOx and smoke emissions. At medium load operations, upto premixed fraction 0.48 engine performance and emission parameters are

definitely better. There is no unfavorable outcome. At low load operations, it can be used only up to PF=0.3, further rise in ethanol causes adverse effect on various engine parameters.
There are promising results of using ethanol as a premixed fraction in a PPCI engine. Also there are plenty of resources of ethanol available in India naturally as well as industrially. So, ethanol could be utilized as a port fuel in future for improving the diesel efficiency and reduction in the NO_X and other emissions.

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