Experimental investigation of Ethanol fuelled Partially premixed compression ignition (PPCI) Diesel engine

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Abstract: Control of emissions from IC engines applies a huge pressure to engine manufacturers. Low temperature combustion concepts such as HCCI, PPCI, and PCCI are promising solutions for reduction of both the NOx and soot particulate from diesel engine. In the present investigation, five different flow rates 0.21, 0.37, 0.51, 0.59 and 0.80 kg/hr of ethanol is injected by port fuel injection. This study investigates the effect of ethanol premixed fraction on PPCI and direct injection, combined combustion mode engine. The motivation of using ethanol fuel is that, it can be obtained from both natural and manufactured resources. The combustion, performance and emission parameter are evaluated for all loads with different premixed fractions and compared with diesel fuel operation. Based on the performance and emission parameters, it is understand that the injection of ethanol limits the stable operation range for different ethanol premixed fraction. In order to increase stable operation range, charge heating is used with different flow rates of ethanol. Results indicated that charge heating is beneficial solution for low load operation. For all stable operation range, the NO_X emission is found to be extremely low than that of diesel, however, the HC and CO emissions are relatively high.

I. INTRODUCTION

Today, majority of automotive industries manufacture gasoline and diesel engines. Both the engines are contrast to each other in terms of thermal efficiency, fuel economy and emissions. As the population of vehicles increases, there is a huge pressure on the engine manufacturers to apply new technologies which can reduce emissions with a better fuel economy. In an SI engine, the fuel and oxidizer is mixed homogeneously, which reduces soot emissions, so gasoline engines are soot free. But, in order to avoid knocking in them, compression ratio is strictly limited to 10:1. In a CI engine, the fuel is compression ignited and has no throttling losses. Hence, CI engines are superior in terms of power efficiency and fuel economy. But, the oxides of nitrogen (NO_X) and soot are considerable problems for CI engines. In order to reduce emissions and use variety of fuels, there is a need to develop highly efficient and environmental friendly combustion systems.

At present, the main pollutants from IC engines are the NOx, unburned hydrocarbon (HC), carbon mono oxide (CO) and soot. These pollutants are responsible for the local as well as global atmospheric pollution. Therefore, there are laws on emission standards, which limit the amount of each pollutant in the exhaust gas emitted by an automobile engine. CO_2 is not considered as a pollutant, but it is also responsible for the global warming. It can be reduced only by reducing fuel consumption which can be obtained only by improving engine efficiency.

Alcohol is obtained from both natural and manufactured sources. That's why alcohols are an attractive alternative fuel. Naturally, it obtained by fermentation process of glucose and manufactured industrially, by hydration reaction. The two most promising form of alcohols are ethanol and methanol.

Ethanol is a renewable fuel receiving more attention by many researchers. It has been used for a long time in many countries in the world. Brazil is one of the largest producer of ethanol. In the United States of America also, ethanol and gasohol fuel stations are available. Gasohol is the combination of ethanol with gasoline. Ethanol has important mixture combinations E85 (85% ethanol and 15% gasoline) and E10 (10% methanol and 90% gasoline). Gasoline mixed with ethanol to eliminate some problem like cold start and tank flammability. A small percentage of ethanol mixed with gasoline can be used in automobile engine without any engine modification. Ethanol obtained naturally by fermentation of grain or sugar, and from the hydrolysis of starch. Ethanol generates less HC emission than gasoline, but more than methanol.

Homogeneous charge compression ignition is a type of combustion in which fuel and oxidizer are mixed homogeneously outside the cylinder and then compressed to the point of auto ignition level in compression stroke. The homogeneous charge compression ignition engine incorporates the best features of conventional gasoline and diesel engine. HCCI produces gasoline like soot emission while diesel like power efficiency.

The two main challenges for the HCCI combustion, that is the direct control of combustion and limited operation range due to misfiring at low load and knock at high load. Partially premixed compression ignition (PPCI) is very useful in order to reduce disadvantages of HCCI especially for a single cylinder engine and enjoys the benefits to operate the engine in dual mode operation. According to this concept, some quantity of fuel is injected by port injection to achieve homogeneity of charge and some quantity of fuel is injected by conventional direct injection as the compensation to achieve combustion control. By adjusting the quantity of port injected fuel and direct injected fuel the optimum results can be obtained. Several studies have revealed the advantage of similar combined combustion mode. [2, 3, 4]

A. Objective of the Present Research Work

In this study initially the existing conventional diesel engine is converted into PPCI combustion mode. Experiments were conducted in the PPCI mode using ethanol as a premixed fuel. Ethanol is inducted in intake manifold, and some quantity of diesel is injected by conventional direct injection. By regulating the fractions of premixed ethanol and direct injected diesel, various premixed fractions obtained. The present study, investigates the effect of premixed fractions on the combustion, performance and emission characteristics of diesel engine.

II. EXPERIMENTAL DETAILS

A. Experimental procedure

Ethanol was injected in the intake manifold using port fuel injector. Along with intake air ethanol entered to the combustion chamber during the suction stroke. In the compression stroke, ethanol and air mixed homogeneously and got compressed. At the end of compression stroke diesel was injected conventionally. Ethanol is having low calorific value, as compared to diesel hence, diesel was used as compensation. In the present investigation, flow rates 0.21, 0.37, 0.51, 0.58 and 0.79 kg/hr of ethanol injected by port fuel injection. The combustion, performance and emission characteristics were evaluated for all loads with different premixed fraction.

B. Premixed fraction

The premixed fractions defined as the ratio of energy contribution of premixed fuel to whole energy contribution. That is

$$\frac{Primixed\ fuel\ energy}{Primixed\ fuel\ energy + Direct\ injected\ fuel\ energy}$$

whereas,

Premixed fuel energy = (mass flow of premixed fuel)* (calorific value of premixed fuel)

Direct injected fuel energy = (mass flow of direct injected fuel)* (calorific value of direct injected fuel)

C. Description of the test engine

This section describes the complete experimental setup such as engine, coupled with alternator, emission test bench, data acquisition system, including piezoelectric pressure transducer and crank angle encoder. Some modifications have been done to achieve homogeneous combustion. The experiments were carried out on a single cylinder, four stroke, naturally aspirated CIDI engine and the schematic layout of the experimental arrangement is depicts in Fig. 1.

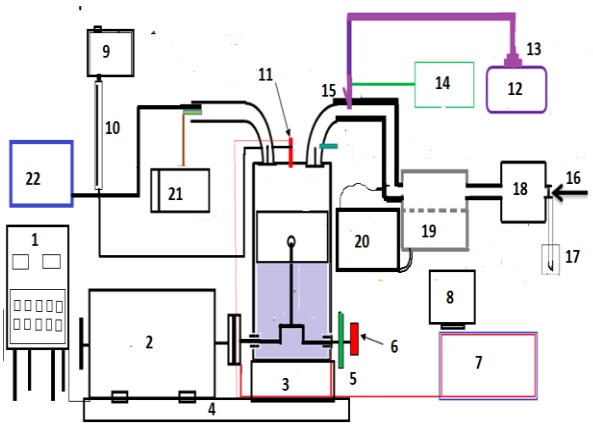


Fig. 1 Schematic layout of experimental set up

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 analyser
6. Encoder	14. ECU	22. Smoke meter
7. DAS.	15. Port injector	
8. Computer	16. Intake air	

The cylinder pressure history, data acquisition and combustion analysis is done using a Lab view based program. All data related to pressure heat release mass fraction burnt power efficiency with respect to load and crank angle provided by this system. A fuel level indicator was used for measuring the diesel fuel consumption. An orifice-meter and a U-tube manometer were used to measure the intake air flow rate of the engine. An air box fixed into the intake manifold of the engine maintains a constant air flow and eliminates cyclic fluctuations.

A K-type thermocouple was installed to measure the exhaust gas temperature. For the analysis of emission AVD DI gas 444 exhaust gas analyser connected to the exhaust pipe. It gives all emission quantity present in gas like NO, CO, CO_2, and HC. AVL 437 smoke meter is used to measure exhaust smoke.

D. Engine modification

For achieving the PPCI combined combustion mode, it was required to operate the engine with some modification. In order to this, port fuel injection system and air pre-heater system were included in the intake manifold the fuel premixing system consists of

1. Fuel injector
2. Injector control circuit
3. Program for electronic circuit
4. Fuel pump and fuel tank

A fuel injector is basically an ECU controlled solenoid valve, which open and closes to allow fuel pass through it. Fuel injector releases a controlled amount of pressurized fuel into the system. The injector is fed a constant supply of power and ECU provides a negative trigger to turn it on at the required time and for required interval.



Fig 2 Fuel premixing system Fuel pump and fuel tank [1], Fuel injector [2], Injector control circuit [3], Program for electronic circuit [4]

A program is required for ECU to control the injection timing, which in turn triggers the fuel injector. At the place of ECU unit, a microcontroller was used here to control both the timing and quantity of fuel. The pulse width decides quantity of fuel injected.

An electrical fuel pump of 12 V DC supply is fitted inside fuel tank is used for boosting the premixed fuel from tank to injector. Property of this pump is to maintain a constant pressure inside fuel line if the pressure increases more than a limit pressure, secondary valve attached in pump automatic open and release the excess pressure.



Fig 3 Intake air pre-heating system.

Fig. 3 shows the photograph of the intake air pre-heater system used in study. The intake air pre-heater is controlled by temperature controller with feedback control, which keeps constant air temperature in intake manifold. The heater consist heating element, which is inserted into cylindrical box of GI sheet. The heating element wound over the ceramic structure. One inch thick glass wool insulation done outer layer of the heater, that provides both insulation as well as cushioning.

Fuel injector

E. Uncertainty Analysis

The total percentage of uncertainty of this experiment = square root of [(uncertainty of TFC)2 + (uncertainty of BP) 2 + (uncertainty of BSFC) 2 + (uncertainty of Brake thermal efficiency) 2 + (uncertainty of CO) 2 + (uncertainty of CO2) 2 +(uncertainty of NO) 2 + (uncertainty of Smoke meter) 2 + (uncertainty of UBHC) 2 + (uncertainty of Pressure pick up) 2 + (uncertainty of crank angle encoder) 2 + (uncertainty of port injector) 2

= √{(1.5)2+ (0.2)2 +(1)2 + (1)2 + (0.2)2 + (0.15)2 + (0.2)2 + (0.2)2 + (0.15)2 +(1)2 + (0.5)2} = ± 2.38

III. RESULTS AND DISCUSSION

Module I: Preliminary investigation on the combustion, performance and emission parameters for PPCI combined combustion mode with the naturally aspirated air without charge heating

A. Combustion Characteristics

Combustion characteristics with respect to a premixed fraction, is varying for low, medium and full load operations. Full load operation has been carried out at 4.4 kW, medium load at 3.3 kW and low load operations done at 2.2 kW and 1.1 kW loads. The start of injection for diesel was set 23^{o}CA bTDC, while ethanol was inducted with the air. By increasing the ethanol premixed fraction leads to a decrease in cylinder pressure near TDC. This is due to the vaporisation cooling of ethanol [4]. Ethanol has a high latent heat of vaporization and it requires a high amount of heat to absorb from the cylinder to change phase results in substantial cooling of the fresh charge called vaporisation cooling of ethanol. For full load, drop in the cylinder pressure shows only for lower ethanol fraction. At higher ethanol fraction, advance pressure rise is observed. This is because of self ignition of ethanol at the end of compression stroke. The heat release rate shows the intensity of rapid combustion.. Auto ignition of ethanol takes place before diesel ignition at full load. Combustion is divided into two phases, by increasing the premixed fraction first phase HRR increases and second phase decreases. It is due to the increase of fuel burnt in the premixed mode and reduction of fuel burnt in diesel mode.

The auto ignition temperature of ethanol is high and it has not able to reach at self ignition level for medium

and low loads. Because of this, ignition is started by diesel fuel. In the premixed mode, diesel burns after diffusion in the mixture of air and ethanol, not only pure air. Moreover, ignition delay increases because of lower pressure and temperature in cylinder. the premixed fraction increases the maximum heat release rate increases. Ignition delay is an important factor to understand the combustion process. It is the time difference between start of injection and start of ignition measured in degree crank angle. The delay increases with the increasing amount of ethanol inducted with the air. It is due to the higher latent heat of ethanol. Fig 4 shows the Variation of ignition delay with respect to premixed fraction. The delay increases from 16^{o} CA to 24^{o} CA for lower loads. It may have two reasons first vaporization cooling and leaner air fuel ratio. For medium load ignition delay reduces up to 21.7 ^{o}CA. At full load ignition delay drop from 21.8 ^{o}CA to 11.6 ^{o}CA for the premixed fraction 0.1 to 0.4 respectively. This may be due to advance combustion of ethanol. Actually, this ignition delay is only for diesel and the early combustion of ethanol provides heat to reduce the ignition delay for diesel fuel.

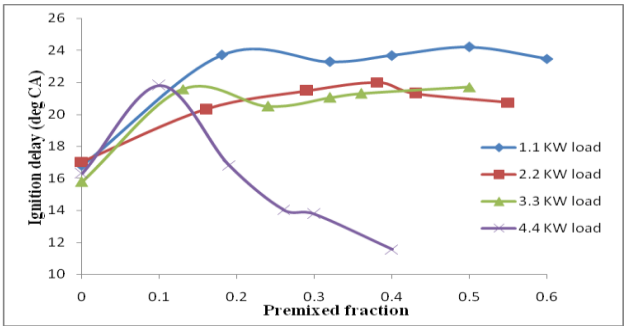


Fig 4 Variation of ignition delay with premixed fraction

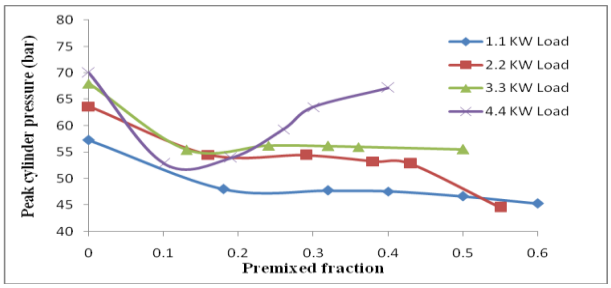


Fig 5 Variation of peak cylinder pressure with premixed fraction

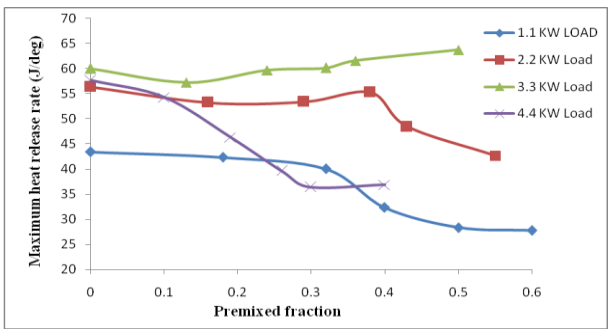


Fig 6 Variation of maximum heat release rate with premixed fraction

B. Performance Characteristics

Brake specific energy consumption is an important parameter to observe the performance of engine. It is the product of brake specific fuel consumption and calorific value of the fuel. Fig 7 shows the variation of BSFC with respect of premixed fraction. Generally BSFC decreases with the increase in the load. For diesel operation, the value of BSFC is 22.0, 14.3, 12.8, 12.04 MJ/kWh at 1.1, 2.2, 3.3, and 4.4 kW load respectively. Increasing premixed fraction, value of BSFC almost same for higher load, but for lower load it consume more energy. The BSFC increases 35% with premixed fraction 0 to 0.6 at 1.1 kW load and increases 15% from premixed fraction 0 to 0.55 at 2.2 kW load. This is due to incomplete combustion and less contribution of ethanol burned energy.

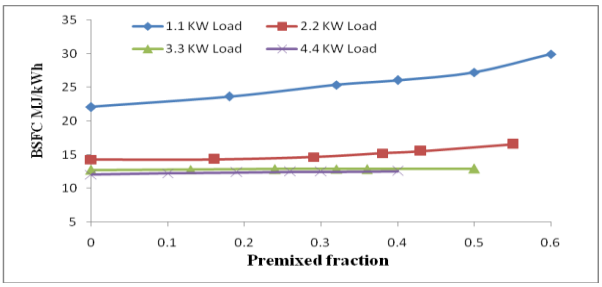


Fig. 7 Variation of BSFC with premixed fraction

C. Emission characteristics

For diesel operation, the CO emission decreases 0.016% to 0.01% with increasing load 1.1 kW to 4.4 kW. Reasons for the higher CO emission are to incomplete combustion, heterogeneity of air fuel mixture and temperature rise inside the cylinder. Homogeneity of charges increases with premixed fraction but for low loads combustion inefficiency, leads to increase in the CO emission. At full load, the homogeneous mixture and two stage heat release leads to lower CO emission. At middle load, Fig. 8 shows almost constant value for all premixed fraction. For all load except full load premixed fraction, the CO emission is high in comparison with diesel operation. The unburned hydrocarbon (HC) emission is the result of oil film absorption, crevice volume misfiring condition or incomplete combustion of hydrocarbon fuels. The HC emission is one of the main disadvantages of premixed charges. The HC emission is directly proportional to ethanol premixed fraction. Fig. 9 shows the variation of HC emission with premixed fraction for all loads. At higher premixed fraction high increment occurs for low load is the result of ethanol quenching at low combustion temperature in the cylinder. Especially after increasing premixed fraction up to 0.04, value of HC emission increasing more than 8 times for 1.1 kW load. The HC emission is very low at full load operation. The maximum value of HC emission at full load is 9 ppm only but for low load it goes to 90 ppm.

The NO emission reduction is the prime objective of all LTC mode operations, either HCCI or PPCI combined

mode. Fig. 10 shows the variation of NO with respect to premixed fraction for all loads. It is seen from figure, the NO emission reduces by increasing premixed fraction due to increasing premixed fraction leads to low temperature combustion. At higher premixed fraction, it shows lesser value at full load which decreases from 121 ppm to 26 ppm with premixed fraction increases up to 0.04. At low load, the NO emission decreases at 2 ppm up to premixed fraction 0.06. The drastically reduction may be due to very low temperature inside the cylinder results is inefficient combustion of ethanol. For the medium load also, it gives moderate result, value of NO emission reduces from 92 ppm to 28 ppm.

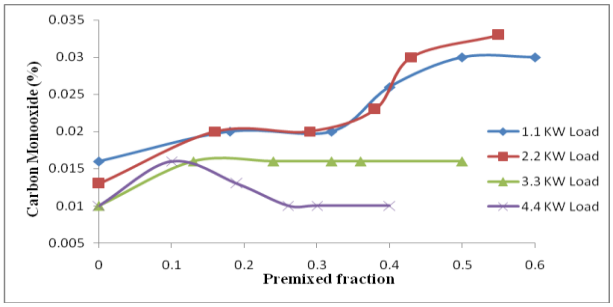


Fig. 8 Variation of carbon monoxide with premixed fraction

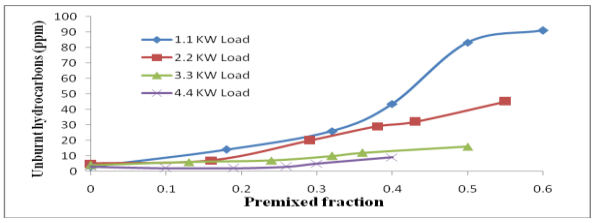


Fig.9 Variation of unburned hydrocarbon with premixed fraction

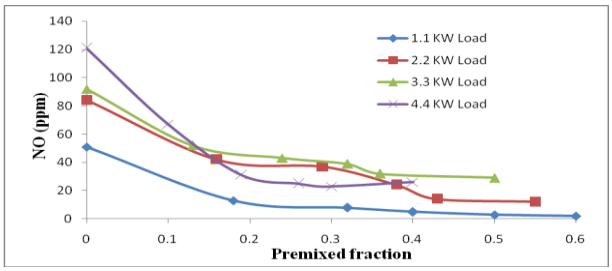


Fig 10. Variation of nitric oxide with premixed fraction

Module II: The combustion, performance and emission parameter for PPCI combustion mode with the naturally aspirated air, heated at 110 degree Celsius.

D. Combustion characteristics

From the first module results, it is clear that the vaporization cooling of ethanol is a big problem at low loads operation because ethanol required high amount of heat from combustion chamber to vaporize. At low loads, if the sufficient heat is not available inside the cylinder, ethanol diffuses in cylinder without contributing energy. Due to this, high BSFC consumption and HC emission at higher premixed fraction. In order to improve the operating limit of premixed mode at low loads, investigated the effect of charge heating on 1.1 kW, 2.2 kW and 3.3 kW loads. Fig 11 depicts the variation of ignition delay with respect to the premixed fraction. Charge heating provides, a sufficient heat to vaporize ethanol, because of this increasing ethanol premixed fraction, leads to shorter ignition delay. Ignition delay reduces for the all load operation, after charge heating. Large reduction in ignition delay is due to two stage combustion at 3.3 kW. For 3.3 kW load ignition delay decreases up to 11.3° CA at premixed fraction 0.24. For 2.2kW and 1.1kW loads the delay not varies more with increasing premixed fraction but its values is 13% to 18% less than the without charge heating values. The peak cylinder pressure represents the combustion condition inside the cylinder. It is depends upon the ignition delay and fuel quality. Fig. 12 depicts the variation of peak cylinder pressure with respect to the premixed fraction in charge heating condition. The maximum heat release rate decreases 40 to 31.1 J/ ^{o}CA with increasing premixed fraction 0.24 to 0.32 at 3.3 kW load. This is due to two stage heat release separately, for ethanol and diesel. Without charge heating it happens only for full load but with charge heating, also happens for 3.3 kW. At 1.1 kW load, heat release rate decreases 37 to 32 J/deg with increase in premixed fraction 0.35 to 0.62, because of a inefficient combustion.

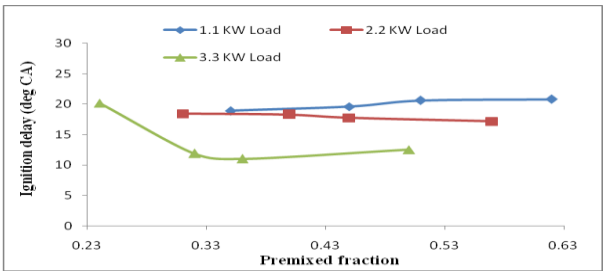


Fig. 11 Variation of ignition delay with premixed fraction

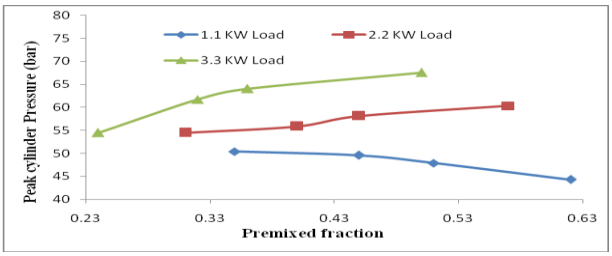


Fig. 12 Variation of peak cylinder pressure with premixed fraction

E. Performance characteristics

Earlier result shows that BSEC increases dramatically with increasing premixed fraction at low load as the result of incomplete combustion and hence, large increment in UHC emission. Charge heating is used to reduce BSFC at low loads.

At 3.3 kW load, there is almost same result in compare to without heating, but for 2.2 kW BSFC decreases in sufficient quantity comparison to without charge heating for respective premixed fraction. BSEC improve also for

1.1 kW load but only up to 50% premixing, further increase in the premixed fraction leads high increment in BSEC.

F. Emission characteristics

Charge heating is reduce the HC and CO emissions and improves the BSEC specially 2.2kW load. Moreover the HC emission increases with the increase in the premixed fraction. At 1.1 kW load, again UHC dramatically increases at higher premixed fraction but this times values are less than without charge heating values. Very small increment occur in NO with increasing premixed fraction in charge heating.

IV. CONCLUSIONS

The combustion, performance and emission characteristics of 4-stroke, direct injection diesel engine developing power output of 4.4 kW at constant speed 1500 rpm, modified into PPCI combined combustion mode engine. Effect of various premixed fraction were investigated for all load separately. The following are the conclusion.

1. At full load operation, low NO_X, smoke opacity, CO and HC emission achieved up to the premixed fraction 0.40. Adverse effect of this is uncontrolled combustion after increase in premixed fraction 0.20.

2. At 3.3 kW load operation, engine gives better combustion, performance and emission characteristics, low NO_X, smoke opacity, CO and HC achieved up to 0.50 premixed fraction. Combustion is completely controlled, ignition start after diesel injection, no adverse effect.

3. At 2.2 kW load operation, low NO_X, smoke opacity but, after increase in premixed fraction more than 0.3, increase in the BSEC as well as HC and CO emission also increases. It can be used only up to 0.3 premixed fractions, after increasing ethanol quantity, negative effect on all three combustion, performance and emissions characteristics.

4. At 1.1 kW load, ethanol shown completely worst results, inefficient combustion. Low NO_X, smoke opacity but dramatically increased in HC emission, higher BSEC. Energy contribution of ethanol at higher premixed fraction almost negligible.

5. After using charge heating of intake air at 110 ^oC, main advantage goes to 2.2 kW load. It has shown excellent improvement in the combustion, performance and emission characteristics with low NO_X, smoke opacity, CO and HC achieved up to 0.62 premixed fractions. Start of combustion also controlled by diesel injection. No adverse effect on engine.

6. 1.1 kW load is also benefited by charge heating but up to some limit, less than premixed fraction 0.43. It is showing normal combustion property up this limit.

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