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EXPERIMENTAL ANALYSIS OF ETHANOL FUELED PARTIALLY PREMIXED COMPRESSION IGNITION 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 fractions. 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 NOX emission is found to be extremely low than that of diesel, however, the HC and CO emissions are relatively high.**

Index Terms— **Premixed fraction, homogeneous, HCCI, combustion, NOX, temperature.**

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 (NOX) 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.

Energy scenario of petroleum products in India.

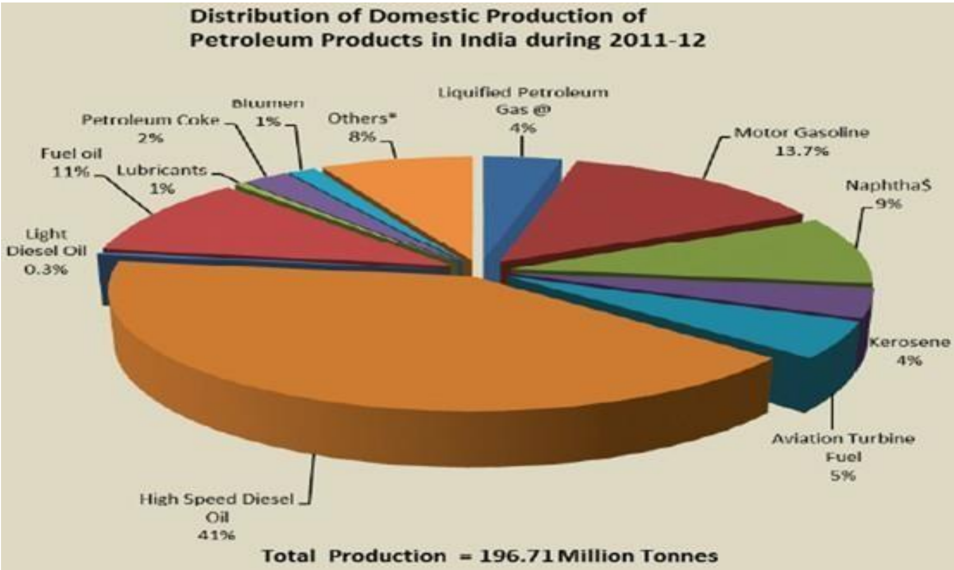

Fig 1.1 Distribution of domestic production of petroleum products in India during 2011-12

Fig 1.1 shows the production of petroleum products in India. According to the Ministry of Statistics and Programme Implementation, Government of India, production of petroleum products report 2013, high speed diesel oil accounted for the maximum share of 41.63%, followed by Motor Gasoline (13.67%), Fuel Oil (9.89%), Naphtha (8.73%), Kerosene (3.8%) and Aviation Turbine Fuel (5.11%). Due to the lower cost of diesel and superior in terms of efficiency, dependency on CI engine increases day by day. In last decade, number of diesel operated vehicles increases as compared to gasoline vehicles.

A. Alternative fuels for CI engines

In CI engines, chemical characteristic of fuel as well as the functional and design conditions of the engine affect engine combustion, performance and emission parameters. Some important fuel characteristics are as follows;

1. Fuel should have a sufficient volatile for good mixing of charges.
2. Low auto-ignition temperature with high cetane rating.
3. CI engine fuel should have a short ignition lag to reduce knocking.
4. Fuel should not produce either smoke or odour after combustion.
5. For smooth flow of fuel viscosity should be sufficiently low.
6. Fuel should be non corrosive and wear resistance.
7. The fuel should have a high flash and a high fire point.

Conventionally, gasoline and diesel are used in SI and CI engine respectively. But, now introduction of alternative fuels that are not derived from fossil fuels seem to be prominent. The alternative fuel may be a renewable or non-conventional fuel derived from various organic substances that can substitute or replace conventional fuels. That is available in a solid, liquid and gaseous form.

B. Homogeneous charge compression ignition (HCCI)

HCCI is a type of combustion in which fuel and air 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. The HCCI engine produces gasoline like soot emission while diesel like power efficiency.

- SI engines: homogeneous charge spark ignition engine (gasoline engine)
- CI engines: stratified charge compression ignition engine (diesel engine)

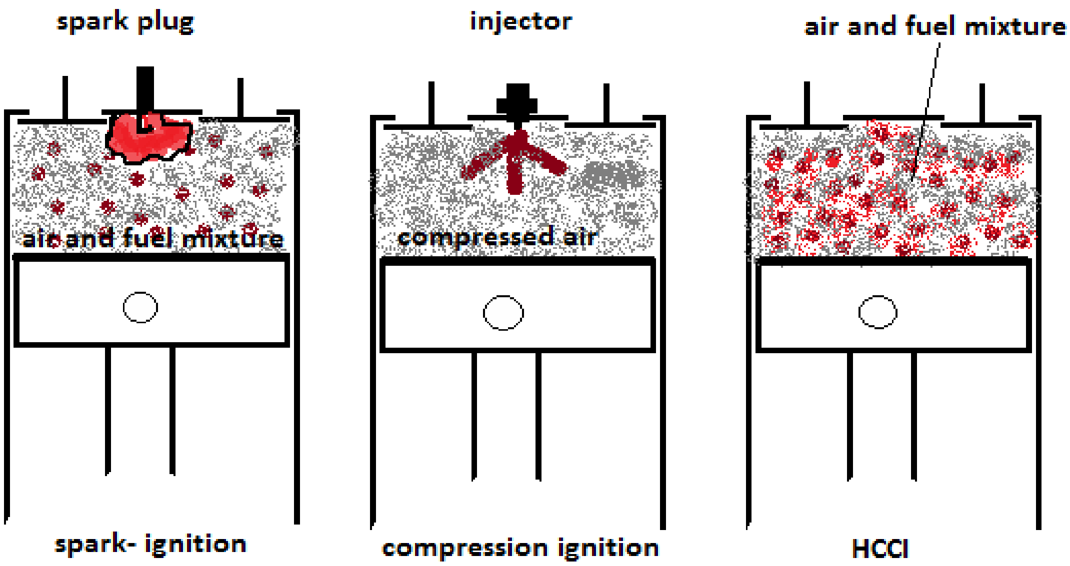

Fig 1.2 Comparison of SI, CI and HCCI combustion

1) Features of HCCI combustion

I. Unlike SI engine there is no throttling loss, lean fuel operation and higher compression ratio. Hence, HCCI gives higher power efficiency and superior in fuel economy.

II. In HCCI combustion, charge is homogeneously mixed, no accumulation of fuel in cylinder, produce less or no soot.

III. In HCCI combustion, start of combustion occurs when charge is auto-ignited. It completely depends on fuel characteristics, engine properties and atmospheric conditions.

IV. Since combustion starts instantaneously, there is no flame propagation shorter combustion duration and avoids knocking problem.

V. In HCCI no more specification of fuel required. HCCI engine is a fuel flexible engine which can be operated by low cetane fuel and several alternative fuels.

VI. Maximum temperature of cylinder reduces, because of overall cylinder combustion reduces simultaneously and hence lower NOxemissions.

VII. By removing a higher pressure injector and other equipment lower cost of engine can be forecast.

VIII. The HCCI engine can save 15- 30% fuel practically, while meeting current emission standard.

2) PPCI concept

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 knocking 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 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. [24, 25, 26]

In this study initially a brief description of commercial diesel engine and technologies related to the engine architecture influencing the combustion process are discussed. The existing conventional diesel engine is converted into PPCI combustion mode. Experiments were conducted in the PPCI mode using ethanol as a premixed fuel.

II. LITERATURE REVIEW

Rakesh Kr Maurya et al. [7] used port fuel injection technique for preparing homogeneous mixture. Twin cylinder engine was converted into HCCI mode in which one cylinder worked on homogeneous charge compression ignition and the left as conventional compression ignition diesel engine. Experiments were performed by altering the intake charge temperature and equivalence ratio at constant speed 1500 rpm in order to achieve the stable HCCI combustion. It was found that stable homogeneous charge combustion was achieved within the range of air-fuel ratio (2.0-5.0). For ethanol, the highest indicated thermal efficiency was found to be 44.78 % and maximum IMEP obtained was 4.3 bar at 2.5 air fuel ratio and 120 oC intake air temperature. The combustion characteristics, combustion efficiency and emissions were also discussed.

Haoyue Zhu et al. [8] conducted an experimental study to the blending of ethanol in biodiesel. Addition of ethanol reduced viscosity, surface tension, where as enhances interaction and wave growth at liquid gas interface. Ethanol and improved spray atomization in order to obtain more homogenous mixture. Engine tests were performed on a one cylinder, based on multi CIDI engine with some modification. They worked for reduction of soot and NOx for diesel, biodiesel, and biodiesel–ethanol. In a moderate exhaust gas recirculation (EGR), premixed low temperature combustion (LTC) mode was investigated. The research focused on blended ethanol, enhance fuel and air mixing rate, prolongs ignition delay, increased fuel oxygen from 10.2% to 15.1% as a result of reduction in soot.

Vittorio Manente et al. [9] conducted an experimental study to perform a sweep, in the start of injection of the pilot and pilot-main interaction ratio at high load. A start of injection SOI sweep was carried out in order to understand the most convenient stratification level that maximized the efficiency and minimized the emissions. The experiment was based on a single cylinder DI engine with modification, engine boosted by using compressed air on external air line. Fuel was injected by using Bosch injection system. The fuel used was ethanol (99.5% by volume, heating value 29 MJ/kg). They perform low load analysis and high load analysis at different operating parameter. Results showed that low NOx , soot, CO and HC can be achieved when EGR rate varied between 40-47% and air fuel ratio between 1.15 and 1.25. Pilot injections were placed at -60o TDC while pilot main ratio was found to be 50-50. The use of some oxygenate were able to reduce soot production.

J. Hunter Mack et al. [10] investigated the effect of water fraction in ethanol on the HCCI engine operating limits, intake temperature, heat release rate and exhaust emissions. The experiments were conducted on Volkswagen 1.9 L 4 cylinder engine. The liquid fuels were port injected by MSD injector and controlled by MSD software. In all the experiments ethanol fuel flow rates were held constant with varying fraction 100%, 90%, 80%, 60%, and 40% of ethanol in water mixtures, and it was concluded that stable HCCI operation was obtained for fuels containing up to 40% water. Results indicated that by increasing intake heating value, HCCI engine can be operated with high fraction of water in ethanol.

III. EXPERIMENTATION

In partial premixed compression ignition engine ethanol is introduced in intake manifold, and some quantity of diesel is injected by conventional direct injection. By regulating the quantity of premixed ethanol and direct injected diesel, various premixed fractions obtained. The present study, investigates the consequence of premixed fractions on the combustion and emission parameters of diesel engine.

A. Fuel selection for the study

Ethanol is used as a premixed fuel because of its high latent heat of vaporization allowing a denser fuel–air charge, and excellent lean-burn properties. When ethanol is burnt, it forms more moles of exhaust gases, which gives higher pressure and more power in the expansion stroke relatively low boiling point and excellent ignition ability. The diesel chosen for direct injection, as a complementation under the condition of misfires or knocks to expand the engine operating range. The engine experiments were conducted over a load range with various premixed fraction, while the engine speed was fixed at 1500 rpm.

B. Fuel properties

The comparison of properties of ethanol used as premixed fuel and diesel used as a directly injected fuel in this study is given in Table3.1.

PROPERTY	ETHANOL	DIESEL
Chemical formula	C_2H_5OH	$C_{12}H_{26}$
Molecular weight (kg/kmol)	46	170
Oxygen percent (weight %)	34.8	Nil
Specific gravity	0.785	0.84-0.88
Boiling temperature at 1 atm (C)	78.3	190-280
Latent heat of vaporization (kJ/kg)	840	270
Stoichiometric air/fuel ratio	9.0	14.5
Lower heating value of fuel (MJ/kg)	26.9	42.5
Research octane number (RON	107	-
Motor octane number (MON)	89	-
Density at 20 C (kg/m3)	790.7	820-900
Auto ignition temp	423	210
Cetane no	5-12	46-51

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 its main technical specifications are summarized in Table 3.2 and the schematic layout of the experimental arrangement is depicts in Fig. 3.2.

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_2, CO, CO_2, and HC. AVL 437 smoke meter is used to measure exhaust smoke.

Table 3.2 Technical data of single cylinder engine

Maker	Kirloskar
Cooling system	Air
Displacement (cm^3)	662
Stroke (mm)	110
Bore (mm)	87.5
Compression ratio	17.5:1
Speed (rpm)	1500
Injection timing	23 degree bTDC
Injection pressure (bar)	200
Rated output	4.4 KW
Injection type	Pump-line-nozzle injection System
Nozzle type	Multi hole

D. 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.

Premixed fraction

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

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

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)

E. Engine modification

For achieving the PPCI 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.

i. **Port fuel injection system**

Fig. 3.1 shows the photograph of the fuel premixing system used in this study. This 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 pressurised 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 3.1 Fuel premixing system Fuel pump and fuel tank [1], Fuel injector [2], Injectorcontrol 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.

In the present study, arduino software is in used to feed the program in microcontroller. Arduino is an open source software written in Java, uses to compile programs and to upload programs to the microcontroller.

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.

ii. Air pre-heating system

Fig. 3.2 Pictorial view of experimental set up

IV. RESULT & DISCUSSION

Module I: Preliminary investigation on the combustion, performance and emission parameters for a PPCI 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.

1) Pressure crank angle diagram at full load operation

Fig 4.1 depicts the variation of cylinder pressure with crank angle at full load for different premixed fractions. The start of injection for diesel was set 23oCA bTDC, while ethanol was inducted with the air. The maximum cylinder pressure at 4.4 kW load for diesel operation is found to be 70 bar.

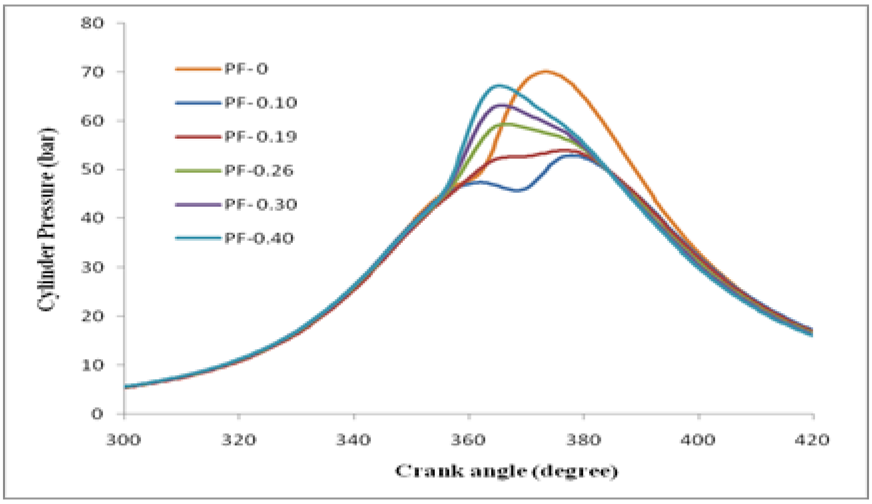

Fig 4.1 Variation of cylinder pressure with respect to crank angle

By increasing the ethanol premixed fraction leads to a decrease in cylinder pressure near TDC. This is due to the vaporisation cooling of ethanol [26]. 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. For the premixed fractions of 0.10, 0.19, 0.26, 0.30, 0.40 the cylinder pressures are 52.87, 53.93, 59.36, 63.51 and 67.18 bar respectively. Further increase in the premixed fraction leads to misfire at 4.4 kW load.

2) Heat release rate with crank angle diagram at full load operation

The variation of heat release rate with crank angle for different premixed fractions at full load is depicted in Fig. 4.2 The heat release from the combustion follows first law of thermodynamics for a closed system using the equation [1]

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} \qquad \text{................} [1]$$

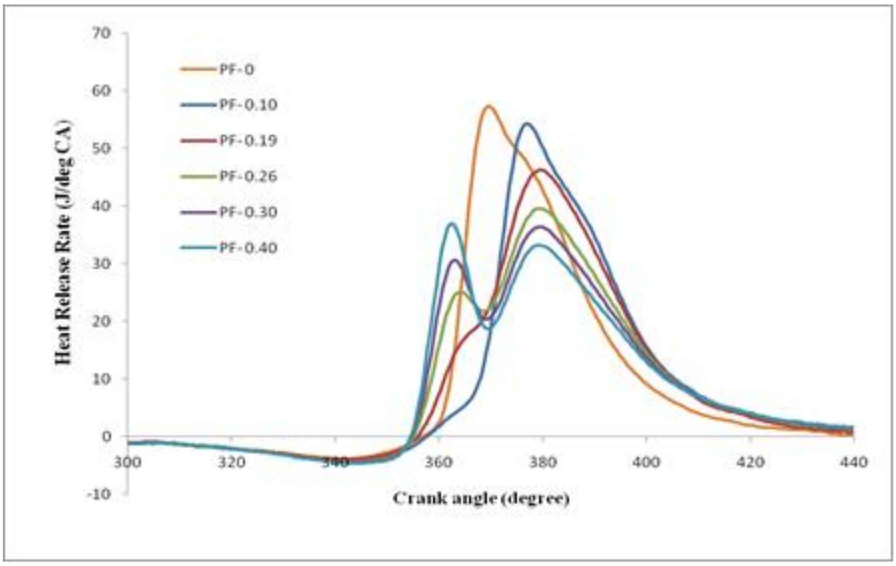

Fig. 4.2 Variation of heat release rate with crank angle at full load

The heat release rate shows the intensity of rapid combustion. Maximum heat release rate for diesel operation at full load is 57.3 J/oCA. 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. At premixed fraction 0.40, the maximum HRR is 36.89 J/deg in the first phase, while in second phase occur is 33.23 J/deg.

B. Performance characteristics

1) Brake specific energy consumption

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 4.3. 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.

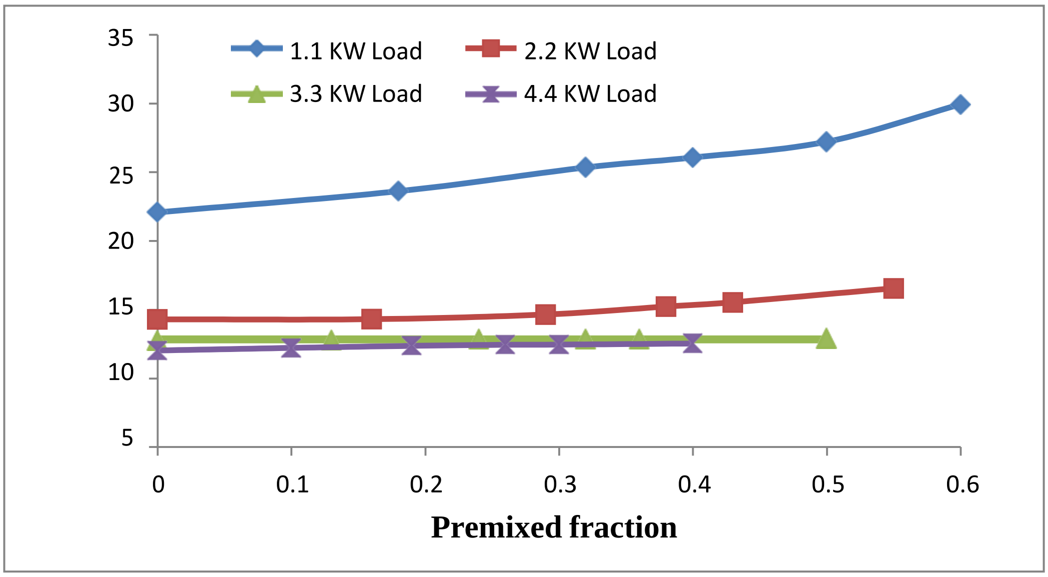

Fig 4.3 Variation of BSFC with premixed fraction

2) Exhaust gas temperature

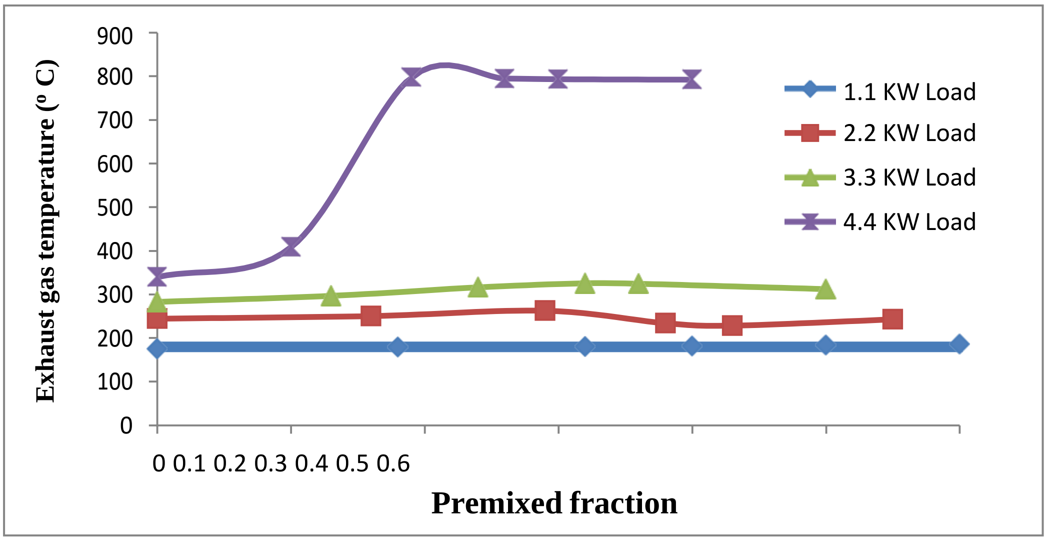

Fig 4.4 Variation of exhaust gas temperature with premixed fraction

The exhaust gas temperature mainly depends upon in cylinder temperature and expansion process. Fig 4.4. shows the variation of exhaust gas temperature with respect to the premixed fraction. The exhaust gas temperature increases with increase load. Figure shows high increment 340 o C to 792 o C in exhaust gas temperature at full load. It may be to two stage combustion or heat release in expansion process, but it proves high temperature inside the cylinder because of this premixed ethanol auto ignited. For low loads, only marginal increment in the exhaust gas temperature occurs. For medium load, maximum 14 % increment in exhaust gas temperature occurs.

C. Emission characteristics

1) Carbon monoxide (CO) emission

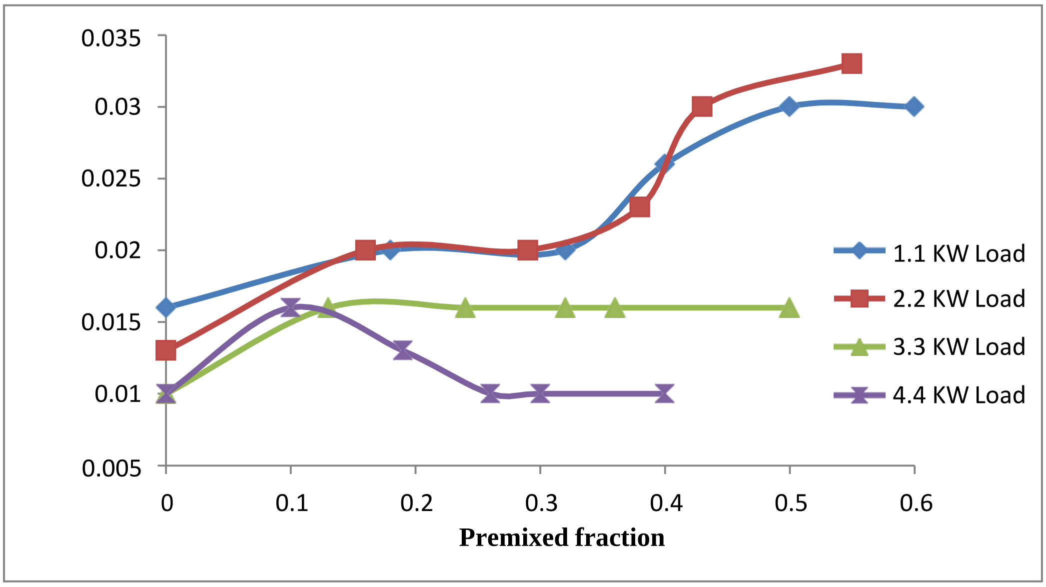

Fig 4.5.Variation of carbon monoxide with premixed fraction

Fig 4.5 shows the variation carbon mono oxide with premixed fraction for all loads. For diesel operation, the CO emission decreases 0.016% to 0.01% with increasing load 1.1 kW to 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, figure 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.

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.

1) Pressure crank angle diagram

Figure 4.6 shows the variation of cylinder pressure with respect to crank angle at 2.2 kW load with charge heating. In the first module, results showed that pressure drop near TDC with increasing premixed fraction due to vaporisation cooling of ethanol. After charge heating, pressure drop near the TDC reduces due to less affect of the vaporization cooling on combustion.

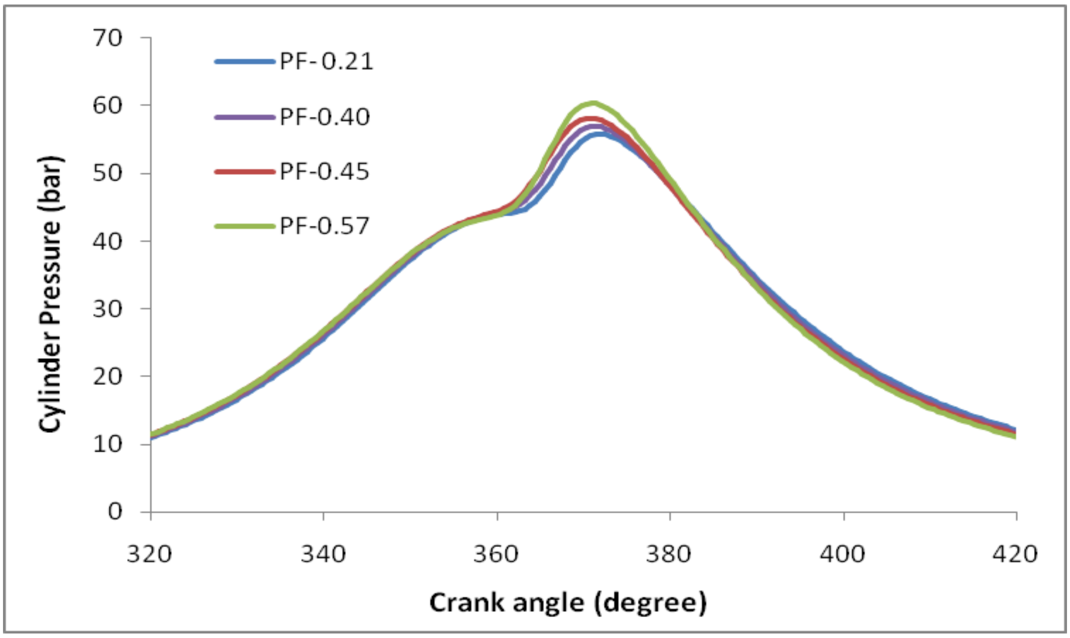

Fig 4.6 Variation of cylinder pressure with crank angle at 2.2 kW load

Peak pressure increases with increasing premixed fraction up to 0.57. The maximum cylinder pressure is 60.3 bar at premixed fraction 0.57. Moreover, maximum pressure at premixed fraction 0.31, 0.40, 0.45and 0.57 occurs at 371.9°, 371°, 370.8° and 370.9° CA respectively. Increasing the premixed fraction, peak pressure increases and shifts towards the TDC.

E. Heat release rate with crank angle diagram at low load operation

Fig 4.7 depicts the variation of heat release rate with respect to crank angle. High heat release rate shows intensity of rapid combustion, by increasing the premixed fraction leads to rapid combustion and shorter combustion duration. In comparison with the first module, heat release rate enhanced by using charge heating with respective premixed fraction at 2.2 kW loads. The maximum heat release rate increases up to 47.4 J/ ^{o}CA at premixed fraction 0.57.

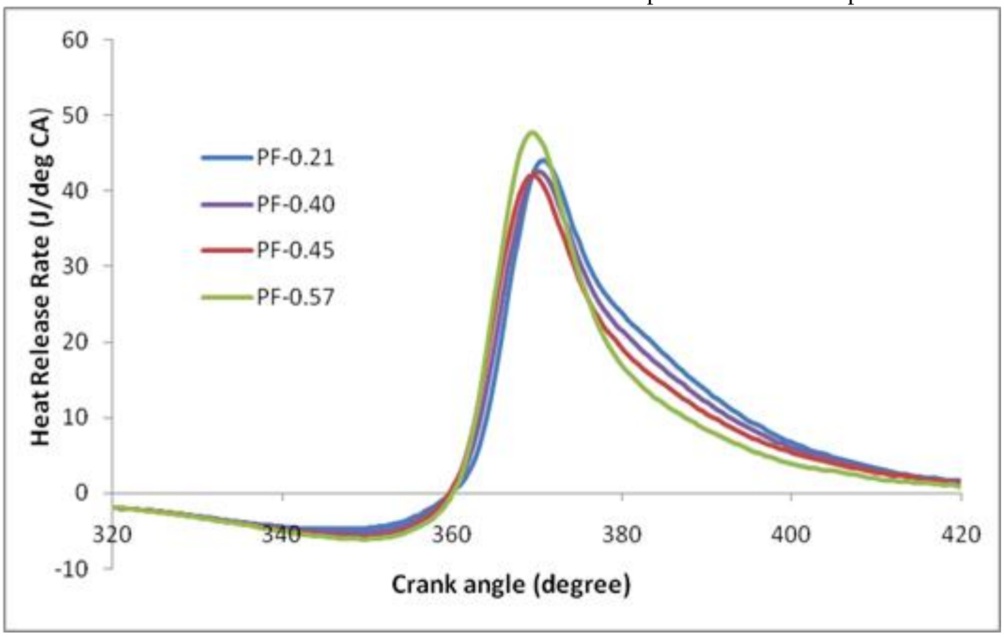

Fig. 4.7 Variation of heat release rate with crank angle at 2.2 kW load

F. Emission characteristics

1) Carbon monoxide

Inefficient combustion leads to the higher carbon monoxide percentage in exhaust gas. High CO emission in the exhaust is the clear indication of incomplete combustion of premixed mixture [6]. Fig. 4.8 depicts the variation of carbon monoxide with respects to premixed fraction. At 1.1 kW load, increment in carbon monoxide at higher premixed fraction indicate incomplete combustion. The CO emission increases almost double with increasing premixed fraction 0.45 to 0.62. For 2.2 kW and for 3.3 kW loads, the CO emission reduces with the increased premixed fraction.

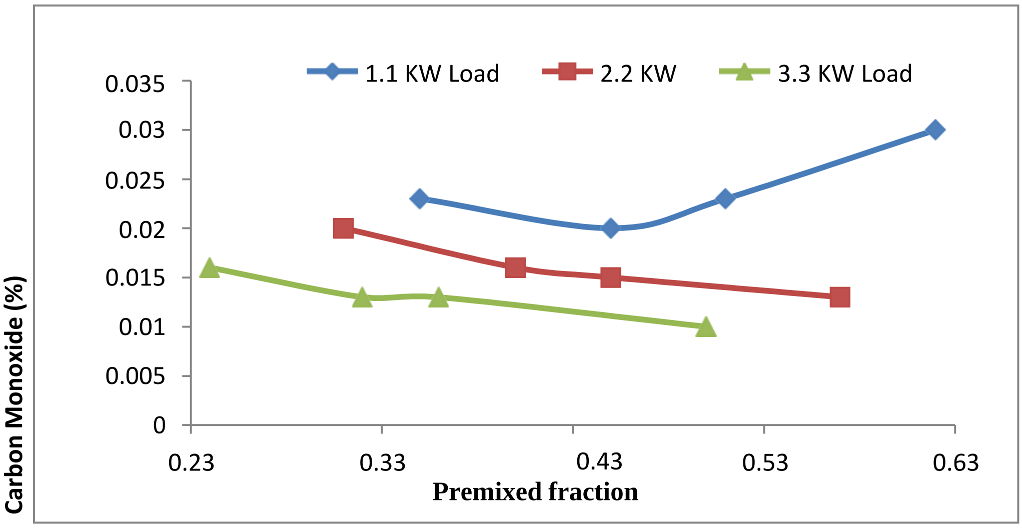

Fig. 4.8 Variation of carbon monoxide with premixed fraction

2) Carbon dioxide

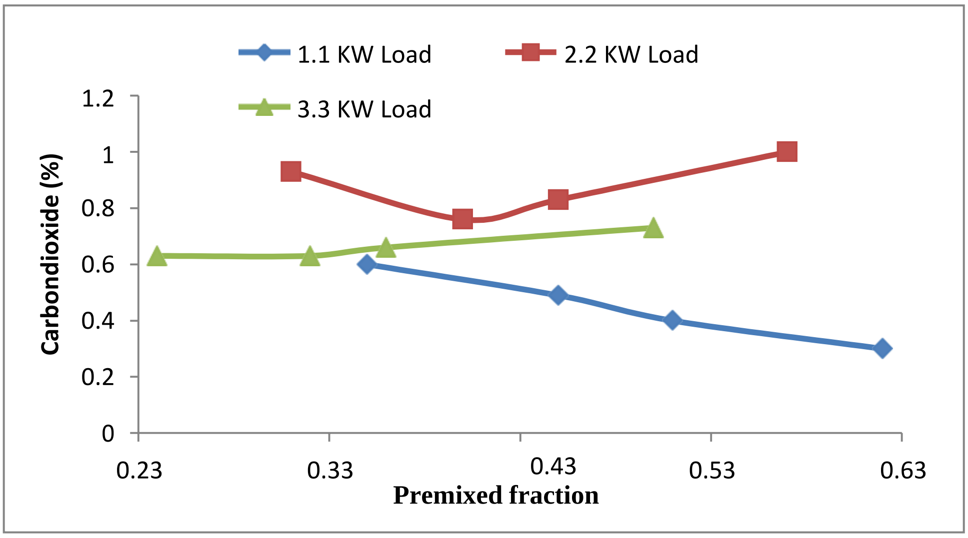

Fig. 4.9 Variation of carbon dioxide with premixed fraction

V. CONCLUSIONS & FUTURE SCOPE

A. CONCLUSIONS

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

The following are the conclusion from the first module;

1. At full load operation, low NOX, 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. Hence, ethanol can be used up to premixed fraction 0.20 without any engine modification.
2. At 3.3 kW load operation, engine gives better combustion, performance and emission characteristics, low NOX, 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 NOX, 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 NOX, smoke opacity but dramatically increased in HC emission, higher BSEC. Energy contribution of ethanol at higher premixed fraction almost negligible.

The conclusions from the second module are as follows,

1. 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 NOX, 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.

2. 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 to this limit.

In India, resources of ethanol are sufficient. It is obtained naturally and manufactured industrially. So, it can utilise in place of diesel to improve efficiency and economic condition both.

B. Future scope

Low temperature combustion technologies shows promising future that is the only reason for leading automobile companies are searching opportunities in HCCI, PPCI, and other LTC combustion technology. As per the experiment, used ethanol with diesel has been proved the potential and fuel flexibility of PPCI combined combustion engine. In order to improving operating limit of this combustion system at higher load following methods can be used:

- Exhaust gas recirculation
- Cool air induction at intake manifold
- Advance injection of diesel

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