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EXPERIMENTAL INVESTIGATION ON HCCI (HOMOGENOUS CHARGE COMPRESSION IGNITION) COMBUSTION ENGINE FUELED WITH GASOLINE AND DEE BLEND

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H I G H L I G H T S

- Gasoline and DEE fuel blend are tested in the HCCI mode.
- Gasoline and DEE blend fuel has no cold start problem in HCCI mode.
- As the load increases, the BTE and EGT increases.
- The HCCI combustion process has obvious two stage combustion characteristics.

ABSTRACT

This paper presents the findings of an experimental investigation on homogeneous charge compression ignition (HCCI) engine fuelled with gasoline and Di-ethyl-ether blend fuel. The experiments were conducted on a single-cylinder, four stroke, DI diesel engine modified to work in HCCI mode using 90% of gasoline in gasoline/DEE blend fuel (by volume) at a fixed engine speed of 1400 rpm. Factors that were investigated and compared with base diesel engine include brake thermal efficiency, combustion pressure, HRR, NOx, HC, CO, CO2, smoke and exhaust gas temperature. The combustion process has obvious two stage combustion characteristics. The results show that within the investigated blends the HCCI operation is possible in the load range from 3.25 bar to 5.27 bar.

Keywords: HCCI engine, Gasoline, Di-ethyl-ether, Renewable fuel, Blend fuels,

I INTRODUCTION

Ever increasing vehicle exhaust emissions and fast decreasing fossil fuel reserves are main challenges for engine researchers and manufacturers. With increasing concern on the effect of exhaust emission on environment, animal, plant and human life automobile vehicle have been subjected to increasingly stringent regulations.

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Spark ignition engines, with precise control of air/fuel ratio and three-way catalysts, are clean power sources. The thermal efficiency of SI engines is lower than that of CI engines due to low compression ratio, high pumping loss, difficult to achieve lean-burn and large cycle variation.

Compression Ignition (CI) engines have much higher thermal efficiency which is favorable for energy saving and CO_2 reduction but emits high NOx, smoke and particulate matter.

So there is demand for new combustion concept which can reduce emission and have high thermal efficiency. The new combustion concept homogeneous charge compression ignition (HCCI) combustion is a promising way to accomplish these tasks [1–9]. In HCCI engines, lean homogeneous air and fuel mixture is taken to the cylinder as in spark ignition engines and ignited spontaneously by compression as in compression ignition engines.

As seen above HCCI is hybrid of SI and CI engines, it has advantages of both engines. The use of lean air and fuel mixture and no flame front formation, combustion temperature is low and so ultra low NOx formation. Compression of air and fuel mixture leads to simultaneous auto-ignition in multiple locations throughout the cylinder. This results in combustion of whole cylinder charge simultaneously and at faster rate than the conventional SI or CI modes nearly at constant volume so high thermal efficiency.

Onishi and Noguchi were the first to have done systematic study on auto-ignition in two stroke engines in 1979. They have achieved reduction in fuel consumption and exhaust emission and were able to reduce the engine instability at part load [1, 2]. Najt and Foster [1983] showed that HCCI combustion could be achieved in a spark ignition (SI), four-stroke gasoline engine under lean fuelling and elevated inlet charge temperatures. They used a mixture of isooctane and n-heptane which are primary reference fuels. The effect of increasing the inlet charge temperature is to advance the auto ignition timing and decrease the combustion duration [3]. Thring tested the four-stroke diesel fueled HCCI mode of operation and was the first to name this type of combustion as homogeneous charge compression ignition [4].

The road of HCCI implementations are blocked by challenges which needs to overcome. These are the control of ignition timing and combustion rate, higher CO and HC emissions particularly at light loads, difficulty with cold start and transient response of the HCCI engines. Auto ignition occurs simultaneously and spontaneously across the combustion chamber. This spontaneous and sudden combustion at higher load causes a rapid heat release rate resulting in knocking. Sustaining combustion at light load without misfire is difficult.

There are many controlling strategies found in literature to control HCCI combustion. The effect of compression ratio on the ignition timing has also been discussed [6, 7]. Christensen et al. [6] varied the compression ratio from 10:1 to 28:1 with a constant air/fuel equivalence ratio of 3.0 using a port injection system. They concluded that gasoline required a compression ratio of 22.5:1 for satisfactory operation without the use of inlet air preheating. The results showed that no smoke was generated and that NOx emissions were very low with increased compression ratio. Since the combustion efficiency decreased with increased compression ratio, the indicated efficiency did not improve with increased compression ratio.

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Both Goldsborough et al. [8] and Marriott [5] conducted a study on the effect of fuel injection timing using a direct injection (DI) gasoline HCCI engine. Gold et al. tested the engine at 1500 rpm with a constant 22:1 air/fuel ratio by varying the SOI timings. Their results showed that as the time between injection and ignition increases, the variability in combustion stability improves. Marriott [5] studied the effect of SOI on HCCI combustion for different engine speeds and equivalence ratios, and concluded that SOI would be an effective and useful needed parameter to control ignition timing for transient engine applications.

S Swaminathan et al [9] investigated on acetylene fuelled HCCI engine with variations of charge temperature and EGR. Initially, the intake air was heated to different temperatures in order to determine the optimum level at every output. Charge temperatures needed were in the range of 40^0c to 110^0c from no load to a BMEP of 4 bar. Subsequently EGR was done at identified charge temperatures and brake thermal efficiency was found to improve. At high BMEP's use of EGR led to knocking. Thus close control over the charge temperature and EGR needed.

It is found that at BMEP of 1.5 and 2 bar, 46% and 16% EGR are required respectively to run the engine without external charge heating. It was possible to operate the engine without the electrical heater at low BMEP's (< 2.5 bar) through the use of hot EGR. It is expected that the engine can be operated without the heater even at BMEP's above 2.5 bar if the temperature of recirculated exhaust gas could be cooled and its flow rate could be finely controlled. NOx and smoke levels were 20 ppm and 0.1 BSU respectively. However, HC levels were very high at about 1700-2700 ppm. Brake thermal efficiencies were comparable or even better than CI mode operation.

Kitae Yeom et al [13] investigated the effects of IVO timing and fuel quantity on exhaust emissions and combustion characteristics in LPG and gasoline fueled HCCI engines controlled by a VVT. Intake valve open timing (CAD) -29,-19, -9, 1, 11 were used. The peak combustion pressure was decreased and the start of the combustion crank angle was retarded as the IVO timing was retarded due to reduced volumetric efficiency and an increase in the residual gas. At earlier IVO timing, IMEP was suppressed by negative work. However, at the latest IVO timing, IMEP was decreased due to incomplete combustion. The optimal IVO timing for the maximum IMEP was retarded as λ_{TOTAL} was decreased. This is because the negative work during compression increased as λ_{TOTAL} was decreased due to early combustion. The CO2 emission was reduced as the IVO timing was retarded. The HC emission was increased due to late combustion at the retarded IVO timing and increased λ_{TOTA}L. The CO emission from the LPG HCCI engine was increased as the IVO timing was retarded due to late combustion occurring after top dead center (ATDC).The CO emission from the LPG HCCI engine increased slightly compared to the gasoline HCCI engine. Lowered combustion temperature and pressure due to late combustion results in weaker oxidation of CO during the expansion stroke.

Magnus Christensen et al [10] experimented on iso-octane, ethanol and natural gas fuelled HCCI engine with supercharging. It was not possible to achieve high IMEP values with HCCI, the limit being 5 bar. Supercharging is one way to dramatically increases IMEP. Two compression ratios, 17:1 and 19:1 were used. The inlet conditions were set to 0, 1 and 2 bar of boost pressure. The highest attainable IMEP was 14 bar using natural gas as fuel at the

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boost pressure of 2 bar and the compression ratio of 17:1. The HC emission decreased with increasing boost pressure and engine load. The CO emission is very dependent on air-fuel ratio and preheating. Close to rich limit and with hot inlet air, very little CO generated. Extremely little NOx was generated in all cases. The values are as low as after a three-way catalyst or better.

Naoya et al [11] experimented on light fuelled single cylinder PCCI engine with in cylinder water injection. The control of ignition timing and suppression of rapid combustion in a PCCI engine was attempted with direct in cylinder injection of water as a reaction suppressor. The water injection significantly reduced the heat release at low temperature oxidation, which suppressed the increase in charge temperature after the low temperature oxidation. The possible engine operating range with ultra low NOx and smokeless combustion was extended to higher load range with the water injection. Rapid combustion was suppressed by reduction in the maximum in-cylinder gas temperature due to water injection while the combustion efficiency suffered. Therefore, the maximum charge temperature needs to be controlled within an extremely limited range to maintain a satisfactory compromise between mild combustion and high combustion efficiency. While the combustion suppression effect increased with increase in the quantity of water injection resulted in increase in THC.

W L Hardy et al [16] used several multiple injection strategies in their experimentation on PCCI combustion. They found that increasing g intake boost pressure simultaneously decreases NOx and PM emission by diluting the mixture. This leads to decreased cylinder temperature for NOx reducing and by decreasing the equivalence ratio, which exposes more PM to oxygen and improve oxidation. Use of close coupled post injection increases in cylinder mixing and decreases PM emissions or allows more EGR to be used for decreased NOx emissions. Splitting the pilot injection into two premixed injection shows promise for reducing fuel spray wall impingement, which lowers PM emissions due to decreased diffusion burn from the resulting wall fuel films. Multiple injection strategies showed potential for satisfying 2007 emission mandate while still preserving engine efficiency, as demonstrated by the good BSFC results of the present studies.

Mohamed H. Morsey [18] has used additives for ignition control of methane fuelled HCCI engine. The control of ignition timing could be achieved either by addition of promoters (additives reducing ignition delay time) or blending low cetane number fuel (such as natural gas and methanol) with high cetane number fuels. He studied the effect of additives of formaldehyde (CH_2O), hydrogen peroxide (H_2O_2) and DME on ignition timing numerically. It was found that an additive free mixture did not ignite for an intake temperature of 400K. A mixture containing small quantity of additives at the same temperature was ignited. For a fixed quantity of additives, it was found that H_2O_2 addition was effective in advancing the ignition timing compared to the other two additives. It was observed that the percentage of additives required achieving near TDC ignition timing increases linearly with the increase of engine speed, whiling decreases with the increase in equivalence ratios. A near TDC ignition timing could be achieved with only a small amount of H_2O_2, while a quantities of 6 to 12 times of that H_2O_2 additives were required for CH_2O and

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DME respectively, for all engine speed examined. A volume fraction of at least10% of DME was required to be added as compared to 5% of CH_2O and 0.8% of H_2O_2. NOx emission was also small.

Diesel has high cetane number due to which it ignites to early and knocking combustion occurs when used in HCCI engine. Diesel has low volatility so there is difficulty in its vaporization and homogeneous air and diesel mixture formation unless there is air heating or diesel vaporizer is provided.

Gasoline has a very low cetane number and will not ignite normally in CI engine within the time available in an engine cylinder, hence some means of ignition must therefore be provided. Kitae Yeon [13] have used VVT to retain IEGR and heat LPG to burn in HCCI engine. Mingfa et al [17] have utilized heated intake air for burning Methanol in the cylinder. K Sudheesh et al [14] used DEE as ignition improver for burning Biogas in HCCI engine. M. Mohamed Ibrahim [15] used diesel as ignition improver in hydrogen fueled HCCI engine. Zhili Chen [12] has used DME as ignition improver in LPG fueled HCCI engine.

In literature, use of DEE as ignition improver for gasoline in HCCI mode has not found. This motivated the authors to undertake the investigation on gasoline and DEE blend fuelled HCCI mode of operation.

In the present work, gasoline and DEE blend is used to run the HCCI engine. External mixture formation techniques are used. Experimental investigation is carried on single cylinder DI diesel engine modified to work in HCCI mode. The blend of gasoline and DEE= 90/10 percentage by volume are tested. The brake thermal efficiency, combustion pressure, HRR, NOx, HC, CO, CO2, Smoke and EGT are recorded at constant speed of 1400 rpm and various engine loads. The results are also compared with the base diesel engine.

II EXPERIMENTAL SET UP

Experiments were conducted on a single-cylinder, four stroke, water-cooled, naturally aspirated, direct-injection diesel engine. The engine was coupled to an eddy current dynamometer for loading and measurement purposes. The engine was modified to operate in HCCI mode by supplying fuel in the intake manifold.

The specifications of the engine are shown in Table 1.

Model	G510 W III
Make	Greaves Cotton Ltd. Aurangabad (M.S.), India.
Type of Engine	Single cylinder, four stroke, direct injection
Bore -MM	85
Stroke- MM	90
Displacement-CM3 (Liters)	510
Compression ratio	17.5:1
Rated RPM	3000
HP	10

Table 1.The engine specifications

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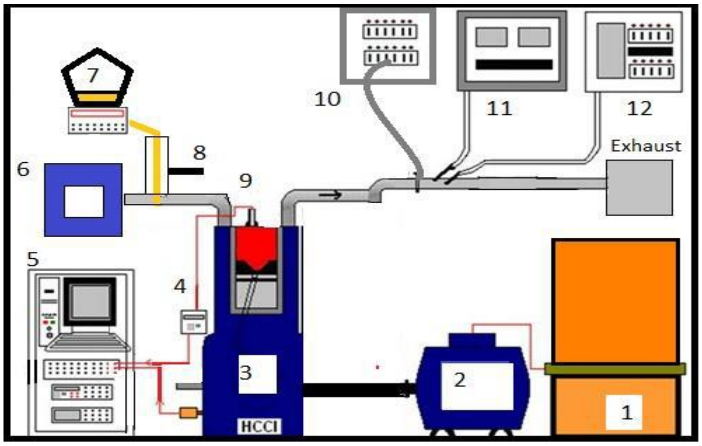

Figure 1 Detailed Schematic Diagram of Experimental Setup.

1. Dynamometer Control	7. Fuel tank with control valve resting on weight balance
2. Dynamometer	8. Fuel Control Valve
3. HCCI Engine	9. TDC Sensor
4. TDC Encoder	10. Exhaust Gas Analyzer
5. High Speed Data Acquisition System	11. NOx Meter
6. Air box connected to intake manifold	12. Smoke Meter

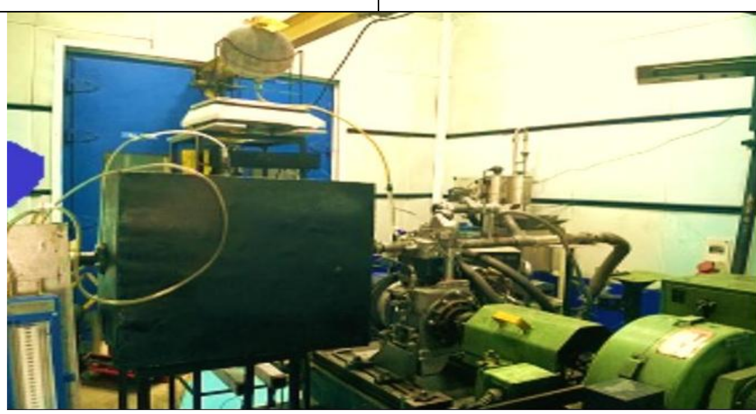

Figure 2 Experimental Setup Photograph

Gasoline-Di-Ethyl-Ether (DEE) blend fuel used. Gasoline was used as a primary fuel. Gasoline has low cetane number and will not ignite under the given operating condition with compression ignition, so Di-Ethyl-Ether (DEE) was used as ignition enhancers. Blend fuel was supplied continually to the intake air with a 3mm copper tube inserted in the intake manifold at 150 mm in the upper reaches of the engine under room temperature and pressure. The blend fuel was supplied against the direction of the air motion, to facilitate mixture formation and it was assumed that the homogenous air and fuel mixture would be formed during the process of intake and compression. During the compression stroke, due to low self-ignition temperature the DEE fuel auto ignites first and acts as an ignition source for entire cylinder of the homogeneous gasoline–air mixture. The gasoline - DEE blend flow rate

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were varied manually according to the load. An electronic weighing scale (0.1g resolution) was employed to weigh the fuel consumption during engine operation.

A 3 kW electric heater was installed in the inlet duct to preheat the intake air. The heater was located upstream of both the fuel entry and the exhaust gas recirculation loop. The inlet temperature was measured in the inlet ports and was used to control the heating power. Thermocouple (K-type) was used to measure exhaust gas temperature.

The cooling water outlet temperature was measured by a resistance temperature detector (RTD).

The specifications of Gasoline, DEE and Diesel were shown in Table 2

S N	Properties	Gasoline	Diethyl ether	Diesel
1	Formula	-	C2H5OC2H5	C8 to C20
2	Density (kg/m3)	750	713	833
3	Viscosity at 20 1C (centipoise0	-	0.23	2.6
4	Boiling point (^{0}C)	-	34.4	163
5	Cetane number/ RON	RON- 92-98	CN-125	CN-40-55
6	Auto-ignition temperature (^{0}C)	280	160	257
7	Stoichiometric air fuel mass ratio	15.17	11.1	14.5
8	Flammability limits, rich (vol%)	-	9.5–36	5
9	Flammability limits, lean (vol%)	-	1.9	1
10	Calorific value (kJ/kg)	44000	33900	42500

Table 2. The specification of gasoline, DEE and diesel.

The cylinder pressure was recorded by PTX 1400 pressure transducer that has a sensitivity of 15pC⁄bar was mounted at the cylinder head. The charge output from this transducer was converted to an amplified voltage using a GM 12 D amplifier and supplied to AVL INDI MICRO data acquisition system. The cylinder pressure signals were recorded on a personal computer using an analogue to digital converter and average pressure was obtained from 100 consecutive cycles. A angle encoder **AVL 365 C 01** was used to provide crank angle degree resolved measurements. The cylinder pressure data were analyzed with a zero-dimensional combustion model, where both air fuel mixture and temperature were assumed to be homogeneous in the whole cylinder volume. Zero-dimensional combustion models were good at predicting heat release process and combustion phasing of homogenous charge compression ignition (HCCI) combustion or premixed compression ignition (PCI) combustion modes, so the combustion progress through the engine cycle was tracked. However, the zero-dimensional combustion model assumes ideally uniform in-cylinder charge and did not consider the processes of fuel spray, droplet vaporization and charge mixing, so some parameters influenced by charge in homogeneity, such as incomplete combustion products, were hardly accurately predicted. The heat release rate was calculated from pressure data with this model.

The NOx was measured with Horiba MEXA-720 NOx analyzer. The HC, CO and CO2 were measured by Horiba MEXA 5543A Automotive Emission Analyzer. The smoke was measured with AVL 437 smoke meter.

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For the introduction of Gasoline/DEE blend fuel, a 3mm copper tube was inserted in the intake manifold. The Gasoline/DEE flow rates were varied manually according to the load. An electronic weighing scale (accuracy 0.1g) was used to measure the mass flow rate of Gasoline/DEE blend fuel.

2.1 Operating conditions:

To insure the repeatability and comparability of the measurements for different fuels and operating condition, the intake charge temperature was fixed at 40^0C, held accurately to within $\pm 1^0$ C. The coolant outlet temperature and lubricant oil temperature remains at 80^0C$\pm 2^0$ C. The engine speed was maintained at 1400 rpm.

2.2 Experimental procedure

First, gasoline-DEE blend fuel of the required percentage of gasoline by volume is filled in the fuel tank. Then, engine is motored at about a speed of 500 rpm, the electric heater is made on and simultaneously gasoline-DEE blend fuel is supplied to the intake manifold. Thus, the engine is started in HCCI mode. The engine speed is allowed to reach about 1400 rev/min, by adjusting flow rate of blend fuel. At this condition, the engine is allowed to run until the coolant temperature reached about 80^0C. Meanwhile, the temperature controller which is set at 40^0C will control the temperature of intake air to 40^0C $\pm$ 1^0C. Then the dynamometer is set to constant speed mode and speed is set to 1400 rpm. After this, increase in blend fuel flow rate will increase the load on engine and speed will be maintained at 1400 rpm. Then fuel flow rates are adjusted manually to obtain the different loads. At each load all reading are recorded. The same procedure is repeated for all gasoline-DEE blend fuels.

III. RESULTS AND DISCUSSIONS

3.1 Brake thermal efficiency.

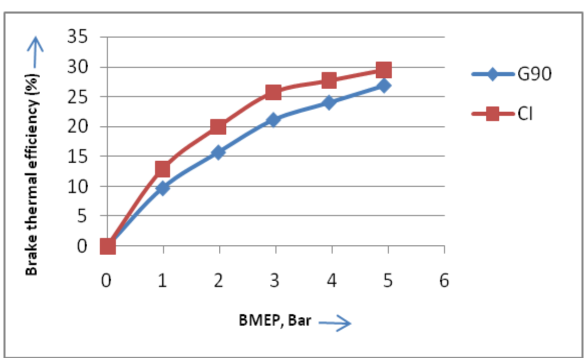

Figure 3 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on brake thermal efficiency.

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Fig. 3 shows the effect of variation of load on brake thermal efficiency of the HCCI engine with gasoline proportion of 90 % by volume in the blend fuel of DEE/gasoline. The brake thermal efficiency increases with load in HCCI as well as in diesel mode. The brake thermal efficiencies in HCCI mode are 9.73%, 15.72%, 21.14%, 24.04% and 26.89% as compared to 12.88%, 19.97%, 25.73%, 27.707% and 29.49% in CI mode at BMEP 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The brake thermal efficiencies in the HCCI mode are 24.45%, 21.28%, 17.83%, 13.23% and 8.81% less as compared to CI mode at BMEP 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The brake thermal efficiencies in the CI mode are more than HCCI mode at all BMEP's. This may be due to two reasons; one late burning of gasoline (high octane number fuel) and second continuous supply of blend fuel to the engine.

3.2 Carbon Dioxide

Fig. 4 shows the effect of variation of load on CO2 emission of the HCCI engine with gasoline proportion of 90 % by volume in the blend fuel of DEE/gasoline. From the figure 6 it can be observe that as the load increases the CO2 increases. The increase in CO2 emission as the load increases is due to burning more fuel. The CO2 emissions are 1.39%, 2.18%, 3.11%, 3.8%, 4.66% and 5.24% volume in HCCI mode as compared to 2.1%, 3.2%, 4.4%, 5.6%, 6.4% and 7.2% volume in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. More CO2 emitted in CI mode than HCCI mode.

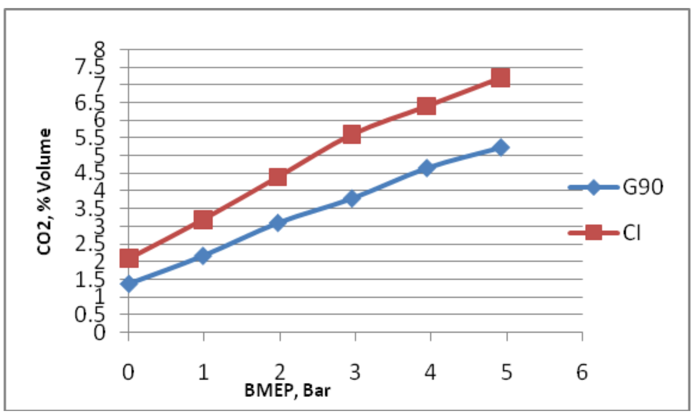

Figure 4 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on CO2, % volume.

3.3 Hydrocarbon, ppm.

The effect of variation of load on the hydrocarbon emission is shown in fig. 5. In HCCI mode the HC emissions are 1303ppm, 1101ppm, 941ppm, 768ppm, 540ppm and 430ppm at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. In HCCI mode, the hydrocarbon emission decreases with increase in the load. At higher

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load more fuel will be burned in the cylinder, so more combustion temperature and complete combustion and hence less HC.

The HC emissions are 20ppm, 29ppm, 38ppm, 54ppm, 82ppm and 107ppm in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The hydrocarbons are formed due to quenching effect at the walls and crevice volumes. At walls of engine cylinder and crevice volumes the temperature are less due to more heat transfer. The available temperature is not sufficient to burn the fuel in these areas. Fuel presents in these areas escape the cylinder without burn or partially burned, hence more HC emission. In HCCI mode, due to lean air and fuel mixture the combustion temperature is less. Due to this complete fuel in the cylinder not burn. This un burn fuel appears as HC in exhaust. So the HC emission in HCCI mode is much higher compared to CI mode at all loads.

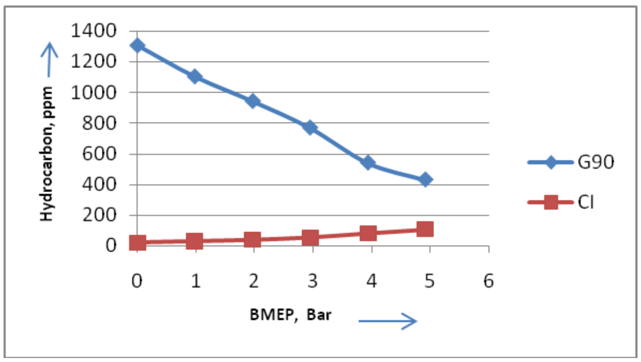

Figure 5 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on hydrocarbon, ppm.

3.4 Carbon monoxide, % volume.

Lower combustion temperature causes in complete combustion of fuel. Some carbon will get converted to CO instead of CO2 due to incomplete combustion of fuel. In HCCI mode due to lean air and fuel mixture the combustion temperature remain less, which causes more CO formation. As the combustion temperature increases CO formation decreases.

From figure 6, the CO emission are 0.55%, 0.52%, 0.49%, 0.45%, 0.42% and 0.39% volume in HCCI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The CO emission decreases with increase in load. This is due to higher cylinder temperature at higher load which helps in complete combustion of fuel and hence less CO.

The CO emission are 0.08%, 0.1%, 0.12%, 0.15%, 0.18% and 0.21% volume in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. Compared to CI mode, CO emission is much higher at all loads in HCCI mode.

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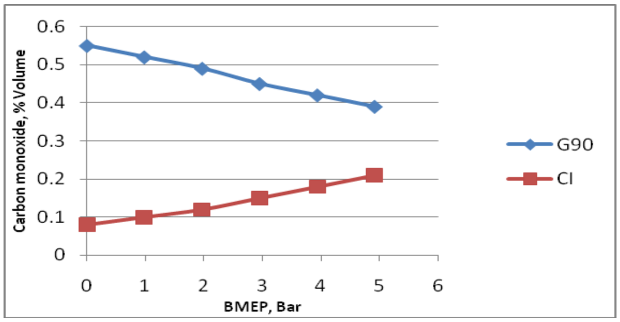

Figure 6 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on carbon monoxide, % volume.

3.5 Nitric oxide, ppm.

Figure 7 shows the variation NOx with load. The Nox emission are 4ppm, 9ppm, 27ppm, 48ppm, 71ppm and 124ppm in HCCI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The NOx emission increases as the load increases in HCCI mode.

In CI mode the Nox emission are 131ppm, 215ppm, 336ppm, 517ppm, 721ppm and 982ppm at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The Nox emission are 96.94%, 95.81%, 91.96%, 90.71%, 90.15% and 87.37% less in HCCI mode as compared to CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The NOx emission is very less in HCCI mode as compared to CI mode. The formation of NOx depends on the combustion temperature. In HCCI mode, due lean air and fuel mixture less heat is generated in engine cylinder and so less temperature. Due to less combustion temperature NOx formation is less.

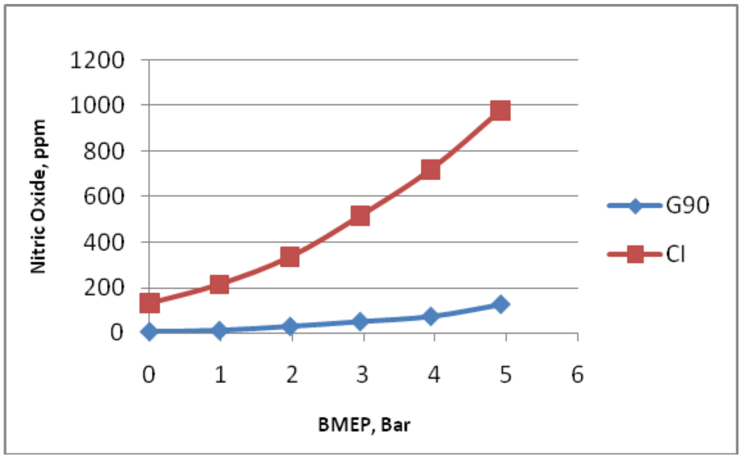

Figure 7 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on Nox, ppm.

3.6 Smoke, HSU.

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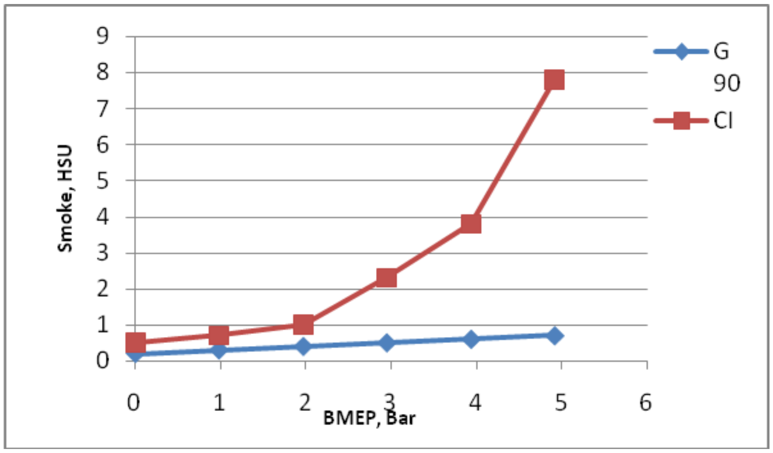

Figure 8 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on smoke, HSU.

Fig.8 shows the variation of smoke with load in HCCI mode as well as in CI mode. Smoke emission in HCCI mode is ultra small. In HCCI mode smoke emission are 0.2 HSU, 0.3 HSU, 0.4 HSU, 0.5 HSU, 0.6 HSU and 0.7 HSU at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. Homogeneous air and fuel mixture is supplied to engine in HCCI mode. So there are no presence of packets of rich fuel and air mixture in the cylinder. The rich fuel and air mixture forms smoke during combustion. As there are no rich fuel and air mixture present in the fuel, the smoke formation is extremely low in HCCI mode.

In CI mode, diesel is injected at end of compression stroke in the compressed air the cylinder, the air and fuel mixture will be heterogeneous. The rich packets of air and fuel in this non homogeneous mixture forms more smoke in CI mode. Smoke emission in CI mode are 0.5 HSU, 0.7 HSU, 1 HSU, 2.3 HSU, 3.8 HSU and 7.8 HSU at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The smoke emission are 60%, 57.14%, 60%, 73.91%, 84.21% and 91.01% less in HCCI mode as compared to CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The smoke emission in CI mode is much higher at load compared to HCCI mode.

3.7 Exhaust gas temperature, degree Celsius.

Figure 9 shows the variation of exhaust gas temperature with load. The EGT in HCCI mode are 145^0C, 151^0C, 165^0C, 190^0C, 230^0C and 301^0C in HCCI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The exhaust temperature increases with load.

The EGT in CI mode are 146^0C, 164^0C, 196^0C 247^0C, 332^0C and 444^0C at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

The EGT are 0.68%, 7.92%, 15.8%, 23.07%, 30.72% and 32.20% less in HCCI mode as compared to CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively.

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The EGT in HCCI mode are lower as compared to CI mode. In HCCI mode due to lean air and fuel mixture combustion temperature is less and so the EGT.

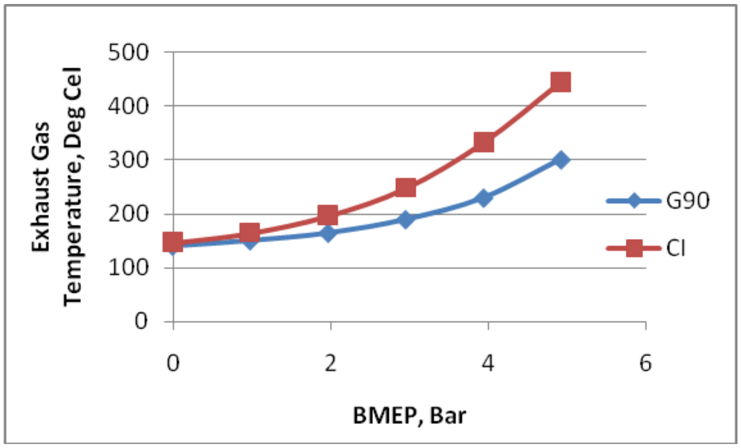

Figure 9 Effect of variation of load at proportion of DEE/Gasoline = 10/90 on exhaust gas temperature, degree Celsius.

IV. CONCLUSIONS

1. The HCCI can run on gasoline/ DEE blend fuel satisfactorily from no load to 5.27 bar BMEP. 1. The brake thermal efficiencies in HCCI mode are 9.73%, 15.72%, 21.14%, 24.04% and 26.89% as compared to 12.88%, 19.97%, 25.73%, 27.707% and 29.49% in CI mode at BMEP 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The brake thermal efficiency increases as the load increases in HCCI mode as well as in CI mode.

2. The brake thermal efficiencies in the HCCI mode are lower than CI mode at all BMEP's. This may be due to two reasons; one late burning of gasoline (high octane number fuel) and second continuous supply of blend fuel to the engine in HCCI mode.

3. The CO2 emissions are 1.39%, 2.18%, 3.11%, 3.8%, 4.66% and 5.24% volume in HCCI mode as compared to 2.1%, 3.2%, 4.4%, 5.6%, 6.4% and 7.2% volume in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The increase in CO2 emission as the load increases is due to burning more fuel. More CO2 emitted in CI mode than HCCI mode.

4. In HCCI mode the HC emissions are 1303ppm, 1101ppm, 941ppm, 768ppm, 540ppm and 430ppm at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. In HCCI mode, the hydrocarbon emission decreases with increase in the load. The HC emissions are 20ppm, 29ppm, 38ppm, 54ppm, 82ppm and 107ppm in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The HC emission in HCCI mode is much higher compared to CI mode at all loads due to low combustion temperature.

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5. The CO emission are 0.55%, 0.52%, 0.49%, 0.45%, 0.42% and 0.39% volume in HCCI mode as compared to 0.08%, 0.1%, 0.12%, 0.15%, 0.18% and 0.21% volume in CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The CO emission decreases with increase in load in HCCI mode. This is due to higher cylinder temperature at higher load which helps in complete combustion of fuel and hence less CO. Compared to CI mode, CO emission is much higher at all loads in HCCI mode.

6. The Nox emission are 4ppm, 9ppm, 27ppm, 48ppm, 71ppm and 124ppm in HCCI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The NOx emission increases as the load increases in HCCI mode. In CI mode the Nox emission are 131ppm, 215ppm, 336ppm, 517ppm, 721ppm and 982ppm at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The Nox emission are 96.94%, 95.81%, 91.96%, 90.71%, 90.15% and 87.37% less in HCCI mode as compared to CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The NOx emission is very less in HCCI mode as compared to CI mode.

7. The smoke emission are 60%, 57.14%, 60%, 73.91%, 84.21% and 91.01% less in HCCI mode as compared to CI mode at BMEP 0 bar, 0.98 bar, 1.97 bar, 2.95 bar, 3.93 bar and 4.92 bar respectively. The smoke emission in HCCI mode is lower at loads in HCCI mode as compared to CI mode.

12. The conventional tradeoff between the NOx and soot are defeated in HCCI mode, both reduced simultaneously.

13. The engine starts from cold easily.

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