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The Effect of Air Preheating on the performance and emission characteristics of a DI Diesel Engine achieving HCCI mode of combustion

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Abstract

The investigation focuses on the effect of diesel vapour induction on the engine performance and to try and achieve Homogeneous Charge Compression Ignition (HCCI) mode of combustion in the engine. An existing Direct injection CI engine is modified to work as an HCCI engine by using a shell and tube heat exchanger which aids in the production of diesel vapour by utilising energy of exhaust gas. Different readings are taken for 100% load conditions. The effect of preheating of air is analysed by heating the inlet air by using a coil type heater. Preheating improved the brake thermal efficiency and brought down CO and HC emissions however it slightly increased NOx emissions. Induction of vapour was continued till the point where engine started to knock. Preheating of vapour was limited to 65 °C due to continuous increase in NOx emissions. The best operating condition for each load was calculated and finally the optimum condition for the operation of HCCI engine was determined.

Keywords: Performance, Emissions, HCCI, Injection Timing, Diesel Vapour Induction, air preheating.

I. INTRODUCTION

The internal combustion engine is one of the key drivers in modern industrial society. There are two types of internal combustion engines: spark ignition (SI) and compression ignition (CI). The conventional SI combustion is characterised by a flame propagation process. The onset of combustion in SI engines can be controlled by varying ignition timing from the spark discharge. Because the mixture is premixed and typically stoichiometric the emissions of soot are orders of magnitude lower than that in the diesel processes. The major disadvantage of SI engine is its low efficiency at partial loads.

Conventional diesel combustion, as a typical representation of CI combustion, operates at higher compression ratios (12–24) than SI engines. In this type of engine, the air–fuel mixture auto-ignites as a consequence of piston compression instead of ignition by a spark plug. A part of the air and fuel will be premixed and burn fast, but for the larger fraction of the fuel, the time scale of evaporation, diffusion, etc. is larger than the chemical time scale. Therefore, the mixture can be divided into high fuel concentration regions and high temperature flame regions. In the high fuel concentration regions, a large amount of soot is formed because of the absence of Oxygen. Some soot can be oxidized with the increase of in-cylinder temperature. The in-cylinder temperature in a conventional diesel engine is about 2700 K, which leads to a large amount of NO_x emissions.

Consequently, the obvious ideal combination would be to find an engine type with high efficiency of diesel engines and very low emissions of gasoline engines with catalytic converters. One such candidate is the process known as homogeneous charge compression ignition, HCCI[2,3]. One of the main drawbacks of the HCCI engines is the lack of an ignition source and its poor performances in low and high load conditions. This lead to the idea of combining the HCCI mode of combustion and CI mode of combustion [4]. Here part of the fuel is fed as homogenous mixture using a heat exchanger and the remaining fuel is fed directly into the combustion chamber which also starts the combustion.

The preheating of air is done to improve the operating range of the engine in HCCI mode of combustion. The high temp of air reduces the possibility of condensation of diesel vapour in intake manifold there by mass fraction of vapor inducted can be increased.

II. **EXPERIMENTAL METHOD**

The existing four stroke single cylinder diesel engine of Kirloskar make is slightly modified with certain additional accessories to run as a HCCI engine. The components are carefully selected and instrumentation system was developed. The experiments are conducted on a computerized single cylinder four stroke naturally aspirated direct injection water cooled diesel engine test rig. The engine is directly coupled to an eddy current dynamometer. The engine and the dynamometer were interfaced to a control panel which is connected to a computer. The software Enginesoft 2.4 is used to record the engine performance and combustion characteristics. The parameters which are measured and used in the present study are

Brake thermal efficiency, brake specific fuel consumption etc. The engine specifications are given in Table 1.

Table 1. Specifications of the Diesel engine test rig

Engine	4 stroke single cylinder CI engine
Make	Kirloskar
Power	5.2 KW @ 1500 RPM
Bore X Stroke	87.5 X 110 mm
Compression ratio	17.5:1
Connecting rod length	234mm
Dynamometer type	Eddy current with load cell
Load measurement	Strain Gauge load cell
Water flow meter	Rotameter
Fuel and air flow measurement	Differential pressure unit
Speed measurement	Rotary encoder
Interfacing	ADC card PCI 1050

AVL Exhaust Gas Analyser is used for the measurement of exhaust gases. CO, HC, CO_2, O_2 and NO_x emissions were measured for different operating conditions. AVL smoke meter is used for measuring smoke opacity.

A heat exchanger is fixed in the exhaust gas flow line of the engine. The hot exhaust gas will enter into the inlet of the exchanger shell and it vapourises the fuel flowing through the exchanger tubes. The flow of fuel is controlled by a butterfly valve. A coil type heater is mounted in the air flow circuit and a temperature sensor is placed to control the temperature of heated air. Schematic diagram of the setup is given in Fig 1.

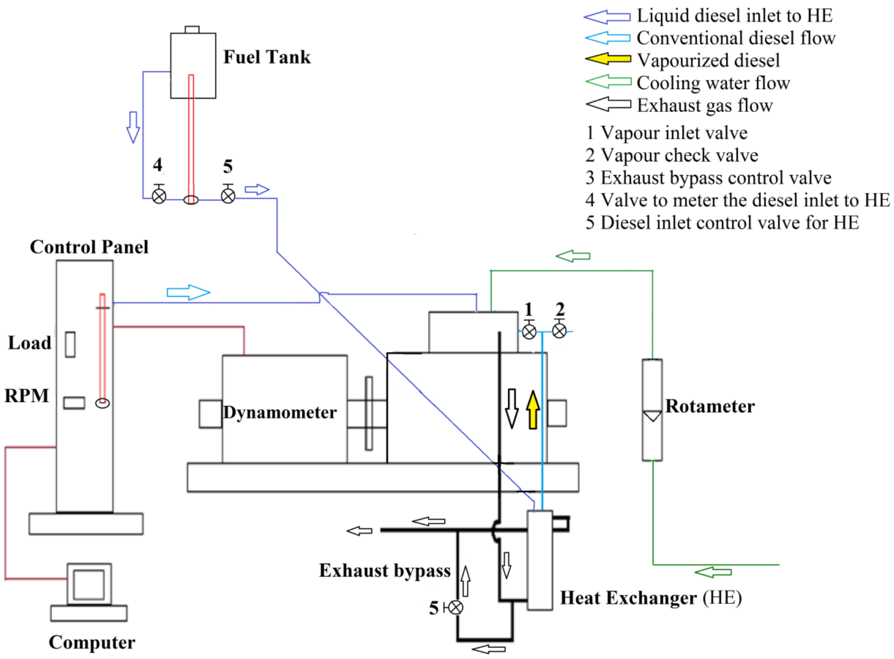

Figure 1. Schematic Diagram of the Experimental Setup.

III. SCHEME OF EXPERIMENTATION

The whole set of experiments are conducted for a constant speed of 1500 rpm and compression ratio of 17.5. The diesel vapour production is limited to 50%, 75% and full load conditions as the exhaust gases are at high temperatures during the above conditions. The mass of fuel vaporised can be measured by using a burette fitted to a separate fuel tank. The flow of liquid diesel to the heat exchanger is controlled by a one way valve fitted to the burette. The term,

$$\%\ \text{vapour induction} = \frac{\text{Mass of vapour fuel inducted}}{\text{Mass of vapour fuel inducted} + \text{Mass of fuel injected}}$$

The air was preheated using an external heating coil and the temperature was varied from 40 to 65 °C . The preheating temperature was limited to 65 °C as temperatures above it caused the engine to knock as well as an increase in the NOx emissions.

IV. RESULTS AND DISCUSSIONS

A. Performance Characteristics

1) Brake Thermal Efficiency

Preheating can be considered as an effective tool in utilizing more diesel vapor with simultaneous improvement in brake thermal efficiency.

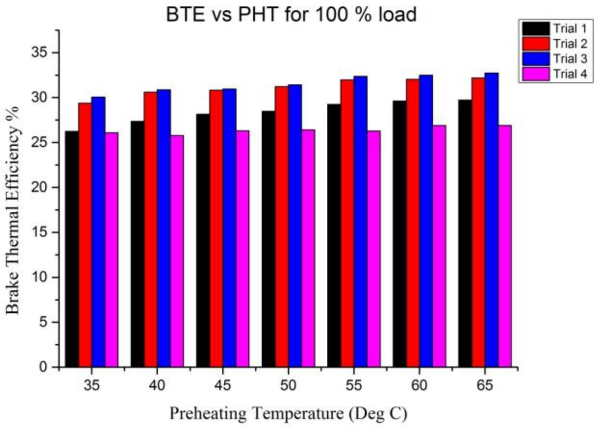

Fig 2. Brake thermal Efficiency V/s Preheating temperature for different vapor induction at 100 % load

The above graph shows the variation in Brake thermal efficiency with an increase in the preheating temperature for different amount of vapor inducted. The amount of vapor inducted was increased through 4 trials so as to know the point where the engine knocks and the trial 4 indicates the knock limited amount of vapor induction. The initil value 35° C indicates the condition without preheating and as the air preheating temperature increases there is an increase in the brake thermal efficiency.

The reason behind this is that the inlet air temperature is an important parameter for creating the homogeneous mixture. An increasing inlet air temperature is used to improve fuel vaporization and create more homogeneity air/fuel charge .A higher temperature of inlet air also reduces the chance of condensation of diesel vapor as both air and vapor mixes near the inlet manifold. Thus higher inlet air temperature promote more complete combustion.

For 100 % load the maximum efficiency obtained is 32.73 % at a preheating temperature of 65 °C where the percentage vapor utilization was found to be 41.6 %. The percentage increase in efficiency for this condition is 40.4% when compared to the conventional engine and 8.91 % when compared to that of HCCI mode without preheating.

Table 2 shows the change in maximum brake thermal efficiency along with the corresponding percentage vapor utilization when preheating was employed.

Table 2. Maximum efficiency at full load condition

Without preheating		With preheating	
Maximum efficiency	Percentage vapour utilisation	Maximum efficiency with preheating	Percentage vapour utilisation
30.05%	38.74 %	32.73 % at 65 °C	41.60 %

2) Brake Specific Fuel Consumption

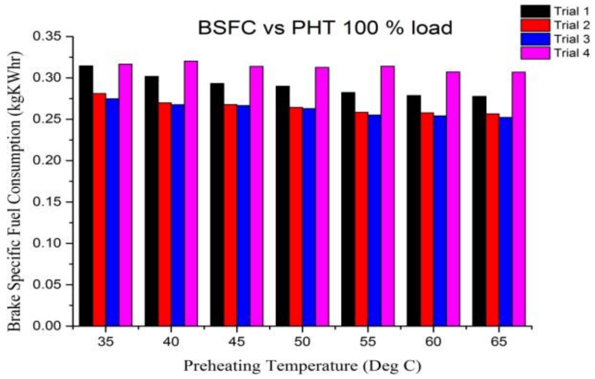

Fig 3. Brake Specific Fuel Consumption V/s Preheating temperature for different vapor induction at 100 % load

In the above figure each trial indicates an increase in the percentage of vapor inducted with trial 4 being the maximum vapor inducted. Each trial indicates the variation in the amount of diesel vapor inducted with trial 4 being the maximum amount. The reason for decrease in BSFC with more amount of vapor is the approaching of HCCI mode for the engine as more vapor is inducted. This makes the mixture more homogeneous and brings down the fuel requirement as explained before.

For 100 % load the minimum BSFC is attained at a preheating of 65 °C and at a percentage vapor of 41.6 % where the efficiency is maximum and the value being 0.2522 kg/KWhr.

B. Emission Characteristics

Automobile emissions are dealt with stringent rules nowadays. The newer emission norms demand very less amount of emissions. There are several researches being carried on to develop technologies that would reduce harmful emissions or at least minimize the need of costlier metals used in after-treatment devices. Here the trends of all the emission parameters at different preheating temperatures at full load condition for diesel vapor induction are being discussed.

1) Unburnt Hydrocarbons

Unburnt hydrocarbons are result of incomplete combustion (which can be caused due to lack of air). As the CI engines work in lean mixture they emit comparatively low amount of UBHC (less than 100 ppm) when compared to SI engines. As HCCI engines run of leaner charge it is expected that the HC emissions will reduce. But this is not the case. A slight increase in HC emissions were observed though the increase was only marginal.

As the engine approach HCCI mode of operation there is a decrease in in-cylinder temperature. This is one of the reasons for the increase in HC emissions. Another reason can be as the homogenous charge is compressed there is a chance that the fuel gets into the crevices and minute cracks in the engine cylinder. These fuel molecules will be kept away from air required for their proper combustion.

The variation in HC emissions for different air temperatures as the engine approaches HCCI mode of operation is shown in figure 4.

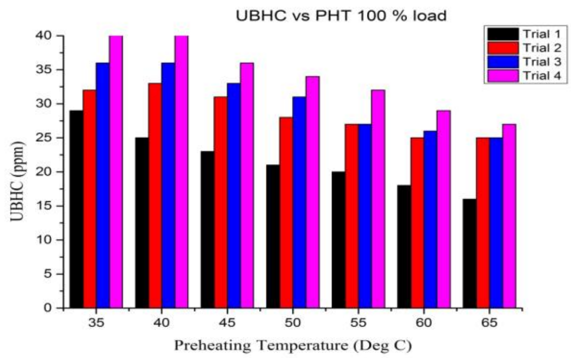

Fig 4. Unburned Hydrocarbons vs Preheating Temperature for different vapor induction for 100 % load

Trial 1 to 4 in fig 4 indicates an increase in percentage vapour. As seen from the Fig 4 for 100 % load the minimum value of unburned hydrocarbons is at 65 °C preheating temperature and at a percentage vapour of 26.45 %

As the preheating temperature increases there is better homogeneity and higher in cylinder temperature. This improves combustion efficiency and results in achieving almost complete combustion. As a result the fraction of unburned hydrocarbons decreases. Even the hydrocarbons trapped in piston rings and crevices take part in combustion due to higher combustion temperature. Thus unburned hydrocarbons reduce to satisfactory levels.

2) Carbon Monoxide

Carbon monoxide is the toxic byproduct of all hydrocarbon combustion. This is the result of incomplete combustion as enough oxygen would not be present for the carbon monoxide to be converted into carbon dioxide which is harmless. It is seen that the carbon monoxide emissions increased as the engine approached HCCI mode of combustion. As the engine approached HCCI mode of operation there was a decrease in in-cylinder temperature. This is one of the reasons for the decrease in carbon dioxide emissions and subsequent increase in carbon monoxide emission. Another reason can be as the homogenous charge is compressed there is a chance that the fuel gets into the crevices and minute cracks in the engine cylinder. These fuel molecules will be kept away from air resulting in incomplete combustion. The variations of CO emissions at different preheating temperatures are given Figure. 5

As seen from the above below the amount of CO decreases with an increase in the preheating temperature. As the preheating temperature increases the in cylinder temperature increases causing better and complete combustion. Complete combustion results in more CO_2 emissions subsequently reducing the more harmful CO emissions. For 100 % load the minimum amount of CO is obtained at 65°C preheating temperature and at a percentage vapour of 26.45%

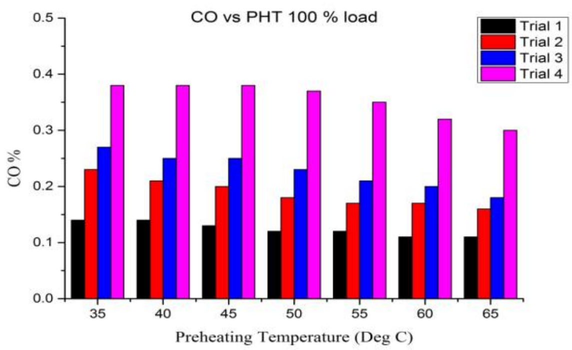

Fig. 5 Carbon monoxide V/s Preheating Temperature for different vapor induction at 100 % load

3) Nitrogen Oxides

Nitrogen oxides are among the major pollutants in engine exhaust. They have far reaching effects and remain in the atmosphere for a long time. One of the major reasons for developing HCCI mode of combustion is reduction of nitrogen oxide emissions. HCCI mode of combustion helps in reducing the peak temperature inside the cylinder and thereby reduces thermal NO_x considerably.One another factor that controls NO_x formation is residence time. As HCCI employs an explosion by production of many auto ignition spots, the residence time will be less and hence lesser NO_x production.

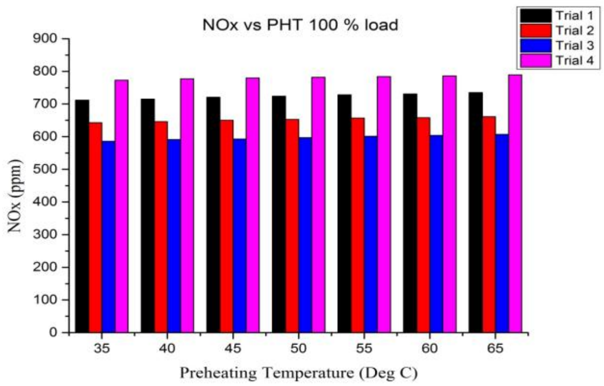

Fig 6. NO_x vs Preheating temperature for 100 % load at different vapor induction

As seen from the fig 6 preheating results in an increase in the NO_x emissions. With increase in the preheating temperature combustion efficiency increases causing a simultaneous rise engine cylinder temperature. This causes an increase in the NO_x emission. Thus minimum NO_x is for HCCI mode without preheating as seen in Fig. 6. Maximum NO_x (789 ppm) was seen for 100 % load at a preheating temperature of 65 °C and at a percentage vapour of 58.064 %.

4) Smoke Opacity

The variation of smoke opacity for at different intake air temperatures and % vapour induction are discussed here. Smoke emissions reduced at higher loads with the introduction of vapour. Also preheating the air reduced the smoke emissions due to better combustion of the charge. For 100 % load the minimum value of smoke is 57 % opacity at 65 °C and a percentage vapour of 41.6 %.

The reason for low smoke emission is due to the absence of rich fuel pocket inside the combustion chamber. HCCI engine uses lean air–fuel charge and combustion takes place at multiple points in the combustion chamber at the same time, which eliminates rich fuel region, due to that, HCCI engine has low smoke emissions than the conventional diesel engine.

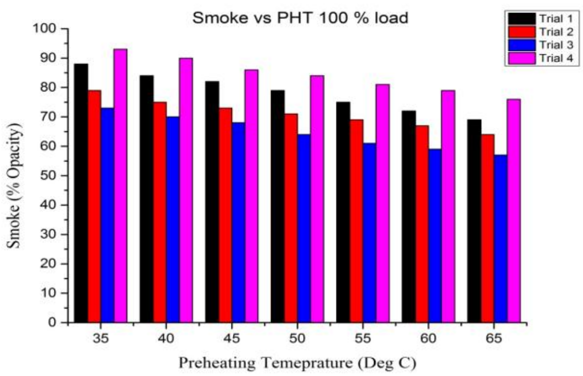

Fig 7. Smoke opacity V/s Preheating temperature for 100 % load for different vapor induction

V. CONCLUSIONS

The investigation is focused on the effect of diesel vapour induction with on the engine performance and to try and achieve HCCI mode of combustion in the engine. It was found that the operation of engine using diesel vapour depends on various parameters. For different conditions the vapour produced from heat exchanger was successfully utilized for combustion.

- ✓ Exhaust gas heat from the engine that was normally wasted was successfully utilized by using a shell and tube heat exchanger.
- ✓ A compact shell and tube type heat exchanger can provide diesel vapours with sufficient degree of superheat required to be inducted in combustion chamber.

- ✓ The total amount of diesel being consumed can be decreased with vapour mode of induction of diesel under proper constraints.
- ✓ For every load there is a limit on the maximum amount of vapour that can be fed without compromising on engine performance
- ✓ Preheating of vapour increased the brake thermal efficiency and for 100 % load the maximum efficiency obtained is 32.73 % at a preheating temperature of 65 °C where the percentage vapor utilization was found to be 41.6 %
- ✓ Preheating also improved the percentage vapour utilisation .A percentage increase of 7.3 % in percentage vapour was found for 100 % load.
- ✓ CO emission was found to increase with increase in vapour fraction but it was successfully reduced by preheating of air. A maximum reduction 57.14 % in CO emissions was attained by preheating of air at 100 % load.
- ✓ Unburned hydrocarbon emissions increases with increase in vapour fraction but it was successfully reduced by preheating of air. A maximum percentage decrease of 44.82 % was attained for 100% load.
- ✓ Nitrogen oxides emission was reduced by inducting more vapour fraction, however preheating resulted in an increase in NOx emissions. Maximum NOx reduction is attained at a percentage vapour of 38.7 % for 100 % load. Maximum NO_x (789 ppm) was seen for 100 % load at a preheating temperature of 65 °C and at a percentage vapour of 58.064 %.
- ✓ Smoke emissions decreased for increase in vapour fraction at full load condition. Also preheating the air reduced the smoke emissions due to better combustion of the charge. For 100 % load the minimum value of smoke is 57 % opacity at 65 °C and a percentage vapour of 41.6 %.
- ✓ The increase in vapor mass fraction improved the performance of the engine. This was mainly because the HCCI mode of combustion was approached. At the same time the start of combustion was still governed by the injection of vapor fuel. This gave a method of control of combustion which is normally absent in HCCI engines.
- ✓ This method can be used in traditional direct injection engines with no modification in the DI system and slight modification in the inlet manifold for inducting diesel vapour

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