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PERFORMANCE AND COMBUSTION CHARACTERISTICS OF A LOW HEAT REJECTION DIESEL ENGINE WITH CARBURETED ETHANOL AND JATROPHA OIL

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ABSTRACT

Investigations are carried out to evaluate the performance of a low heat rejection (LHR) diesel engine consisting of air gap insulated piston with 3-mm air gap, with superni (an alloy of nickel) crown and air gap insulated liner with superni insert with normal temperature condition of jatropha oil and carbureted ethanol with varied injection timing and injection pressure. Performance parameters are determined at various magnitudes of brake mean effective pressure. Pollution levels of smoke and oxides of nitrogen (NOx) are recorded at the peak load operation of the engine. Combustion characteristics of the engine are measured with TDC (top dead centre) encoder, pressure transducer, console and special pressure-crank angle software package. Conventional engine (CE) and LHR engine showed improved performance at recommended injection timing of 27^{o}bTDC and recommend injection pressure of 190 bar, when compared with CE with pure diesel operation. Peak brake thermal efficiency increased by 20%, smoke levels decreased by 45% and NOx levels decreased by 40% with LHR engine at its optimum injection timing with maximum induction of ethanol when compared with pure diesel operation on CE at manufacturer's recommended injection timing of 27^{o}bTDC (before top dead centre).

KEYWORDS: Crude Jatropha Oil, Ethanol, LHR Engine, Performance, Pollution Levels, Combustion Characteristics

INTRODUCTION

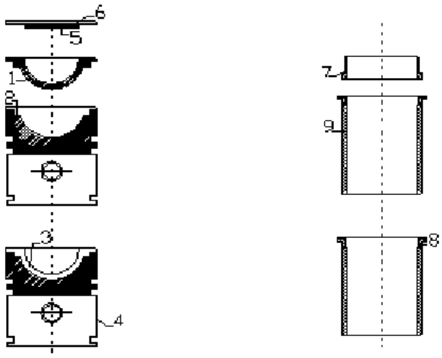
The rapid depletion of petroleum fuels and their ever increasing costs have lead to an intensive search for alternate fuels. The most promising substitutes for petroleum fuels are alcohols. So, it is high time that scientists developed an alternate and renewable fuel that would run on the existing engines without many modifications and also one that would cater to the ever increasing power needs of the countries and domestic market. It is well known fact that small amount of energy is left for useful purpose in engine remaining energy is wasted through friction, heat loss through coolant and heat loss through exhaust gas. Hence the concept of the LHR engine is to minimize the heat loss to the coolant, by providing resistance in the path of heat flow to the coolant thereby gains the heat energy. Several methods adopted for achieving LHR to the coolant are i) using ceramic coatings on piston, liner and cylinder head ii) creating air gap in the piston and other components with low-thermal conductivity materials like superni, cast iron and mild steel etc. Investigations were carried out on bio-diesel by many researchers [1-5] by coating with low thermal conductivity materials like ceramics on engine components like cylinder head, cylinder liner, valves and piston crown, and it was reported that ceramic coated engines improved specific fuel consumption (SFC) and decreased pollution levels. However, low degree of insulation provided by these researchers was not able to burn effectively high viscous crude vegetable oils. Creating an air gap in the piston involved the complications of joining two different metals. Air gap was created [6] in the piston by screwing the crown made of low thermal conductivity material, nimonic (an alloy of nickel) to the body of the piston, by keeping a gasket, made of nimonic, in between these two parts. But investigations are restricted to pure diesel operation. It was reported from these investigations that SFC was improved and pollution levels of smoke decreased at advanced injection timing. Experiments were also conducted on conventional engine (CE) with either with blends of vegetable oil and diesel [7-8] or with blends of bio-diesel and diesel [9-10] and it was reported that these blends improved the efficiency of the engine and decreased the pollution levels. Experiments were also conducted [11] on waste fried vegetale oil collected from restaruents and reported CO and smoke emissions were reduced using preheated waste frying oil at 135°C. Investigaons were conducted [12] with vegetabl oils and reported that all emissions parameters were within maximum limits and concluded safer use as an alternate fuel.on vegetable oils. Compression ratio [13] was also increased with CE with vegetable oil based bio-diesel and it was reported that poor performance was obtained at lower compression ratio and performance of the engine was improved at compression ratio of 18:1. Experiments were conducted [14-15] on vegetable oil based bio-diesel on CE and reported improvement in BTE, exhaust emissions but increased NOx emissions and slight increased brake specific fuel consumption (BSFC). There are many techniques available to induct ethanol into the engine, out of which carburetion technique is simple. Carbureted ethanol was used [16] in CE and in LHR engine with air gap in the insulated piston and insulated liner and vegetable oil was injected in conventional manner and reported that exhaust gas emissions decreased with LHR engine, when compared with pure diesel operation on CE as high heat generated in the combustion space due to adiabatic conditions improved alcohol combustion. However, in their investigations, combustion characteristics and performance parameters were not reported. Vegetable oils have cetane number comparable with diesel fuel, but they have high viscosity and low volatility. Alcohols have low cetane fuels, though they have got high volatility. In order to take advantage from high cetane number and high volatility, both vegetable oils and alcohols have to be used in LHR engine.

The present paper attempts to evaluate the performance of LHR engine, which contains air gap piston and air gap liner with crude jatropha oil with carbureted ethanol with varying engine parameters of change of injection pressure and injection timing and compared with pure diesel operation on CE at recommended injection timing and injection pressure.

METHEDOLOGY

Figure 1 gives the details of insulated piston, insulated liner and ceramic coated cylinder head employed in the experimentation. LHR diesel engine contains a two-part piston; the top crown made of low thermal conductivity material, superni-90 screwed to aluminum body of the piston, providing a 3mm-air gap in between the crown and the body of the piston. The optimum thickness of air gap in the air gap piston is found to be 3-mm [6], for better performance of the engine with superni inserts with diesel as fuel.

A superni-90 insert is screwed to the top portion of the liner in such a manner that an air gap of 3mm is maintained between the insert and the liner body. At 500^{o}C the thermal conductivity of superni-90 and air are 20.92 and 0.057 W/m-K respectively.



1. Crown
2. Gasket
3. Air gap
4. Body

Insulated piston

5 Insert
6. Air gap
7. Liner

Insulated liner

Figure 1: Assembly Details of Insulated Piston, Insulated Liner and Ceramic Coated Cylinder Head

Experimental setup used for the investigations of LHR diesel engine with jatropha oil based bio-diesel is shown in Figure 2. CE has an aluminum alloy piston with a bore of 80 mm and a stroke of 110mm. The rated output of the engine is 3.68 kW at a rate speed of 1500 rpm. The compression ratio is 16:1 and manufacturer's recommended injection timing and injection pressures are 27^{o}bTDC and 190 bar respectively. The fuel injector has 3 holes of size 0.25mm.

The combustion chamber consists of a direct injection type with no special arrangement for swirling motion of air. The engine is connected to electric dynamometer for measuring brake power of the engine. Alcohol is inducted through the variable carburetor jet, located at the inlet manifold of the engine at different percentages of diesel flow rate by mass basis and crude vegetable oil is injected in conventional manner**.** Two separate fuel tanks and burette arrangements are made for measuring vegetable oil and alcohol consumptions**.**

Air-consumption of the engine is measured by air-box method. The naturally aspirated engine is provided with water-cooling system in which inlet temperature of water is maintained at 60^{o}C by adjusting the water flow rate. The engine oil is provided with a pressure feed system. No temperature control is incorporated, for measuring the lube oil temperature.

Copper shims of suitable size are provided in between the pump body and the engine frame, to vary the injection timing and its effect on the performance of the engine is studied, along with the change of injection pressures from 190 bar to 270 bar (in steps of 40 bar) using nozzle testing device.

The maximum injection pressure is restricted to 270 bar due to practical difficulties involved. Exhaust gas temperature (EGT) is measured with thermocouples made of iron and iron-constantan.

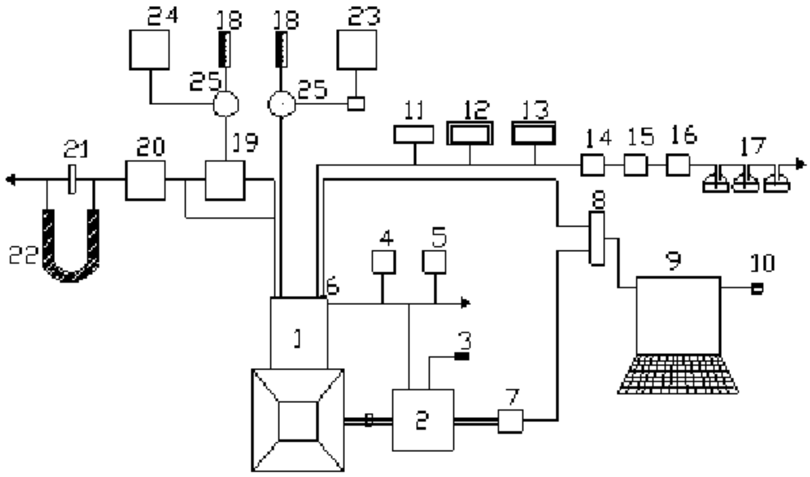


Figure 2: Experimental Set-up

1. Engine, 2.Electical Dynamo meter, 3.Load Box, 4. Outlet jacket water temperature indicator, 5.Outlet-jacket water flow meter Orifice meter, 6. Piezo-electric pressure transducer, 7. TDC encoder 8.Console, 9. Pentium Personal Computer, 10. Printer, 11.Exhaust gas temperature indicator,12.AVL Smoke meter, 13. Netel Chromatograph NOx Analyzer, 14. Filter, 15.Rotometer, 16.Heater, 17. Round bottom flask containing DNPH solution, 18.Burette, 19. Variable jet carburetor, 20. Air box, 21.Orifice meter, 22. U-tube water manometer, 23.Vegetable oil tank, 24.Alcohol tank, 25. Three-way valve

Pollution levels of smoke and NO_x are recorded by AVL smoke meter and Netel Chromatograph NOx analyzer respectively at the peak load operation of the engine. With alcohol-vegetable mixture operation, the major pollutant emitted from the engine is aldehydes. These aldehydes are carcinogenic in nature, which are harmful to human beings. The measure of the aldehydes is not sufficiently reported in the literature. DNPH method [6] is employed for measuring aldehydes in the experimentation. The exhaust of the engine is bubbled through 2,4 dinitrophenyl hydrazine (2,4 DNPH) solution. The hydrazones formed are extracted into chloroform and are analyzed by employing high performance liquid chromatography (HPLC) to find the percentage concentration of formaldehyde and acetaldehyde in the exhaust of the engine.

Piezo electric transducer, fitted on the cylinder head to measure pressure in the combustion chamber is connected to a console, which in turn is connected to Pentium personal computer. TDC encoder provided at the extended shaft of the dynamometer is connected to the console to measure the crank angle of the engine. A special P-θ software package evaluates the combustion characteristics such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise (MRPR) from the signals of pressure and crank angle at the peak load operation of the engine.

RESULTS AND DISCUSSIONS

Performance Parameters

Investigations are carried out with the objective of determining the factors that would allow maximum use of ethanol in diesel engine with best possible efficiency at all loads.

The variation of brake thermal efficiency (BTE) with brake mean effective pressure (BMEP) with different percentages of ethanol induction in conventional engine (CE) at 27^{o}bTDC and at an injection pressure of 190 bar, is shown in Figure.3. Variation of BTE with BMEP with pure diesel operation on CE is also shown for comparison purpose. BTE increased at all loads with 35% ethanol induction and with the increase of ethanol induction beyond 35%, it decreased at all loads in CE when compared with CE with diesel operation (standard diesel). The reason for improving the efficiency with the 35% ethanol induction is because of improved homogeneity of the mixture with the presence of ethanol, decreased dissociated losses, specific heat losses and cooling losses due to lower combustion temperatures. This is also due to high heat of evaporation of ethanol, which caused the reduction the gas temperatures resulting in a lower ratio of specific heats leading to more efficient conversion of heat into work. Induction of ethanol resulted in more moles of working gas, which caused high pressures in the cylinder. The observed increased in the ignition delay period would allow more time for fuel to vaporize before ignition started. This means higher burning rates resulted more heat release rate at constant volume, which is a more efficient conversion process of heat into work.

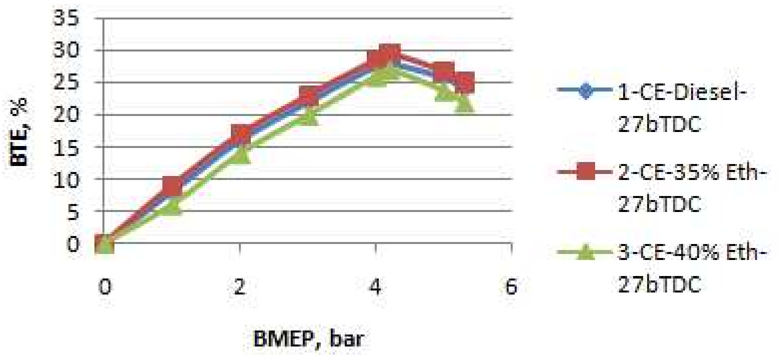


Figure 3: Variation of Brake Thermal Efficiency (BTE) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) at Different Percentages of Ethanol Induction

Figure 4 shows the variation of BTE with BMEP with different percentages of ethanol induction in LHR engine at the recommended injection timing and pressure. LHR engine showed an improvement in the performance with the carbureted ethanol at all loads when compared to the standard diesel engine. This is due to recovery of heat from the hot insulated components of LHR engine due to high latent heat of evaporation of the ethanol, which lead to increase in thermal efficiency. The maximum induction of ethanol is 50% in LHR engine, which showed improvement in the performance at all loads when compared to standard diesel engine. However when the ethanol induction is increased more than 50% in LHR engine, BTE is deteriorated at all loads when compared with standard diesel.

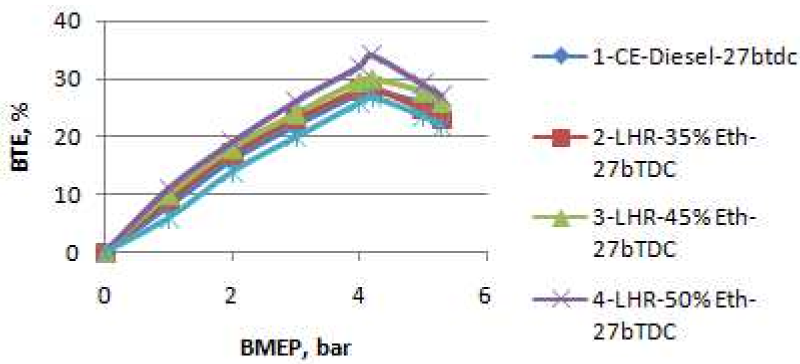


Figure 4: Variation of Brake Thermal Efficiency (BTE) with Brake Mean Effective Pressure (BMEP) in Low Heat Rejection (LHR) Engine at Different Percentages of Ethanol Induction

The optimum injection timings are at 33°bTDC for CE, and at 31°bTDC for LHR engine with pure diesel operation [16]. Similar trends are observed on the variation of BTE with BMEP in CE and LHR engine with alcohol-vegetable oil operation when the injection timings are advanced to 31°bTDC in LHR engine and 33°bTDC in CE as in the case of 27°bTDC in both versions of the engine. However, the maximum induction of alcohol is limited to 45% in the LHR engine at 31°bTDC against 50% induction at 27°bTDC, while maximum induction of alcohol is the same in CE at 33°bTDC as in the case of 27°bTDC. Ethanol is inducted at these respective injection timings for CE and LHR engine.

The variation of BTE with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at a pressure of 190 bars is shown in Figure.5. LHR engine with 45% ethanol induction at its optimum injection timing showed improved performance at all loads when compared with other versions of the engine. This is due to higher amount of ethanol substitution and improved combustion at advanced injection timing caused better evaporation leading to produce higher BTE.

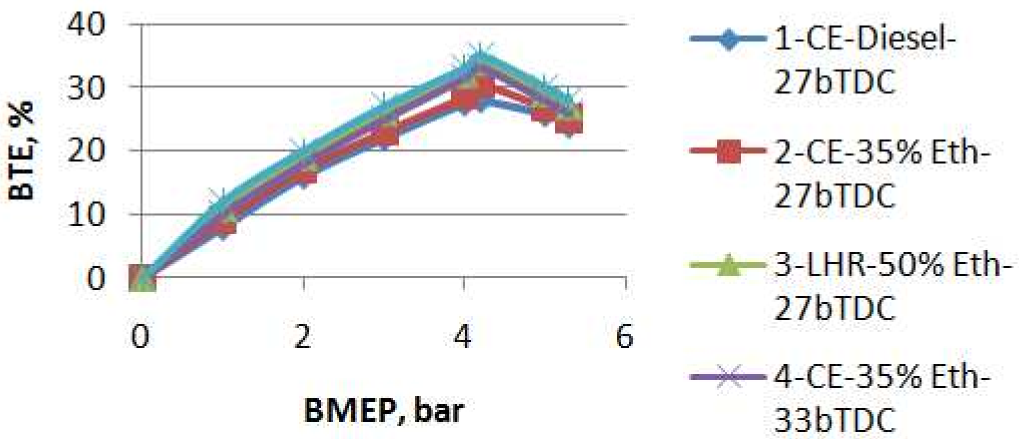


Figure 5: Variation of BTE with BMEP with Maximum Percentage of Ethanol Induction in CE and LHR Engine at Recommended and Optimum Injection Timings

There is a limitation to use ethanol due to low cetane number and having higher self-ignition temperature than vegetable oil to use in CE without increasing injection pressure because as percentage of ethanol increases more heat is utilized to evaporate alcohol fuels and less heat is available to evaporate vegetable oil. Therefore a major quantity of alcohol which burns late in the expansion stroke, will not be fully utilized. In order to avert this, injection pressure is increased, which reduces fuel droplet size, increases surface to volume ratio and requires comparatively less heat to evaporate vegetable oil droplet.

The trend exhibited by both versions of the engine with dual fuel operation at higher injection pressure of 270 bars is similar to the corresponding to the injection pressure of 190 bars. However, the maximum induction of alcohol is 40% in CE at an injection pressure of 270 bars against 35% at 190 bars, while maximum alcohol induction remained same with LHR engine at 270 bars as in the case of 190 bars.

Figure.6 shows bar charts which represents the variation of brake specific energy consumption (BSEC) at peak load operation with different versions of the engine at maximum induction of ethanol at recommended and optimum injection timings. BSEC decreased with the increase of ethanol induction, as higher amount of alcohol substitution caused better evaporation and produced lower BSEC in both versions of the engine. BSEC is lower in LHR engine at its optimum injection timing, which shows the suitability of the engine for alternate fuels. It also decreased with the increase of injection pressures in both versions of the engine. This is due to early initiation of combustion with improved fuel spray characteristics.

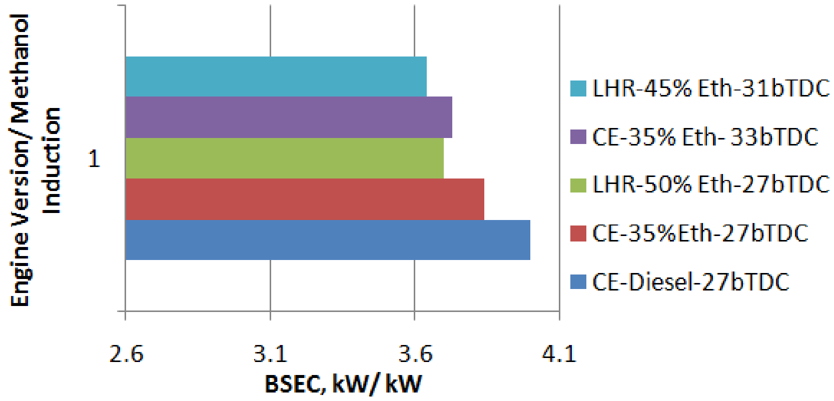


Figure 6: Bar Chart Showing the Variation of Brake Specific Energy Consumption (BSEC) at Peak Load Operation with Induction of Ethanol in CE and LHR Engine at Recommended and Optimum Injection Timings

Variation of exhaust gas temperature (EGT) with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at an injection pressure of 190 bars is shown in Figure.7. The magnitude of EGT decreased with the increase of percentage of ethanol induction in both versions of the engine. At the recommended injection timing, the magnitude of EGT is lower in CE with 35% induction of ethanol induction at all loads when compared with standard diesel engine. Lower exhaust gas temperatures are observed in the LHR engine with 50% ethanol induction when compared with CE with 35% ethanol induction. This showed that the performance of the LHR engine is improved with 50% ethanol induction over CE with 35% ethanol induction. EGT further decreased, when the injection timings are advanced in both versions of the engine. This is due to increase of thermal efficiency, reduction of coolant load and decrease of gas temperatures.

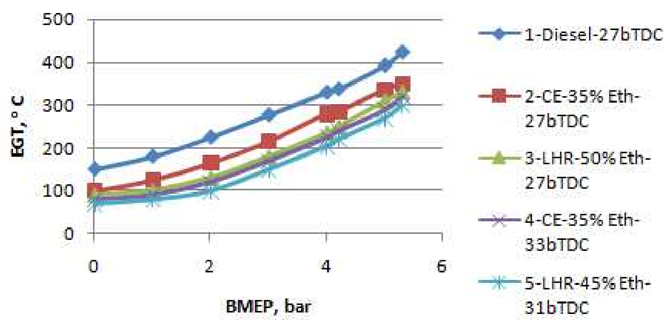


Figure 7: Variation of Exhaust Gas Temperature (EGT) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

Variation of coolant load (CL) with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at an injection pressure of 190 bars is shown in Figure.8. Coolant load is less in both versions of the engine at different percentages of ethanol induction at all loads when compared with pure diesel operation on CE. This is due to the reduction of gas temperatures with ethanol induction. Cooling load is less in the LHR engine with 50% ethanol induction when compared with CE with 35% ethanol induction at all loads. This is due to the insulation provided in LHR engine. Cooling load increased in CE and decreased in the LHR engine with the advancing of injection timing and increase of injection pressure. This is due to increase of gas temperatures in CE and decrease of the same in LHR engine, when the injection timing is advanced.

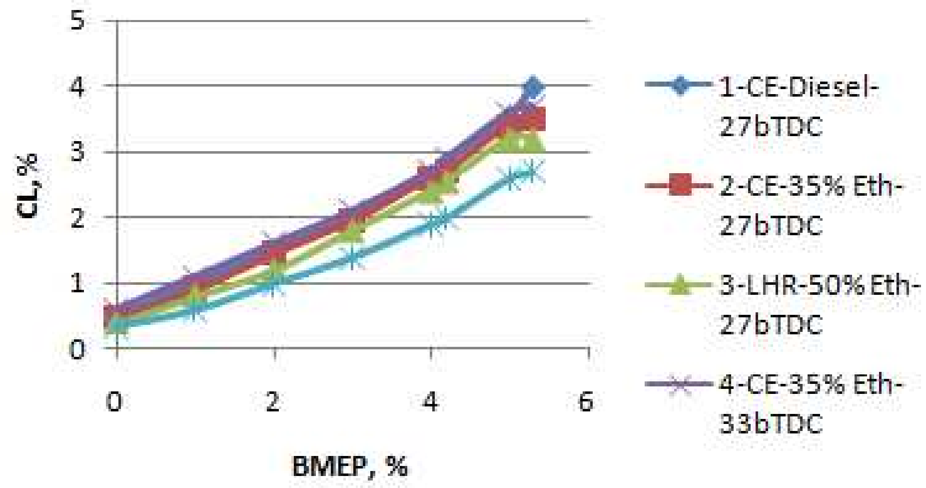


Figure 8: Variation of Coolant Load (CL) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

Variation of volumetric efficiency (VE) with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at an injection pressure of 190 bars is shown in Figure.9. VE decreased marginally in both versions of the engine with the dual fuel operation when compared with pure diesel operation on CE, as percentage of alcohol induction increased, the amount of air admitted into the cylinder of the engine reduced. However, CE with different percentage of ethanol induction showed higher volumetric efficiency when compared with LHR engine. This is because of increase of temperatures of insulated components in LHR engine, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle is lower. VE increased marginally with the increase of injection pressure in both versions of the engine. This is due to improvement of air utilization and combustion with the increase of injection pressure. However, these variations were very small.

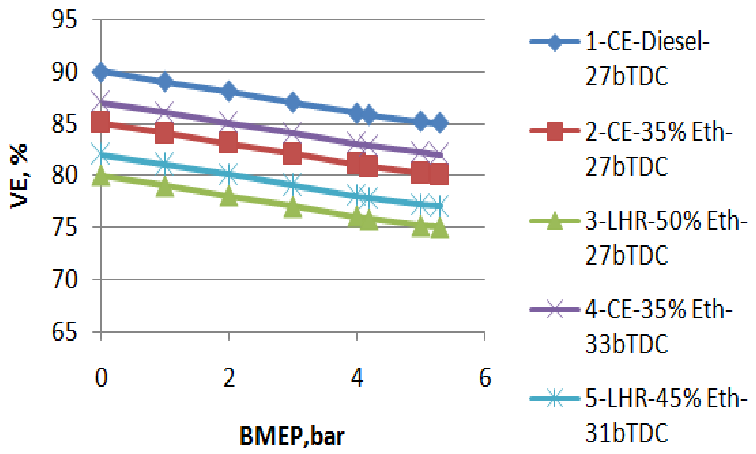


Figure 9: Variation of Volumetric Efficiency (VE) with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

Pollution Levels

Figure 10 shows the variation of smoke levels with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at an injection pressure of 190 bars. It is seen that for the same load, the smoke density decreased with induction of alcohol. The combustion of injected fuel in case of pure vegetable oil operation is predominantly one of oxidation of products of destructive decomposition. In this case, there are greater chances of fuel cracking and forming carbon particles. On the other hand, the combustion of alcohol is predominantly a process of hydroxylation and the chances of fuel cracking are negligible. Ethanol does not contain carbon-carbon bonds and therefore cannot form any un-oxidized carbon particles or precursor to soot particles. One of the promising factor for reducing smoke levels with the alcohols is they contained oxygen in their composition which helped to reduce soot density. Soot emissions increased linearly with the increase of carbon to hydrogen atoms (C/H) ratio provided the equivalence ratio is not altered. This is because higher C/H lead to more concentration of carbon dioxide, which would be further, reduced to carbon. Consequently, induction of alcohol reduced the quantity of carbon particles in the exhaust gases as the magnitudes of C/H for diesel fuel, vegetable oil and ethanol are 0.45, 0.83 and 0.25 respectively. Lower smoke levels are observed in both versions of the engine in dual fuel mode when compared with pure diesel operation on CE. LHR engine with 60% ethanol induction showed lower smoke levels when compared with CE with 35% ethanol induction. Smoke levels decreased with the increase of ethanol induction in both versions of the engine. In dual fuel operation, smoke levels further decreased with the advancing of the injection timing and with increase of injection pressure in both versions of the engine, due to efficient combustion at higher injection pressures, which improved the atomization hence faster rate of combustion and shorter combustion duration at the advanced injection timings caused to reduce the smoke density in both versions of the engine.

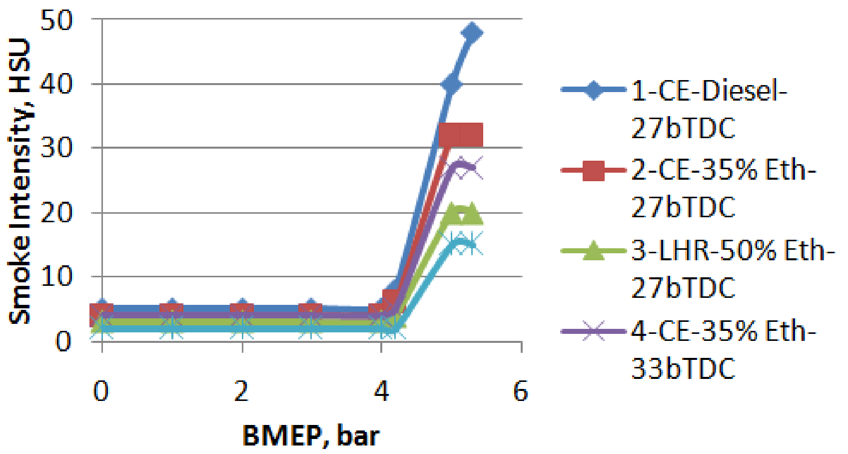


Figure 10: Variation of Smoke Levels in Hartridge Smoke Unit With Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

Variation of NOx levels with BMEP in CE and LHR engine with maximum induction of ethanol at recommended and optimum injection timings and at an injection pressure of 190 bars is shown in Figure.11. NOx emissions decreased with the increase of percentage of ethanol induction in both versions of the engine, due to lower combustion temperatures. The low value of C/H ratio in ethanol has indirect effect in reducing oxygen availability in the gases, which leads to the reduction of NOx.

However, LHR engine with different percentages of ethanol induction showed higher NOx levels compared with CE with 35% ethanol induction, due to increase of gas temperatures in LHR engine. NOx levels further decreased with the increase of ethanol induction in both versions of the engine.

NOx levels increased marginally in CE while they decreased in LHR engine with the advancing of the injection timing and with the increase of injection pressure. This is due to reduction of gas temperatures in the LHR engine at 31^{o}bTDC.

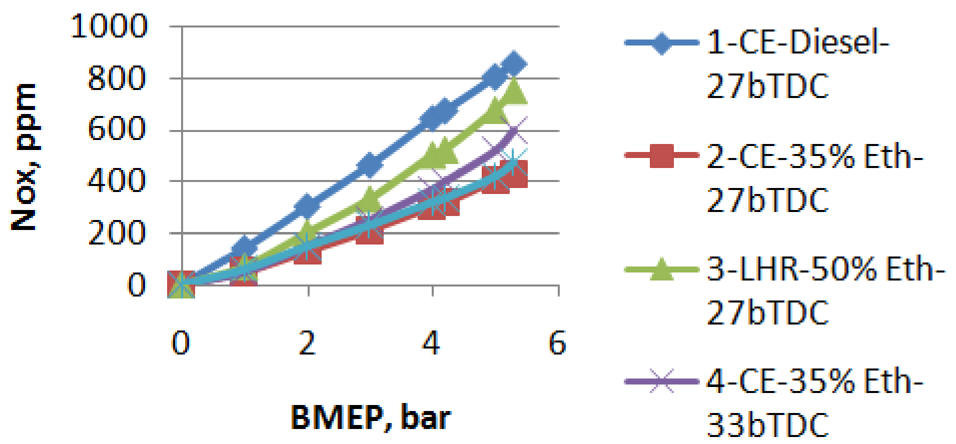


Figure 11: Variation of Nox Levels with Brake Mean Effective Pressure (BMEP) in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

These aldehydes are responsible for pungent smell of the engine and affect the human beings when inhaled in the large quantities. The volatile aldehydes are eye and respiratory tract irritants. Though Government legislation has not been pronounced regarding the control of aldehyde emissions, when more and more alcohol engines are coming to existence severe measures the controlling of aldehydes emitted out through the exhaust of the alcohol run engines will have to be taken as serious view. Figure 12 (a) shows the variation of formaldehyde concentration while Figure.12 (b) acetaldehyde concentration in CE and LHR engine at recommend injection timing and optimum injection timing at an injection pressure of 190 bar with maximum induction of ethanol. It could be seen that aldehyde emissions are low with pure diesel operation in both CE and LHR engine. Formaldehyde emissions increased drastically with ethanol induction in both CE and LHR engine. With increased induction of ethanol upto 50%, CE registered very high value of formaldehyde emissions in the exhaust, which showed the significant reduction in LHR engine. Hot environment of LHR engine completed combustion reactions and reduced the emissions of intermediate compounds, aldehydes. Hence it is concluded that LHR engine is more suitable for alcohol engines in comparison with pure diesel operation. Advanced injection timing and increase of injection pressure also improved the combustion performance in LHR engine by reducing the intermediate compounds like formaldehyde and acetaldehydes.

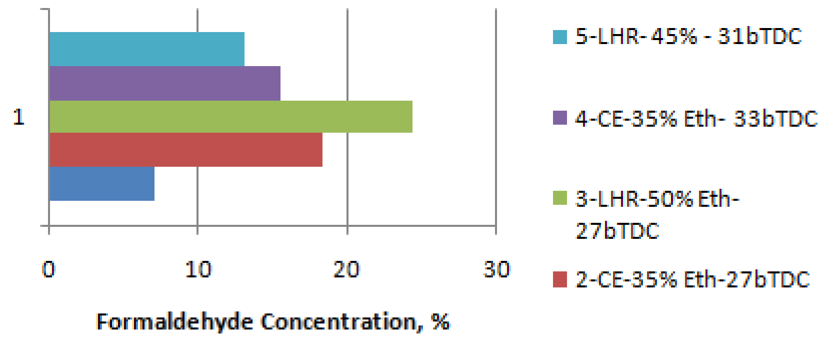


Figure 12: (A) Formaldehyde Concentraiton

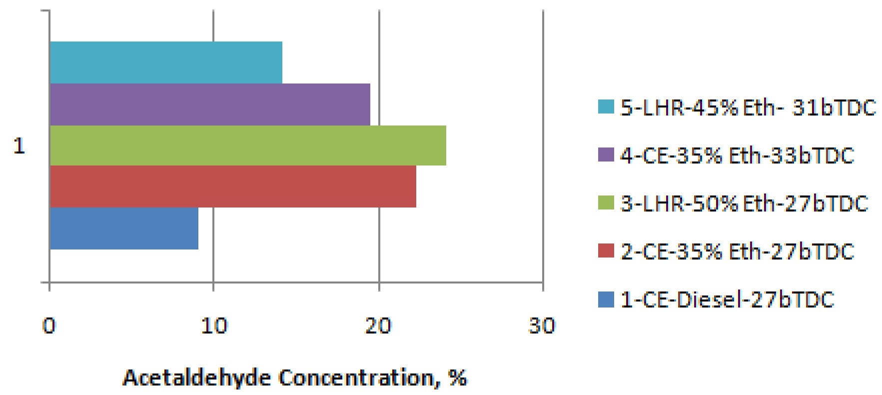


Figure 12: (B) Acetaldehyde Concentraiton

Figure 12: Variation of Aldehyde Concentration in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Induction of Ethanol

Combustion Characteristics

Variation of combustion parameters like PP, TOPP and MRPR with maximum induction of ethanol induction in different versions of the engine at recommended injection timing and optimum injection timing and at an injection pressure of 190 bars are represented by Figure. 13(a), 13(b) and 13(c), respectively. From Figure, 13(a), it can be noticed that the magnitude of PP increased with increase of ethanol induction in both versions of the engine. The magnitude of PP increased with advancing of the injection timing in both versions of the engine, with ethanol induction. With the same amount of ethanol induction, LHR engine exhibited higher PP compared with CE with 50% of ethanol induction at 27^{o}bTDC and at injection pressure of 190 bar. This is due to increased amount of ethanol with LHR engine. With maximum induction of ethanol, LHR engine at 31^{o}bTDC produced higher PP compared with CE at 33^{o}bTDC.From the Figure.13 (b), it can be noticed that magnitude of TOPP decreased with the increase of ethanol induction with both versions of the engine.

When the ethanol induction is increased to 50% in LHR engine, the magnitude of TOPP is lower (shifted towards TDC) when compared with CE with 35% ethanol induction. This is once again confirmed by the observation of higher PP and lower TOPP in LHR engine with dual fuel mode, that the performance of LHR engine with 50% alcohol induction is improved over CE with 35% ethanol induction.

The magnitude of TOPP decreased with advancing of the injection timing with both versions of the engine. From the Figure.13©, it can be observed that LHR engine showed higher MRPR when compared with CE at different injection timing. This is due to higher amount of ethanol induction in LHR engine. MRPR increased with the advancing of the injection timing in both versions of the engine. These combustion characteristics improved with increase of injection pressure.

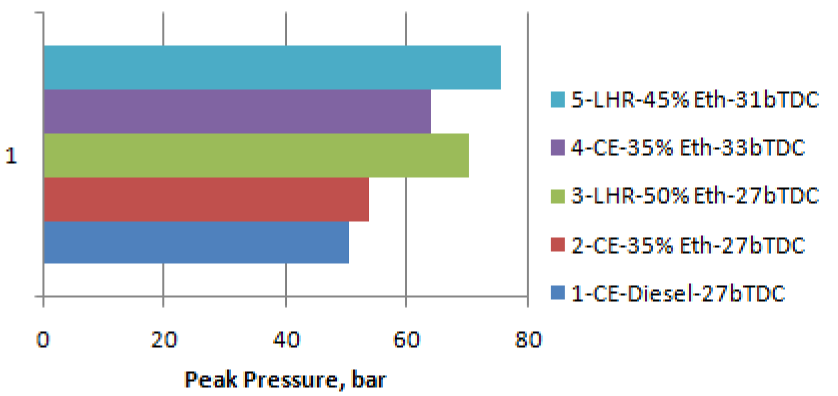


Figure 13: (a) Peak Pressure

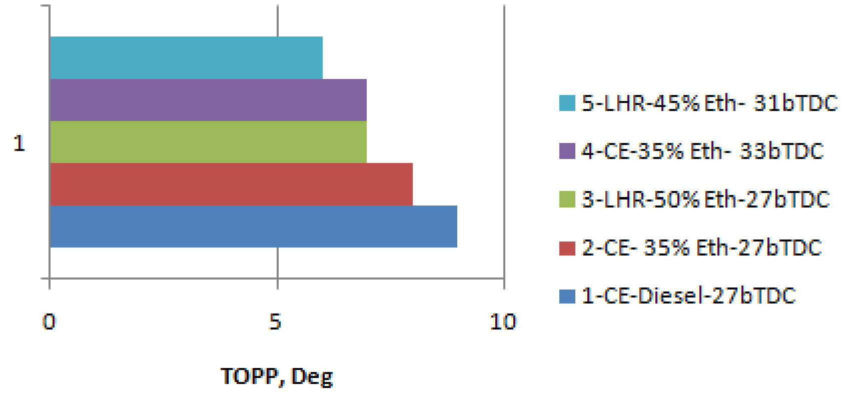


Figure 13: (b) TOPP

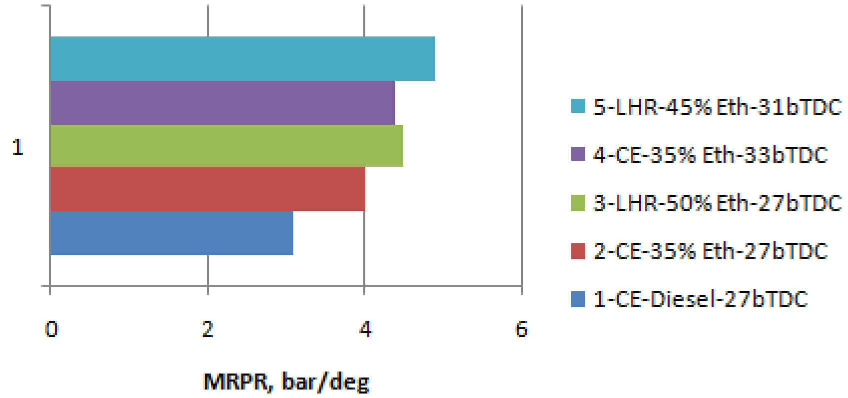


Figure 13: (c)MRPR

Figure 13: Variation of Combustion Parameters in Conventional Engine (CE) and Low Heat Rejection (LHR) Engine at Recommend Injection Timing and Optimized Injection Timings with Maximum Percentage of Ethanol Induction

CONCLUSIONS

Maximum induction of alcohol was 35% on mass basis with best possible efficiency at all loads in CE while it is 50% in the LHR engine. LHR engine with 50% alcohol induction showed improved performance when compared to CE with 35% alcohol induction**.** The maximum induction of alcohol is 35% in CE at 33^{o}bTDC, while it is 45% in LHR engine at 31^{o}bTDC. Performance, pollution levels (smoke, NOx and aldehyde levels) and combustion characteristics improved in both versions of the engine with maximum induction of alcohol when the injection timings are advanced and with the increase of injection pressure.

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